1 Basics of thermodynamics

Page 19: At the end of Section 1.7 the following paragraph is added:

Note that water vapour is only a small component of humid air. From Dalton's law we conclude that the water vapour pressure must be much lower than the atmospheric pressure and that water can vaporise at temperatures much lower than 100 °C. This process is called evaporation and will is discussed in Section 4.2 on page 102.

Page 19: After the Section 1.7, two new sections are added:

1.8 Building energy balance

In Section 1.5 we have learned that energy is spontaneously transferred from the system with the higher temperature to the system with the lower temperature, and that the transferred energy is called heat. The undesirable consequence of this process is that a building loses energy during the winter (cold) period and gains energy during the summer (warm) period, as shown in Fig. 1.12 (a). In order to keep the temperature inside the building constant, we need to replace the energy lost during the winter period and dispose of the energy gained during the summer period, that is, to establish the energy balance.

However, it is impossible for heat to flow spontaneously from the system with the lower temperature to the system with the higher temperature. This obvious fact is the consequence of the second law of thermodynamics, which we will also study in Section 1.9.1. Nevertheless, there are several viable strategies for achieving an energy balance, as discussed below.

One possible strategy to replace the lost energy is the use of *furnace*, as shown in Fig. 1.12 (b). Chemical energy in the form of fossil fuels (heating oil, gas, coal) or biomass (wood) is first fed into the furnace, where it is converted into internal energy by combustion. This energy is then supplied to the building in the form of heat. The energy conversion is almost perfect, so that the efficiency of the furnace is almost 100 %.

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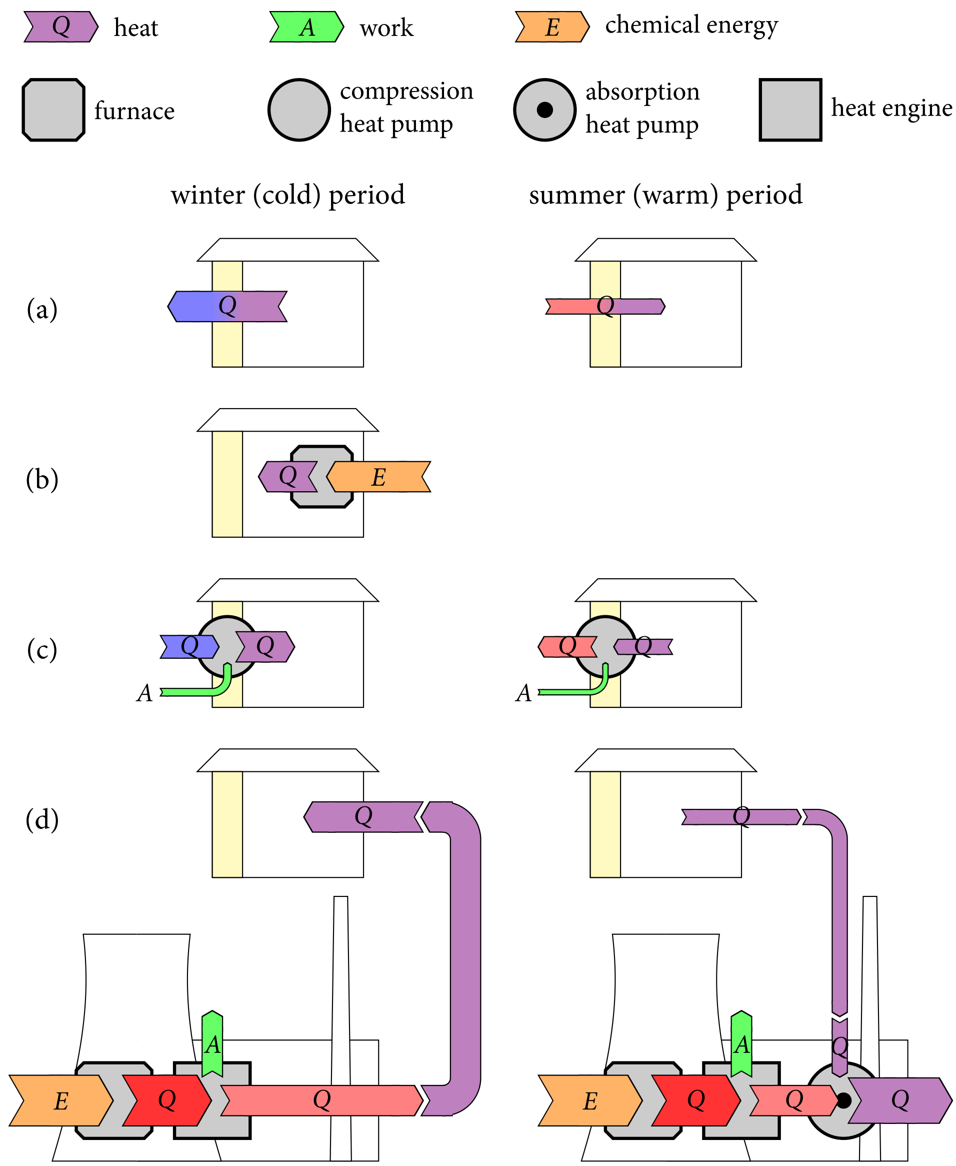

Figure 1.12: Building energy balance. The energy lost during the winter period can be replaced by a furnace, a heat pump or in conjunction to a CHP system. The energy gained during the summer period can be disposed of by a heat pump or in conjunction to a CCHP system. Red and blue tones indicate the temperature at which heat is transferred, with the red and blue tones corresponding to the temperature above and below room temperature, respectively.

Another possible strategy for establishing the energy balance is the use of the *heat pump*. A heat pump transfers heat from the lower temperature to the higher temperature, which seems to contradict the second law of thermodynamics. However, heat can flow from a lower to a higher temperature without violating the second law of thermodynamics if additional energy in the form of work or heat is provided to support the process.

In *compression heat pumps* we provide the work to reverse the heat flow, as shown in Fig. 1.12 (c). The amount of transferred energy, that is heat, is larger than the provided work, which means that the efficiency is above 100 %. In fact, the typical efficiency of a heat pump is many times larger than that of a furnace. Another advantage compared to the furnace is that it can be used both to replace the energy lost during the winter period and to dispose of the energy gained during the summer period. Compression heat pumps, which are capable of transferring heat only from the inside to the outside of the building and are therefore only used during the summer period, are generally referred to as air conditioners.

Info box

Heat pumps are more efficient than furnaces because they do not convert between forms of energy, but move energy between two environments.

The energy balance can also be achieved in conjunction to power stations, as shown in Fig. 1.12 (d). Power stations are industrial facilities that generate electrical energy. They basically consist of a furnace that converts chemical energy into internal energy, a heat engine that provides work when heat flows through it, and an electrical generator that converts work into electrical energy.

The by-product of the process is waste heat, which the power station can either release to the environment or supply to a building, as shown on the left side in Fig. 1.12 (d). In the latter case, the use of a power station is referred to as cogeneration or Combined Heat and Power (CHP).

The waste heat from the heat engine can also be used for cooling, as shown on the right side in Fig. 1.12 (d). The *absorption heat pump* absorbs some additional heat when the main heat flow passes through it. In this case, the use of a power station is referred to as trigeneration or Combined Cooling, Heat and Power (CCHP).

Note that in this section we have used work and electrical energy interchangeably for simplicity. In fact, work is converted into electrical energy by an electric generator, and electrical energy is converted into work by an electric motor. However, because the efficiency of both devices is virtually 100 %, this does not affect the results of our considerations. The exact functioning of heat pumps and heat engines is explained in more detail in Section 1.9.

1.9 Heat pumps and engines

In Section 1.8 we mentioned that the building energy balance can be achieved by using a compression heat pump, a device that in various forms becomes a standard building installation. Now we will take a closer look at the operating principle of the compression heat pump, which until the end of this section is simply referred to as a heat pump. For comparison, we will also look at the operating principle of its close sibling, the heat engine.

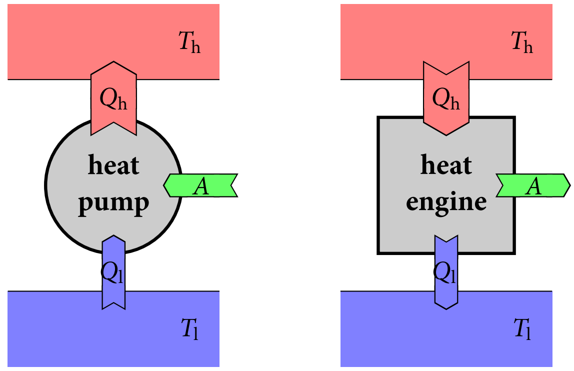

Figure 1.13: The principles of the heat pump (left) and the heat engine (right). When the heat pump is supplied with work, it transfers heat from an environment at lower temperature T_l to an environment at higher temperature T_h. When the heat engine is brought into contact with two environments at different temperatures T_h and T_l, it generates work.

1.9.1 Introduction

As we have already mentioned, heat pumps and heat engines exchange energy with the environment in the form of both heat and work.

Heat pump

Info box
A heat pump can transfer heat from a lower to a higher temperature because it is supplied with work.

The *heat pump* (Fig. 1.13, left) is a device that transfers heat from the environment at a lower temperature T_l to the environment at a higher temperature T_h. Because the direction of heat flow is opposite to the direction of spontaneous heat flow, work A must be provided, usually by an electric motor. The internal energy of the heat pump remains unchanged in the process, so we can write

$$A + Q_l = Q_h.$$

The heat pump can be used in heating mode to heat the environment at a higher temperature (heating system), or in cooling mode to cool the environment at a lower temperature (refrigerator, air conditioner). If we define the *coefficient of performance* COP as the ratio of what is gained, the heat transferred from or to the environment, to what is given, the work provided by an electric motor, we obtain different coefficients in heating and cooling mode.

In cooling mode, the gain corresponds to the heat extracted from the environment at a lower temperature Q_l, so the coefficient is

$$\mathrm{COP}_{\mathrm{cooling}} = \frac{Q_l}{A} = \frac{Q_l}{Q_h - Q_l}, \tag{1.33}$$

where Q_h is heat disposed to the environment at a higher temperature.

In heating mode, on the other hand, the gain corresponds to the heat provided to the environment at a higher temperature Q_h, so the coefficient is

$$\mathrm{COP}_{\mathrm{heating}} = \frac{Q_h}{A} = \frac{Q_h}{Q_h - Q_l}, \tag{1.34}$$

where Q_l is heat absorbed from the environment at a lower temperature. The coefficient of performance is larger in heating mode than in cooling mode.

Ideally, the heat could be transferred spontaneously from the environment at T_l to the environment at T_h, without the need for work, that is $A = 0\,\mathrm{J}$; In this case the coefficient would be infinite. The fact that it is impossible to design a machine that continuously transfers heat from the environment at lower temperature to the environment at a higher temperature without the input of work is a distinct form of the *second law of thermodynamics*.

Heat engine

The *heat engine* (Fig. 1.13, right) is a device that generates work from the heat. Two environments at a higher temperature T_h and at a lower temperature T_l must be provided. The heat flows spontaneously from the higher to the lower temperature, and this fact is used to generate work A in the process. The internal energy of the heat pump remains unchanged in the process, so we can write

$$Q_h = A + Q_l.$$

We define *efficiency* η as the ratio of what is gained, the work A, to what is given, the heat supplied by the environment at a higher temperature Q_h

$$\eta = \frac{A}{Q_h} = \frac{Q_h - Q_l}{Q_h} = 1 - \frac{Q_l}{Q_h}, \tag{1.35}$$

where Q_l is the heat disposed to the environment at a lower temperature. Ideally, the work could be obtained without the intervention of the environment at T_l, that is $Q_l = 0\,\mathrm{J}$; In this case all of the heat provided Q_h would be converted into work and the efficiency would be $\eta = 1$. The fact that it is impossible to design a machine that continuously converts all the heat from the environment into work is a distinct form of the *second law of thermodynamics*. The efficiency is therefore always $\eta < 1$.

1.9.2 Gas processes

The intention of this book is to present a simple heat pump and heat engine using the ideal gas as a working medium. But before that, we have to familiarise ourselves with four ideal gas processes:

1. The *isothermal process* is the process in which the temperature of the gas T is constant. If we put constant temperature into the ideal gas law (1.12), we see that the product of pressure and volume is constant

$$p\,V = \frac{mRT}{M} = \text{constant}. \tag{1.36}$$

2. The *isobaric process* is the process in which the pressure of the gas p is constant. If we put constant pressure into the ideal gas law (1.12), we see that the quotient of volume by temperature is constant

$$\frac{V}{T} = \frac{mR}{Mp} = \text{constant}. \tag{1.37}$$

3. The *isochoric process* is the process in which the volume of the gas V is constant. If we put constant volume into the ideal gas law (1.12), we see that the quotient of pressure by temperature is constant

$$\frac{p}{T} = \frac{mR}{MV} = \text{constant}. \tag{1.38}$$

4. The *adiabatic process* is the process in which the gas does not exchange heat Q with the environment. In this case all gas state variables (p, V, T) change and the relationship between them can be determined by the total differential of the ideal gas law (1.12)

$$p\,\mathrm{d}V + V\,\mathrm{d}p = \frac{m}{M} R\,\mathrm{d}T.$$

If we put $\mathrm{d}Q = 0$ in the first law of thermodynamics (1.22) and combine it with (1.24), we get

$$m\,c_V\,\mathrm{d}T + p\,\mathrm{d}V = 0.$$

Eliminating $\mathrm{d}T$ from these two expressions and using (1.26c) we get

$$p\,\mathrm{d}V + V\,\mathrm{d}p = -\frac{R}{M\,c_V} p\,\mathrm{d}V = -\frac{c_p - c_V}{c_V} p\,\mathrm{d}V = (1-\gamma)\,p\,\mathrm{d}V,$$

where γ is the ratio of heat capacities

$$\gamma = \frac{c_p}{c_V}.$$

Ordering and integrating the expression we get

$$\begin{aligned} \frac{\mathrm{d}p}{p} + \gamma \frac{\mathrm{d}V}{V} &= 0, \\ \ln p + \gamma \ln V &= \text{constant}, \\ p\,V^{\gamma} &= \text{constant}. \end{aligned} \tag{1.39}$$

With the help of the ideal gas law (1.12), we can also write the relationship between temperature and volume as

$$T\,V^{\gamma-1} = \text{constant}. \tag{1.40}$$

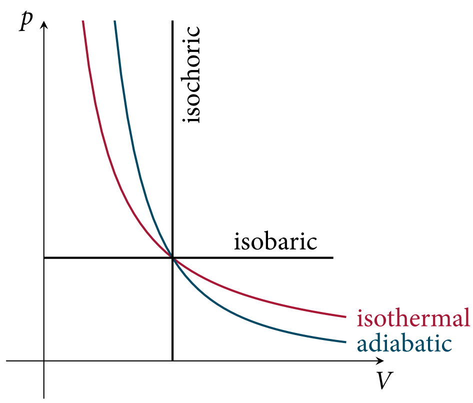

Figure 1.14: The curves for four ideal gas processes in the PV diagram. Note that the adiabatic curve is steeper than the isothermal curve.

The gas processes are normally examined in the pressure versus volume plot, called PV diagram. This is because in this case the area under the curve represents the work done by the gas (1.13d). The PV diagram for all four processes is shown in Fig. 1.14. Because $c_p > c_V$ (1.25c), $\gamma > 1$ and the adiabatic curve is steeper than the isothermal curve.

At the end of this section, we will derive the expression for the modulus of compression K for the adiabatic processes that we will need in Section 6.1.1 on page 163. If we take the definition of the modulus as

$$\Delta p = -K\frac{\Delta V}{V},$$
$$\frac{\Delta p}{\Delta V} = -\frac{K}{V},$$

write it in differential form and use $pV^\gamma = C$ (1.39), we get

$$\frac{K}{V} = -\frac{\mathrm{d}p}{\mathrm{d}V} = -\frac{\mathrm{d}}{\mathrm{d}V}(CV^{-\gamma}) = \gamma C V^{-\gamma-1} = \gamma\,\frac{p}{V}$$
$$\implies K = \gamma\,p. \tag{1.41}$$

1.9.3 Carnot cycle

Finally, we will demonstrate a simple heat pump and a simple heat engine using the ideal gas as a working medium. The gas is located in a cylinder fitted with a movable piston at one end. The process will consist of two isothermal and two adiabatic processes called the Carnot cycle. We have chosen this process because it is the most efficient. Note that proving this fact would go beyond the scope of this book.

Info box

All heat pumps and heat engines are based on cyclical thermodynamic processes.

Heat pump

The heat pump cycle is shown in Fig. 1.15 on the left, with the graphic representation above and the PV diagram below. The four processes are:

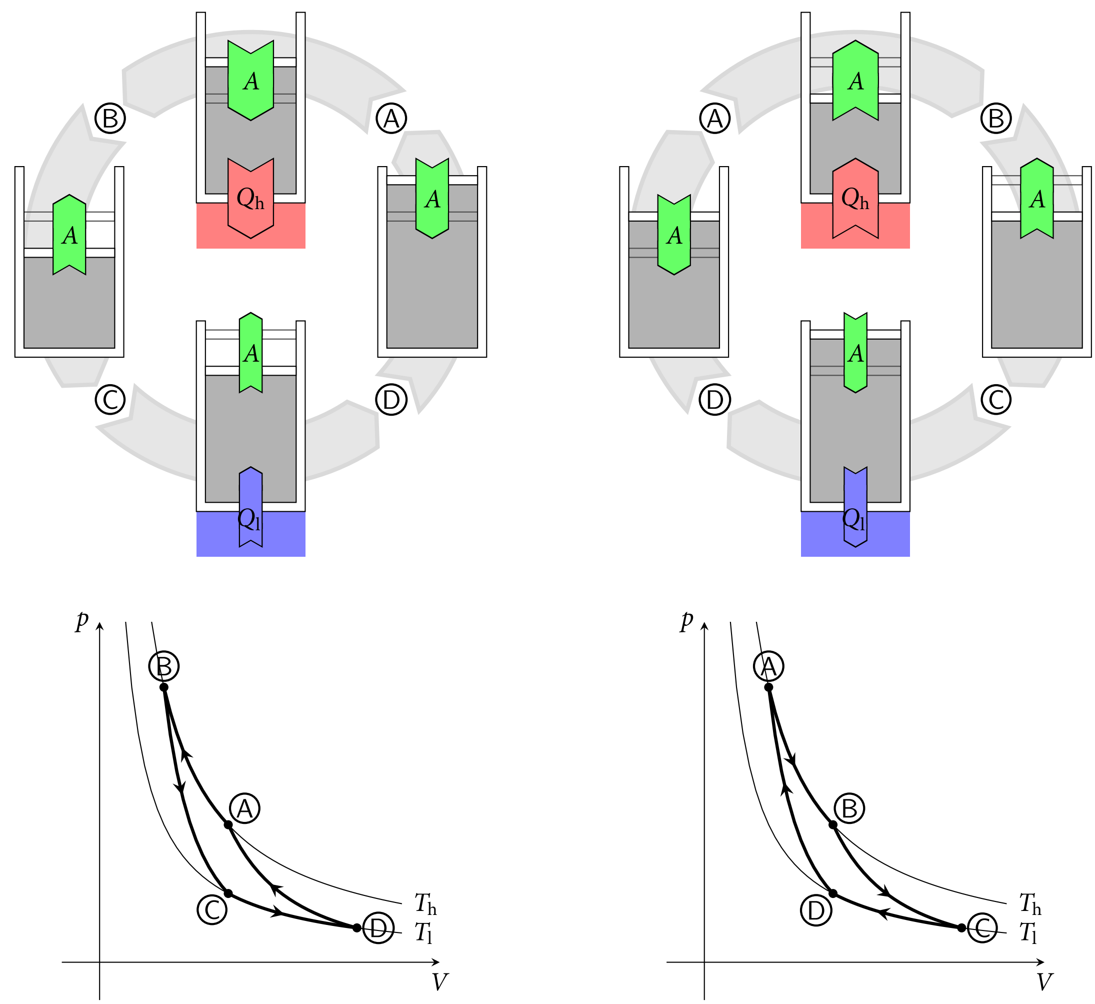

Figure 1.15: Heat pump (left) and heat engine (right) using Carnot cycle. The figures above are graphical representation, while the plots below are corresponding PV diagrams. Note that the heat pump and heat engine use the same cycle in the opposite direction.

1. Process Ⓐ to Ⓑ: The gas is in thermal contact with the environment at T_h. Because we compress the gas isothermally, its internal energy remains constant. Due to the first law of thermodynamics, this means that the provided work is completely converted into heat, which is released into the environment. This process can only take place up to a certain maximum pressure or minimum volume. Therefore we have to bring the gas into contact with the environment at T_l to absorb more energy and then return to point Ⓐ.
2. Process Ⓑ to Ⓒ: Before we bring the gas into contact with the environment at T_l, we have to cool it down. So we let it expand adiabatically and harness the work that is provided in this way. The process ends when the gas reaches the temperature T_l.
3. Process Ⓒ to Ⓓ: The gas is in thermal contact with the environment at T_l. We let the gas expand isothermally so that its internal energy remains constant. Due to first law of thermodynamics, this means that the heat that the gas absorbs from the environment is completely converted into the work, which we harness. This process can only take place up to a certain minimum pressure or maximum volume.
4. Process Ⓓ to Ⓐ: Before we bring the gas into contact with the environment at T_h, we have to warm it up. So we compress it adiabatically until the gas reaches temperature T_h. The cycle is complete.

The PV diagram allows us to estimate the work we have invested and harnessed. The work we have invested is the area under the Ⓓ–Ⓐ–Ⓑ curve, and the work we have harnessed is the area under the Ⓑ–Ⓒ–Ⓓ curve. So the work we have harnessed is smaller than the work we have invested, which is the condition to transfer the heat from the environment at T_l to the environment at T_h.

Heat engine

The heat engine cycle is shown in Fig. 1.15 on the right, with the graphic representation above and the PV diagram below. The four processes are:

1. Process Ⓐ to Ⓑ: The gas is in thermal contact with the environment at T_h. Because we let the gas expand isothermally, its internal energy remains constant. Due to the first law of thermodynamics, this means that all the heat that the gas absorbs from the environment is completely converted into the work that we harness. This process can only take place up to a certain maximum volume or minimum pressure, and we must return to point Ⓐ to gain more work. Of course it makes no sense to use the same path, because then we would be spending all the work we just harnessed. However, it turns out that if we compress gas at a lower temperature, we will need less work to get to that point. So we have to bring the gas into contact with the environment at T_l first.
2. Process Ⓑ to Ⓒ: Before we bring the gas into contact with the environment at T_l, we have to cool it down. So we let it to expand adiabatically and

harness the work that is provided in this way. The process ends when the gas reaches the temperature T_l.

3. Process Ⓒ to Ⓓ: The gas is in thermal contact with the environment at T_l. We compress the gas isothermally so that its internal energy remains constant. Due to the first law of thermodynamics, this means that the provided work is completely converted into heat, which is released into the environment. This process can only take place up to a certain maximum pressure or minimum volume.
4. Process Ⓓ to Ⓐ: Before we bring the gas into contact with the environment at T_h, we have to warm it up. So we compress it adiabatically, until the gas reaches temperature T_h. The cycle is complete.

The PV diagram allows us to estimate the work we have invested and harnessed. The work we have invested is the area under the Ⓒ–Ⓓ–Ⓐ curve, and the work we have harnessed is the area under the Ⓐ–Ⓑ–Ⓒ curve. So the work we have harnessed is larger than the work we have invested, which has been achieved by transferring heat from the environment at T_h to the environment at T_l.

Calculation

We will now calculate the heat that is absorbed or released by the gas during the isothermal processes. As already mentioned, due to the first law of thermodynamics, heat is equal to work

$$Q = A = \int_{V_1}^{V_2} p \, dV = \frac{mRT}{M} \int_{V_1}^{V_2} \frac{dV}{V} = \frac{mRT}{M} \ln\left(\frac{V_2}{V_1}\right),$$

where we have used the ideal gas law (1.12) and the expression for isothermal processes (1.36). Using the formula for Ⓐ–Ⓑ and Ⓓ–Ⓒ processes we obtain

$$|Q_h| = \pm \frac{mRT_h}{M} \ln\left(\frac{V_B}{V_A}\right), \qquad |Q_l| = \pm \frac{mRT_l}{M} \ln\left(\frac{V_C}{V_D}\right),$$

where the signs '+' and '-' stand for the heat pump and the heat engine respectively. On the other hand, we can connect volume pairs by the expression for adiabatic processes (1.40)

$$T_h V_B^{\gamma-1} = T_l V_C^{\gamma-1}, \qquad T_h V_A^{\gamma-1} = T_l V_D^{\gamma-1}.$$

Dividing the two equations we get

$$\left(\frac{V_B}{V_A}\right)^{\gamma-1} = \left(\frac{V_C}{V_D}\right)^{\gamma-1} \implies \frac{V_B}{V_A} = \frac{V_C}{V_D}.$$

Now we can calculate the ratio of the heats as

$$\frac{|Q_h|}{|Q_l|} = \frac{T_h \ln\left(\frac{V_B}{V_A}\right)}{T_l \ln\left(\frac{V_C}{V_D}\right)} = \frac{T_h}{T_l}. \tag{1.42}$$

If we insert (1.42) in (1.33) and (1.34), we get for coefficients of performance

$$\mathrm{COP}_{\mathrm{cooling}} = \frac{T_\mathrm{l}}{T_\mathrm{h} - T_\mathrm{l}}, \tag{1.43}$$

$$\mathrm{COP}_{\mathrm{heating}} = \frac{T_\mathrm{h}}{T_\mathrm{h} - T_\mathrm{l}}. \tag{1.44}$$

Info box
The coefficient of performance of the heat pump depends on the temperature difference between two environments.

Note that the coefficients depend on the temperature of the environments: If the temperature difference is smaller, the coefficient of performance is larger.

If we put (1.42) into (1.35) we get for efficiency

$$\eta = 1 - \frac{T_\mathrm{l}}{T_\mathrm{h}}. \tag{1.45}$$

Note that the efficiency depends on the temperature of the environments: If the temperature ratio is larger, the efficiency is also larger.

We have already pointed out the advantage of the Carnot cycle. Unfortunately, it is not technically feasible to design the heat pump and the heat engine with this cycle. Nevertheless, it helps us to reveal the operating principles of heat pumps and heat engines and provide us with the top theoretical values of the coefficient of performance and the efficiency.

Page 20: New problems were added:

1.12 An average adult human being consumes most energy on heat loss, through radiation, convection, perspiration, respiration and conduction (listed in order of importance). Calculate the power due to perspiration and respiration under assumptions:

- For perspiration and respiration, an average person consumes 0.8 L of liquid water at 20 °C per day, with the water being converted into vapour at 37 °C. The specific heat capacity of water is 4200 J/(kg K) and the specific heat of vaporisation of water at 37 °C is 2410 kJ/kg.
- An average person inhales and exhales 11 m^3 of air in the room, with the air being heated from 20 °C to 37 °C. The density of inhaled air is 1.2 kg/m^3 and the specific heat capacity is 1010 J/kg.

We will calculate other heat losses in problem 2.6 in Chapter 2. (26 W)

1.13 The room of rectangular floor plan with dimensions 10.0 m × 20.0 m and height 3.0 m contains dry air of density 1.165 kg/m^3 at temperature 30 °C. In the room we place a bucket with 12 L of liquid water at 5 °C. After a while, the temperatures of air and water will equalize at 20 °C. Calculate the mass of the evaporated water. Take the specific heat capacity of dry air to be 1005 J/(kg K), the specific heat capacity of liquid water to be 4200 J/(kg K) and the specific heat of vaporisation of water at 20 °C to be 2450 kJ/kg. (2.6 kg)

2 Heat transfer

Page 34: A new text was added before the last paragraph:

It should also be noted that in our above considerations we have assumed that the air temperature θ_e is equal to the temperature at the surface of the building element or the ground θ_{se}. As we will see in Chapter 3.1.1 on page 59, this is not the case, but the two temperatures can be related using Newton's law of cooling (2.30) and radiative heat exchange formula (2.46)

$$q = h_r(\theta_{se} - \theta_e) + h_c(\theta_{se} - \theta_e) = h(\theta_{se} - \theta_e),$$

where $h = 1/R_{se}$ is the convective-radiative surface coefficient. It can be shown that we can still use the same solution (2.26), albeit with different amplitude and time delay

$$\theta(z,t) = \bar{\theta} + A'\Delta\theta\, e^{-z/d} \sin\left(\frac{2\pi}{T}(t - t_0 - t_0')\right),$$

where the additional amplitude factor A' is

$$A' = \frac{1}{\sqrt{\left(1 + \frac{b}{h}\sqrt{\frac{\pi}{T}}\right)^2 + \left(\frac{b}{h}\sqrt{\frac{\pi}{T}}\right)^2}}$$

and the additional time delay t_0' is

$$t_0' = \frac{T}{2\pi} \operatorname{atan}\left(\frac{\frac{b}{h}\sqrt{\frac{\pi}{T}}}{1 + \frac{b}{h}\sqrt{\frac{\pi}{T}}}\right).$$

Larger effusivity results in a smaller additional amplitude factor and a larger additional time delay. This confirms the conclusion in Section 2.2.6 that material with the *larger effusivity* resists temperature change *at the surface* more strongly. The effect can be significant in the case of the building element. For example, for concrete we get $A' = 0.65$ and $t_0' = 1.2\,\mathrm{h}$.

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Page 40: A new figure was added:

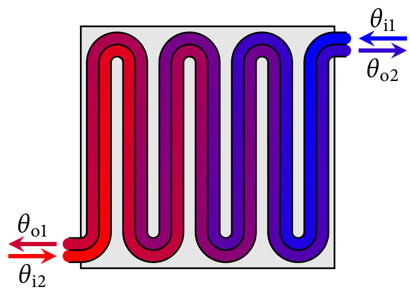

Figure 2.19: The simple model of the recuperator consists of two metal tubes in thermal contact. During the winter period, the initially warmer exhaust air and the initially colder fresh air flow through the tubes in opposite directions. In this process heat is transferred from the exhaust air to the fresh air. The inlet temperature of the fresh air is θ_{i1} and the outlet temperature θ_{o1}. The inlet temperature of the exhaust air is θ_{i2} and the outlet temperature θ_{o2}. Note that $\theta_{o1} < \theta_{i2}$ and $\theta_{o2} > \theta_{i1}$.

Page 40: A new text was added before the last paragraph:

The efficiency of heat exchangers can be easily evaluated when two identical fluids of the same mass and the same specific heat capacity are considered. In the ideal case, the temperature of the first, initially colder fluid increases from its own inlet temperature θ_{i1} to the inlet temperature of the second fluid θ_{i2} and vice versa (Fig. 2.19). The ideal heat transfer is therefore

$$Q_{id} = mc(\theta_{i2} - \theta_{i1}).$$

However, the outlet temperatures of the first and second fluids are θ_{o1} and θ_{o2}, respectively. The real heat transfer is therefore

$$Q_{re} = mc(\theta_{o1} - \theta_{i1}) = mc(\theta_{i2} - \theta_{o2}).$$

The efficiency η is the ratio of two, which gives

$$\eta = \frac{\theta_{o1} - \theta_{i1}}{\theta_{i2} - \theta_{i1}}. \tag{2.1}$$

The concept of recuperators, that is gas to gas heat exchangers, is closely related to the concept of *enthalpy exchangers*. In enthalpy exchangers, the metallic barrier between exhaust air and fresh air is replaced by the water-permeable membrane through which water vapour is transferred by diffusion (Section 4.5 on page 118). This modification reduces the efficiency of *sensible* heat transfer as expressed in (2.1). However, when water vapour is transferred from the warmer to the colder air, the total energy transfer can be increased because gaseous water molecules carry a high *latent* heat (Section 1.6 on page 13). Because the enthalpy accounts for both sensible and latent heat, these recuperator types are called enthalpy exchangers. The efficiency of enthalpy exchangers can be evaluated by using two fluids of the same mass to obtain

$$\eta = \frac{h_{o1} - h_{i1}}{h_{i2} - h_{i1}}, \tag{2.2}$$

where h_{i1} and h_{i2} are the inlet specific enthalpies of two gases and h_{o1} is the outlet specific enthalpy of the initially colder gas.

Page 52: New sentences were added after "…of area A_s, temperature T_s and emittance ε_s."

Here ε_s takes into account the emittance in the most important wavelength range of the room temperature radiative spectrum, 5.5 µm–50 µm. The single value is obtained by weighting the measured emittance with the room temperature spectral density of heat flow rate [70].

Page 57: A new problem was added:

2.6 In problem 1.12 in Chapter 1 we calculated the heat flow rate due to perspiration and respiration of an adult human being. Now calculate the net heat flow rate of radiation, and the heat flow rates of convection and conduction of a clothed inactive person in a conditioned room. For radiation and convection assume that a person is a vertical convex surface of emittance 0.97 and surface area 1.8 m^2, so that surface coefficients typical for building physics can be used. Take the surface temperature of a clothed person to be 28 °C and the temperature of the room to be 20 °C. For conduction assume that the surface area of the shoe sole is 5 dm^2 and that the thermal resistance of the shoe sole is 0.25 m^2 K/W. Take the temperature of the foot sole skin to be 33 °C and the temperature of the floor to be 20 °C. (83 W, 36 W, 2.6 W)

3 Heat transfer in building components

Page 72: The text of the Section 3.1.5 is replaced with the text:

So far, we have neglected the effect of solar radiation, whose density of heat flow rate in zenith position at the Earth's surface is approximately $1000\,\mathrm{W/m^2}$. The actual solar density of heat flow rate q_{sol} (which includes both short- and long-wavelength radiation) varies depending on the geographic latitude, the period of year and the period of day.

In Section 2.4.1 on page 44, we have already shown that the transparent building elements transmit solar radiation to the internal environment with a density of heat flow rate of $\tau_{\mathrm{e}} q_{\mathrm{sol}}$ (Fig. 3.12, top). On the other hand, the radiation absorbed by the transparent and nontransparent building element with a density of heat flow rate of $\alpha_{\mathrm{e}} q_{\mathrm{sol}}$, increases building element temperature.

For a nontransparent building element (Fig. 3.12, left) and for a single glazed building element (Fig. 3.12, middle), the solar direct transmittance τ_{e} and the solar direct absorptance α_{e} simply take into account the transmittance and absorptance of the element in the most important wavelength range of the solar radiative spectrum, $0.3\,\mu\mathrm{m}$–$2.5\,\mu\mathrm{m}$ [69, 72]. However, in the case of a double glazed building element (Fig. 3.12, right), the solar direct transmittance and absorptance must take into account the radiative properties of both panes. In particular, the solar direct transmittance τ_{e} takes into account not only the radiation that is directly transmitted by both panes, but also the radiation that is transmitted by the outer pane, then (multiple times) reflected between two panes and finally transmitted by the inner pane. Similarly, the solar direct absorptance of the outer pane α_{e1} takes into account not only the radiation that is directly absorbed by the outer pane, but also for the radiation that is transmitted by the outer pane, then (multiple times) reflected between two panes and finally absorbed by the outer pane, while the solar direct absorptance of the inner pane α_{e2} takes into account not only the radiation that is transmitted by the outer pane and then directly absorbed by the inner pane, but also the radiation that is transmitted by the outer pane, then (multiple times)

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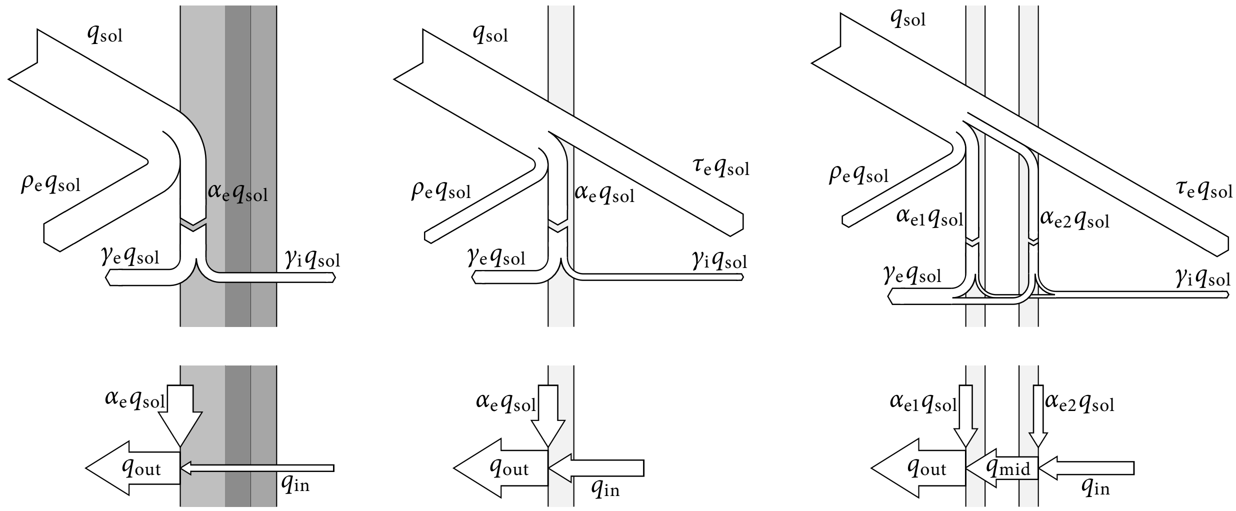

Figure 3.12: Solar gains for a nontransparent building element (left), a transparent or single glazed building element (middle) and a double glazed building element (right). As shown in the sketches above, a part of the absorbed solar radiation ($\alpha_e q_{sol}$) and all of the transmitted solar radiation ($\tau_e q_{sol}$) provide additional densities of heat flow rate towards the internal environment. As shown in the sketches below, the absorbed radiation changes the densities of the heat flow rate from the internal to the external environment.

reflected between two panes and finally absorbed by the inner pane. All solar direct transmittances and absorptances are weighted by the solar spectral density of the heat flow rate [69, 72].

In the winter period the higher temperature of the building element (Fig. 3.12, bottom)

- increases the temperature difference and consequently the density of the heat flow rate from the building element to the external environment q_{out} and
- decreases the temperature difference and consequently the density of the heat flow rate from internal environment to the building element q_{in}.

This can be perceived as an additional density of the heat flow rate towards the external environment $\gamma_e q_{sol}$ and an additional density of the heat flow rate towards the internal environment $\gamma_i q_{sol}$, both resulting from the absorbed energy (Fig. 3.12, top). Heat transfer due to solar radiation thus has two components, and both should be taken into account when studying solar gain.

Nontransparent building element

In the case of the nontransparent building element, the density of the heat flow rate of solar gain has only one component $\gamma_i q_{sol}$ (Fig. 3.12, left). We assume that the entire absorption process takes place only on the external surface, which means that the density of the heat flow rate from the internal environment to the building element is equal to the density of the heat flow rate through the building element. We can therefore write

$$q_{\text{in}} = \frac{\theta_{\text{i}} - \theta_{\text{se}}}{\sum_i R_i + R_{\text{si}}}, \tag{3.14}$$

$$q_{\text{out}} = \frac{\theta_{\text{se}} - \theta_{\text{e}}}{R_{\text{se}}}. \tag{3.15}$$

Note that q_{in} represents the actual heat losses. By writing the relationship between three densities of heat flow rate meeting at the external surface and using (3.15), we can also express it as

$$q_{\text{in}} = q_{\text{out}} - \alpha_{\text{e}} q_{\text{sol}} = \frac{\theta_{\text{se}} - \theta_{\text{e}} - R_{\text{se}}\alpha_{\text{e}}q_{\text{sol}}}{R_{\text{se}}}. \tag{3.16}$$

If we put (3.14) and (3.16) together, we finally get

$$q_{\text{in}} = \frac{\theta_{\text{i}} - \theta_{\text{e}} - R_{\text{se}}\alpha_{\text{e}}q_{\text{sol}}}{R_{\text{se}} + \sum_i R_i + R_{\text{si}}} = U(\theta_{\text{i}} - \theta_{\text{e}} - R_{\text{se}}\alpha_{\text{e}}q_{\text{sol}}).$$

We can rewrite this expression in a similar form to expression (3.5)

$$q = U(\theta_{\text{i}} - \theta_{\text{sol-air}}), \tag{3.17}$$

if we replace the external temperature with the equivalent, *sol-air temperature* $\theta_{\text{sol-air}}$

$$\theta_{\text{sol-air}} = \theta_{\text{e}} + \alpha_{\text{e}}\, R_{\text{se}}\, q_{\text{sol}}. \tag{3.18}$$

The sol-air temperature can also contain additional infrared radiation due to the difference of the external air temperature θ_{e} and apparent sky temperature θ_{sky}

$$\Delta q_{\text{sky}} = F_{\text{sky}}\, h_{\text{re}}\, (\theta_{\text{e}} - \theta_{\text{sky}}),$$

where F_{sky} is the view factor between the building component and the sky, for example, 1.0 for horizontal and 0.5 for vertical building components. If we add Δq_{sky} in the same way as we added $\alpha_{\text{e}}\, q_{\text{sol}}$ before, we get for the sol-air temperature

$$\theta_{\text{sol-air}} = \theta_{\text{e}} + \alpha_{\text{e}}\, R_{\text{se}}\, q_{\text{sol}} - R_{\text{se}}\, \Delta q_{\text{sky}}.$$

Transparent or single glazed building element

In the case of the transparent or single glazed building element, the density of the heat flow rate due to solar gain has two components, $\tau_{\text{e}} q_{\text{sol}}$ and $\gamma_{\text{i}} q_{\text{sol}}$ (Fig. 3.12, middle). We assume that the whole absorption process takes place only on the external surface, which means that the density of the heat flow rate from the internal environment to the building element is equal to the density of the heat flow rate through the building element. We can therefore write

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Solar gain in transparent building elements has two components: The first is the heat transferred into the building and the second is the heat that warms the building element and reduces the heat transferred into the environment.

$$q_{\text{in}} = \frac{\theta_{\text{i}} - \theta_{\text{se}}}{R + R_{\text{si}}}, \tag{3.19}$$

$$q_{\text{out}} = \frac{\theta_{\text{se}} - \theta_{\text{e}}}{R_{\text{se}}}. \tag{3.20}$$

Note that q_{in} represents the actual heat losses. By writing the relationship between three densities of heat flow rate meeting at the external surface and using (3.20), we can also express it as

$$q_{\text{in}} = q_{\text{out}} - \alpha_{\text{e}} q_{\text{sol}} = \frac{\theta_{\text{se}} - \theta_{\text{e}} - R_{\text{se}}\alpha_{\text{e}} q_{\text{sol}}}{R_{\text{se}}}. \tag{3.21}$$

If we put (3.19) and (3.21) together, we finally get

$$q_{\text{in}} = \frac{\theta_{\text{i}} - \theta_{\text{e}} - R_{\text{se}}\alpha_{\text{e}} q_{\text{sol}}}{R_{\text{se}} + R + R_{\text{si}}},$$

which can be written as the sum of a density of heat flow rate from the internal to the external environment without the presence of solar gain q and an additional solar density of heat flow rate towards internal environment due to the solar gain $\gamma_{\text{i}} q_{\text{sol}}$

$$\begin{aligned} q_{\text{in}} &= q - \gamma_{\text{i}} q_{\text{sol}}, \\ q &= \frac{\theta_{\text{i}} - \theta_{\text{e}}}{R_{\text{se}} + R + R_{\text{si}}} = U(\theta_{\text{i}} - \theta_{\text{e}}), \\ \gamma_{\text{i}} &= \frac{R_{\text{se}}\alpha_{\text{e}}}{R_{\text{se}} + R + R_{\text{si}}} = U\, R_{\text{se}}\, \alpha_{\text{e}}. \end{aligned} \tag{3.22}$$

We define the *solar factor* g as the ratio of the additional densities of the heat flow rate to the internal environment due to solar radiation q_{add} to the density of heat flow rate of the incident solar radiation

$$\begin{aligned} g &= \frac{q_{\text{add}}}{q_{\text{sol}}} = \frac{\tau_{\text{e}} q_{\text{sol}} + \gamma_{\text{i}} q_{\text{sol}}}{q_{\text{sol}}} = \tau_{\text{e}} + \gamma_{\text{i}} \\ \Longrightarrow\quad g &= \tau_{\text{e}} + U\, \alpha_{\text{e}}\, R_{\text{se}}. \end{aligned} \tag{3.23}$$

Note that the first part of the solar factor corresponds to the solar radiation that is transmitted through the building component and the second part corresponds to the solar radiation that is absorbed by the building component and then released to the internal environment. The additional contribution to the density of heat flow rate towards the internal environment due to solar radiation is then

$$q_{\text{add}} = g\, q_{\text{sol}}. \tag{3.24}$$

In the United States, the quantity Solar Heat Gain Coefficient (SHGC) is used instead of the solar factor. The solar gain of glazing is an important issue for the *passive house* concept, which is discussed in Section 3.4.

Standards EN 410 and ISO 9050 [69, 72] provide for the calculation of the solar gain of glazing. The standard prescribes special calculations of τ_{e}, α_{e} (0.38 µm–2.5 µm), as well as a calculation of $h_{\text{e}} = 1/R_{\text{se}}$ and $h_{\text{i}} = 1/R_{\text{si}}$ (5.5 µm–50 µm), whereby the thermal resistance of the glazing is neglected $R = 0\,\text{m}^2\,\text{K/W}$ in (3.22). This leads to the final expression

$$\gamma_{\text{i}} = \frac{h_{\text{i}}\alpha_{\text{e}}}{h_{\text{e}} + h_{\text{i}}}.$$

Double glazed building element

The case of the double glazed building element is somewhat more complex because both panes absorb part of the solar radiation (Fig. 3.12, right). We assume that the entire absorption process takes place only on the external surface of the outer pane and on the internal surface of the inner pane. It can be shown (see Problem 3.2) that the temperature of each pane is practically constant, so this assumption has only a negligible effect on the result. Because energy is absorbed on two planes, we must consider three heat transfers: density of heat flow rate from the building element to the external environment q_{out}, density of heat flow rate through the building element q_{mid} and density of heat flow rate from the internal environment to the building element q_{in}. They can be expressed as

$$q_{\mathrm{in}} = \frac{\theta_{\mathrm{i}} - \theta_{\mathrm{si}}}{R_{\mathrm{si}}}, \tag{3.25}$$

$$q_{\mathrm{mid}} = \frac{\theta_{\mathrm{si}} - \theta_{\mathrm{se}}}{R + R_{\mathrm{g}} + R}, \tag{3.26}$$

$$q_{\mathrm{out}} = \frac{\theta_{\mathrm{se}} - \theta_{\mathrm{e}}}{R_{\mathrm{se}}}. \tag{3.27}$$

Note that q_{in} represents the actual heat losses. If we write the relationship between three densities of heat flow rate meeting at the internal surface and use (3.26), we can also express it as

$$q_{\mathrm{in}} = q_{\mathrm{mid}} - \alpha_{\mathrm{e2}} q_{\mathrm{sol}} = \frac{\theta_{\mathrm{si}} - \theta_{\mathrm{se}} - (R + R_{\mathrm{g}} + R)\alpha_{\mathrm{e2}} q_{\mathrm{sol}}}{R + R_{\mathrm{g}} + R}. \tag{3.28}$$

Finally, if we write the relationship between three densities of heat flow rate meeting at the external surface and use (3.27), we can also express it as

$$q_{\mathrm{in}} = q_{\mathrm{out}} - (\alpha_{\mathrm{e1}} + \alpha_{\mathrm{e2}}) q_{\mathrm{sol}} = \frac{\theta_{\mathrm{se}} - \theta_{\mathrm{e}} - R_{\mathrm{se}}(\alpha_{\mathrm{e1}} + \alpha_{\mathrm{e2}}) q_{\mathrm{sol}}}{R_{\mathrm{se}}}. \tag{3.29}$$

If we put (3.25), (3.28) and (3.29) together, we finally get

$$q_{\mathrm{in}} = \frac{\theta_{\mathrm{i}} - \theta_{\mathrm{e}} - R_{\mathrm{se}}(\alpha_{\mathrm{e1}} + \alpha_{\mathrm{e2}}) q_{\mathrm{sol}} - (R + R_{\mathrm{g}} + R)\alpha_{\mathrm{e2}} q_{\mathrm{sol}}}{R_{\mathrm{se}} + R + R_{\mathrm{g}} + R + R_{\mathrm{si}}}, \tag{3.30}$$

which can be written as the sum of a density of heat flow rate from the internal to the external environment without the presence of solar gain q and an additional solar density of heat flow rate towards the internal environment due to the solar gain $\gamma_{\mathrm{i}} q_{\mathrm{solar}}$

$$\begin{aligned} q_{\mathrm{in}} &= q - \gamma_{\mathrm{i}} q_{\mathrm{sol}}, \\ q &= \frac{\theta_{\mathrm{i}} - \theta_{\mathrm{e}}}{R_{\mathrm{se}} + R + R_{\mathrm{g}} + R + R_{\mathrm{si}}} = U(\theta_{\mathrm{i}} - \theta_{\mathrm{e}}), \\ \gamma_{\mathrm{i}} &= \frac{R_{\mathrm{se}}(\alpha_{\mathrm{e1}} + \alpha_{\mathrm{e2}}) + (R + R_{\mathrm{g}} + R)\alpha_{\mathrm{e2}}}{R_{\mathrm{se}} + R + R_{\mathrm{g}} + R + R_{\mathrm{si}}}. \end{aligned} \tag{3.31}$$

For the solar factor of a double glazed building element we finally get

$$g = \tau_e + U\left[(\alpha_{e1} + \alpha_{e2})R_{se} + \alpha_{e2}(R + R_g + R)\right]. \tag{3.32}$$

The standards EN 410 and ISO 9050 [69, 72] prescribe a special calculation of τ_e, α_{e2}, α_{e1} (0.3 μm–2.5 μm), as well as $h_e = 1/R_{se}$ and $h_i = 1/R_{si}$ (5.5 μm–50 μm). The thermal conductance between the external and internal surfaces of the glazing $\Lambda = 1/(R + R_g + R)$ is calculated according to EN 673 [12] (5.5 μm–50 μm). This leads to the final expression

$$\gamma_i = \frac{\frac{\alpha_{e1}+\alpha_{e2}}{h_e} + \frac{\alpha_{e2}}{\Lambda}}{\frac{1}{h_e} + \frac{1}{\Lambda} + \frac{1}{h_i}}.$$

Page 88: At the end of Section 3.3.1 the following paragraph is added:

The standard ISO 6946 [21] also specifies the calculation of heat losses through pitched roofs:

1. Roof structure consisting of a flat, insulated ceiling and a pitched roof, with naturally ventilated roof spaces. The roof space can be considered as a thermally homogeneous layer with a thermal resistance that depends on the properties of the roof.
2. Heated space under an insulated pitched roof.
    - The values for horizontal heat flow apply to heat flow directions ±30° from the horizontal plane.
    - The values for convective surface coefficients can be obtained by linear interpolation between horizontal and vertical values. Note that radiative surface coefficient is independent of the slope.

Page 91: After Equation (3.28), new section is added:

Note here that the air flow rate is closely related to the *air change rate* n (1/s or 1/h), which is the quotient of the air flow rate by the total volume of building V

$$n = \frac{\dot{V}}{V}, \tag{3.29}$$

and which essentially indicates how many times is the air within the building completely exchanged within the given period.

Page 91: The last sentence at the end of Section 3.3.3 is replaced with the text:

Apart from obvious reasons, such as mechanical air exhausts (range hood), ducts (chimneys), vents and airing (natural air exchange through window openings), there are always ventilation losses through leaks in the building envelope. This contribution is usually described either by the building *airtightness*, that is, the

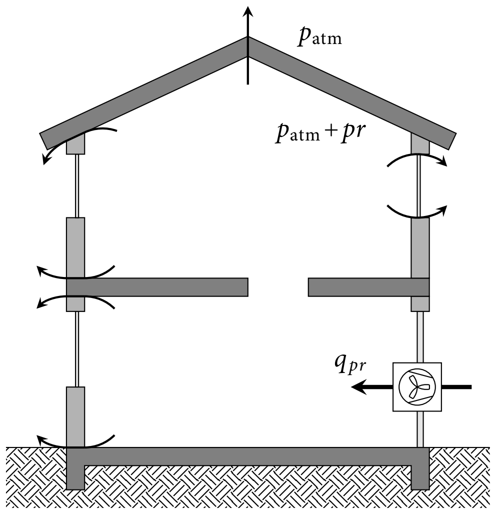

Figure 3.13: Setup for the blower door measurement of the whole building. Here, blower door generates a pressure difference pr by a controlled air flow rate in the building q_{pr}. The latter is compensated by air leakage through leakage points or areas in the building envelope.

resistance of the building to air leakage through *unintentional* leakage points or areas in the building envelope, or by the opposite building *air permeability*. Typically, leakage occurs at junctions between walls, floors, roof, window frames and door frames, and through electrical and other installations. Leaks in the building envelope contribute significantly to the total heat losses, must be rigorously controlled and are given special consideration in the passive house concept (Section 3.4).

Quantitative methods were developed to calculate or measure the air permeability of buildings. The calculation of the air flow rate $\dot{V}$ is described for example in EN 16798-7 [71] and ISO 52019-2 [74].

On the other hand, the air permeability of buildings can be measured by the fan pressurization method defined by the standard ISO 9972 [73]. This method is usually performed by a device called a blower door, which is an assembly mounted on the door and contains a fan or blower that creates a controlled air flow rate in or out of the building, as shown in Fig. 3.13. Due to the air flow rate through the fan, the air pressure inside the building rises above or below the air pressure outside. At a constant pressure difference, the air flow rate through the blower door is equal to the air flow rate through the building envelope, which is called the *air leakage rate at the reference pressure difference* q_{pr} ($\mathrm{m^3/s}$ or $\mathrm{m^3/h}$). Similarly to air flow rate, the air leakage rate is the quotient of the volume of air transferred through the building envelope V_{a} by time in which the air is transferred

$$q_{pr} = \frac{V_{\mathrm{a}}}{t}. \tag{3.30}$$

The typical pressure difference used is 50 Pa and the corresponding *air leakage rate at* 50 Pa is referred to as q_{50}.

The air leakage rate is closely related to the *air change rate at the reference pressure difference* n_{pr} (1/s or 1/h), which is the quotient of the air leakage rate at the reference pressure difference by the total volume of the building V

$$n_{pr} = \frac{q_{pr}}{V}. \tag{3.31}$$

For a typical pressure difference of 50 Pa the corresponding *air change rate at* 50 Pa is referred to as n_{50}. Alternatively, the air changes per hour (ACH) is often used as a term for this quantity. A less commonly used quantities are the specific leakage rates as the quotients of the air flow rate by the total envelope area or the air flow rate by the total floor area.

Note that units for air flow rate, air leakage rate and air change rate are usually given *per hour*, although basic SI units are given *per second*. In this case the specific heat capacity c_p in (3.27) must be given in W h/(kg K). Upper limits for the air change rate at 50 Pa are usually prescribed by standards [68] or national legislation.

Measurements of the air permeability of buildings can also be used for a more reliable calculation of air flow rate q_V. In its simplest form, the air change rate at 50 Pa n_{50} and the air change rate (in the absence of pressure difference) n can be related by an empirical equation

$$n = \frac{n_{50}}{N}, \tag{3.32}$$

where N is the leakage-infiltration ratio. Similarly, the air flow rate q_V can be determined from the air leakage rate at 50 Pa q_{50}

$$\dot{V} = \frac{q_{50}}{N}.$$

The comprehensive model for North America [76] provided the leakage-infiltration ratio for various climates and also took into account the height, shielding and leakage of the building. The sum of the leakage-infiltration ratio and all corrections gives a range of values between 6 and 44. On the other hand, the standard ISO 13789 [35] prescribes a value $N = 20$.

Currently, in Northern America building air permeability is measured according to the standard ASTM E779 [67], with a conventional reference pressure of 4 Pa. ASHRAE 62.2 [66] then prescribes a more accurate determination of air flow rate, taking into account the vertical distance between the lowest and highest above-grade points within the pressure boundary and the weather and shielding factor, which is specified for more than a thousand US and Canadian locations.

Page 93: The text of Section 3.4 is replaced with a text:

Passive house is a rigorous, voluntary standard for energy-efficient building:

> A passive house is a building, for which thermal comfort (ISO 7730) can be achieved solely by post-heating or post-cooling of the fresh air mass, which is required to achieve sufficient indoor air quality conditions – without the need for additional recirculation of air. [59]

In a passive house, the conventional means of heating and cooling (for example, furnaces, central heating or air conditioners), which heat or cool the internal air, are not allowed. Regardless of that, a ventilation of minimum air flow rate $\dot{V} = 30\,\mathrm{m^3/h} = 8.3 \times 10^{-3}\,\mathrm{m^3/s}$ per person is required in order to maintain a reasonable indoor air quality. The standard allows for fresh air to be heated or cooled, so the building is essentially conditioned by ventilation only.

To achieve thermal comfort, several conditions must be met, which can be roughly divided into four categories.

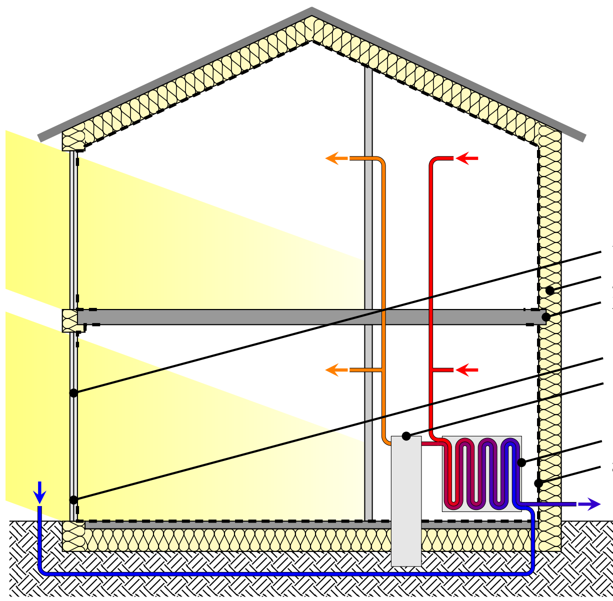

Figure 3.29: The passive house design. To achieve low energy requirements for the building energy balance, several conditions must be met. These conditions can be roughly divided into four categories: small heat losses, efficient heating, efficient ventilation and shading (not shown).

1. *Small heat losses.*

   Direct heat losses, heat losses through the ground and heat losses through unconditioned spaces are sufficiently reduced under these conditions (Fig. 3.29):

   - A low thermal transmittance of all nontransparent building components should be achieved using good thermal insulation.
   - Triple-glazed windows of thermal transmittance $U_\mathrm{W} \leq 0.8\,\mathrm{W/(m^2\,K)}$ [60] should be used.
   - Thermal bridges should be avoided or their effects minimised.

2. *Efficient ventilation.*

   In order to reduce heat losses through ventilation, we need to address two issues (Fig. 3.29):

- The air flow rate should be reduced by increasing the *airtightness* of the building (Section 3.3.3). This is ensured by the passive house requirement that the air change rate at 50 Pa should be less than 0.6/h [60], which corresponds approximately to a natural air change rate of 0.03/h (3.32).
- Regardless of this, as already mentioned, ventilation of minimum air flow rate $\dot{V} = 30\,\mathrm{m^3/h} = 8.3 \times 10^{-3}\,\mathrm{m^3/s}$ per person is necessary to maintain adequate indoor air quality, and corresponding heat losses are unavoidable. If the dwelling has $30\,\mathrm{m^2}$ of living space per person with a ceiling height of 2.5 m, this results in an air change rate of 0.4/h, which is well above the permissible natural air change rate. To reduce ventilation losses, passive houses use forced ventilation through the *recuperator* (Section 2.3.1 on page 38).

3. *Efficient warming*.

   It is impossible to avoid heat losses completely, therefore a certain amount of external energy is necessary to maintain thermal comfort (Fig. 3.29):

   - The standard permits post-heating of fresh air as a means of heating. To increase post-heating efficiency, a *heat pump* must be used.
   - In many climates, heating through ventilation will not be sufficient, so that part of the heating will be provided by *solar gain* (Section 3.1.5 on page 72), in particular energy harvested by exploiting the *greenhouse effect* (Section 2.4.1 on page 44). This means that passive houses must have *large, equator-oriented transparent building components* (glass surfaces) of solar factor $g \geq 0.5$ [60].

4. *Shading*.

   On the other hand, large solar gain can be very disadvantageous in warmer parts of the world and for warmer seasons, when the interior of the building needs to be cooled rather than heated. In these cases, shading systems with a low solar factor must be used.

With the preceding assumptions, it is possible to calculate the upper limit of energy consumption independently of climatic conditions. The fresh air optionally flows first through a ground-coupled heat exchanger (underground pipes) and then through the recuperator to increase its temperature almost to the internal temperature. The air is then additionally heated for a maximum of $\Delta\theta = 30\,°\mathrm{C}$ to avoid pyrolysis of dust, which starts at about 50 °C. Using minimum air flow rate of $q_V = 8.3 \times 10^{-3}\,\mathrm{m^3/s}$ and (3.27), we see that the maximum permissible heat consumption is $\Phi_\mathrm{V} = 300\,\mathrm{W}$ per person. The general assumption is that the dwelling has $30\,\mathrm{m^2}$ of living space per person, which gives a more practical maximum permissible heat consumption of $10\,\mathrm{W/m^2}$. Furthermore, for a continental climate, the annual heating energy should not exceed $15\,\mathrm{kW\,h/m^2}$ of the net living space (treated floor area) [60].

4 Moisture in building components

Page 118: A new figure, text and a supplementary multimedia content were added:

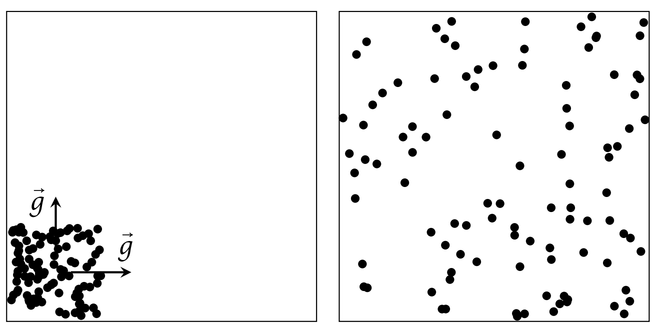

Figure 4.29: The model for the diffusion of black substance particles within a white (invisible) fluid. Initially, all the black substance is located in a small fraction of the container and its concentration is high there, whereas elsewhere the concentration is zero (left). A net movement of black particles from the high concentration zone to the low concentration zone. After longer period of time the concentration of the black substance balances out within the whole container (right).

Diffusion in general is the net movement of anything from a region of higher concentration to a region of lower concentration, as shown in Fig. 4.29.

Page 146: New problem is added:

4.13 In a room of rectangular floor plan with dimensions 4.0 m × 5.0 m and height 2.5 m, there is dry air of density 1.20 kg/m^3 at temperature 21 °C. In the room we place a pot with 3.0 L of liquid water at temperature 41 °C. What is the mass of evaporated water at the moment when temperatures of air and liquid water in the room drop to 16 °C? What is the relative humidity then? The specific heat capacity of dry air is 1.0 kJ/(kg K), the specific heat capacity of liquid water

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is 4.2 kJ/(kg K) and the specific heat of evaporation of water at 16 °C is 2460 kJ/kg. The molar mass of water is 18 g/mol. (0.25 kg, 37 %)

5 Basic of waves

Page 154, **Figure 5.7**: A supplementary multimedia content was added.

Page 155, **Figure 5.7**: A supplementary multimedia content was added.

Page 159, **Figure 5.12**: A supplementary multimedia content was added.

Page 160: New problem is added:

5.3 The ear canal is a tube of length 25 mm, open on one side and closed on the other side with the tympanic membrane. Calculate frequencies of all harmonics of the standing wave in the ear canal that can be heard. (3.4 kHz, 10.2 kHz, 17.0 kHz)

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6 Sound propagation

Page 170: A new text was added before the first paragraph:

Another viable advantage is a more intuitive description of the sensation. It is well known that the magnitude of human sensation is not directly proportional to the intensity of the stimulus. The exact relationship is still debatable, but one prediction, the Weber-Fechner law, states that the magnitude of human sensation is proportional to the logarithm of the intensity of the stimulus.

The disadvantage of levels is that they are essentially dimensionless, although their unit is decibel. This means that all four level quantities in this book, the sound power level, the linear sound power level, the surface sound power level and the sound pressure level, all have the same unit. As a result, we can no longer identify the quantity from the unit and must be much more careful in our calculations.

Page 171: A new text was added after the last paragraph:

We could also come to the same conclusion with a more tangible procedure. As shown in Fig. 5.1 on page 147, agitation at one point on the water surface forms circular wave fronts. As a wavefront moves away from this point, its length increases, but its crest becomes lower. This is due to the fact that the energy and power—energy divided by the period of oscillation—of a particular wavefront are constant, which means that energy and power per wavefront length should decrease.

Similarly, a point sound source in space forms spherical wavefronts. As a particular wavefront moves away from the source, its area increases, which means that the energy and power per area of a wavefront decrease. The intensity is obtained by dividing the source sound power by the wavefront area corresponding to the area of the sphere.

Page 185: The text of Section 6.4.3 is replaced with a text:

Road traffic is the single largest contributor of noise in urban areas (Table 6.3). For this reason, we will loosely describe a calculation method for its sound power level.

The European commission directive 2015/996 [75] established common noise assessment methods in the European Union, usually referred to by the acronym

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CNOSSOS-EU (**c**ommon **no**ise a**ss**essment meth**o**d**s**). With these methods the traffic noise source of roads or road lanes is represented by linear noise sources 0.05 m above the road surface. Their linear sound power levels depend on several parameters:

- *Vehicle type*. Vehicles are divided into four categories: light motor vehicles, medium heavy vehicles, heavy vehicles and powered two-wheelers, the latter being further subdivided into two subcategories.
- *Average velocity*. The average velocity is required for each vehicle category.
- *Traffic volume*. The number of vehicles per hour is required for each vehicle category.
- *Road gradient*. The road gradient affects both engine load and engine speed through the choice of gear and therefore the propulsion noise emission of the vehicle. The effect depends on the slope.
- *Traffic flow type*. Acceleration and deceleration before and after crossings with traffic lights and roundabouts affect the rolling and propulsion noise. The effect depends on distance to the nearest intersection.
- *Road surface*. Porous road surfaces generate less rolling noise and absorb more propulsion noise. Velocity dependent corrections are defined for fourteen different types of road surfaces in relation to the reference road surface.
- *Air temperature*. The rolling sound power level slightly decreases as the air temperature increases.
- *Studded tyres*. If a significant number of light vehicles use studded tyres for several months of the year, the induced velocity dependent effect on rolling noise is taken into account.

The CNOSSOS-EU methods specify the calculation of the linear sound power level (6.32) of a road with one vehicle per hour. The sound power level depends on the velocity and the vehicle category. Fig. 6.15 shows the linear sound power level for the first three vehicle categories. Note that in most practical situations (higher velocities, light motor vehicles), the linear sound power level of the road increases with velocity.

The noise emission is the result of three contributions:

Table 6.3: Estimated number of people in the European Union exposed to noise levels $L_{\mathrm{den}} \geq 55\,\mathrm{dB}$ [63].

Noise Source	Exposed People
road traffic	125 000 000
rail traffic	8 000 000
aircraft traffic	3 000 000
industry	300 000

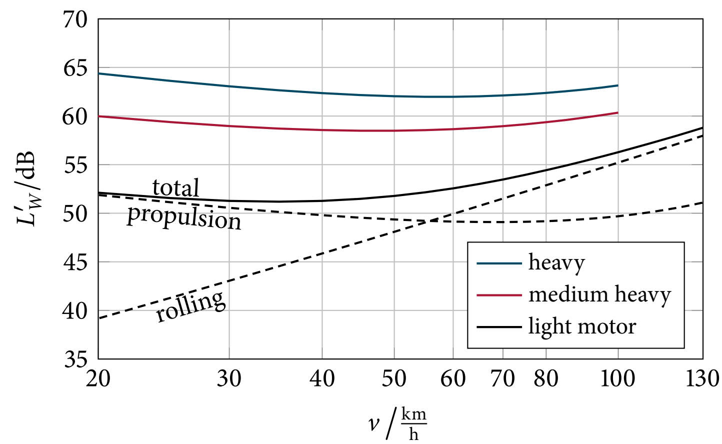


Figure 6.15: Linear sound power levels of a road with one vehicle per hour for first three vehicle categories, according to the CNOSSOS-EU methods. The contribution of the rolling and the propulsion is shown for the light motor vehicles.

1. *Propulsion noise* (engine and exhaust system) is the dominant contribution to the sound power at lower velocities; however, this contribution is constantly being reduced by new car designs.
2. *Rolling noise* (interaction between road surfaces and rubber tires) is the dominant contribution at higher velocities. This contribution can be influenced by both tire design and surface pavement.
3. *Aerodynamic noise* increases with vehicle velocity. Note that in CNOSSOS-EU aerodynamic noise is incorporated in the rolling noise source (Fig. 6.15).

The traffic volumes for all vehicle (sub)categories must be determined by automatic traffic counters or traffic demand models. If n_1, n_2, n_3, n_{4a} and n_{4b} are the number of vehicles per hour for all vehicle (sub)categories, whereas L'_{W1}, L'_{W2}, L'_{W3}, L'_{W4a} and L'_{W4b} are the linear sound power levels of one vehicle per hour for all vehicle (sub)categories, the total linear sound power level is (6.37):

$$L'_W = 10\lg\left(n_1\,10^{0.1L'_{W1}} + n_2\,10^{0.1L'_{W2}} + n_3\,10^{0.1L'_{W3}} + n_{4a}\,10^{0.1L'_{W4a}} + n_{4b}\,10^{0.1L'_{W4b}}\right).$$

The CNOSSOS-EU methods [75] also established assessment methods for railway noise, industrial noise, aircraft noise and noise propagation. For example, the railway traffic noise source is represented by two linear noise sources 0.5 m and 4.0 m above the tracks.

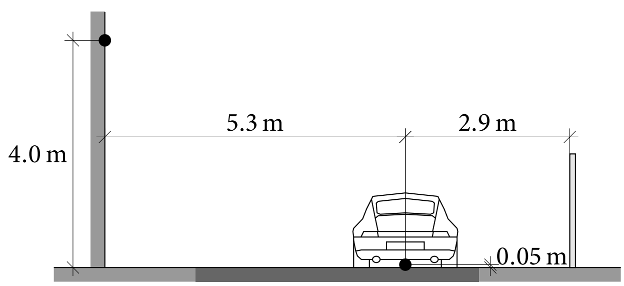
Page 190: New problem is added:

6.7 The sound pressure level gain due to physiological effects, such as the shape of the head and the outer ear, is 20 dB [65]. For the quietest sound still detectable, calculate the sound power entering the human body through the tympanic membrane, if the effective area of the tympanic membrane is 43 mm^2. (4.3×10^{-15} W)

7 Building acoustics

Page 213: New problem is added:

7.9 The horizontal distance between the right traffic lane and the building is 5.3 m and between the right traffic lane and the transparent noise barrier is 2.9 m, as shown in the figure. Calculate the sound pressure level on the façade of the building 4.0 m above the ground, if absorbance of the barrier is 0.20. Assume for the traffic lane that the sound source is 5.0 cm above the ground and take the linear sound power level to be 70 dB. Take into account the direct sound and the sound reflected by the barrier. (55.4 dB)



7.10 A living room of rectangular foor plan of dimensions $l_2 \times w_2 = 5.0\,\mathrm{m} \times 3.0\,\mathrm{m}$ is located directly adjacent to the engine room of rectangular foor plan of dimensions $l_1 \times w_1 = 2.0\,\mathrm{m} \times 2.0\,\mathrm{m}$, as shown in the figure. Both rooms are of height 2.6 m. The sound reduction index between the rooms is 57 dB, whereas the reverberation time of the living room is 0.50 s. What is the highest acceptable sound pressure level in the engine room, if the highest permissible sound pressure level in the room is 35 dB? There is a device of sound power 100 dB in the engine room. In order not to exceed the highest acceptable sound pressure level, sound absorbers are installed on all room surfaces. What is their smallest absorbance? Neglect the increase of the sound reduction index due to the installation of absorbers. (95.9 dB; 0.36)

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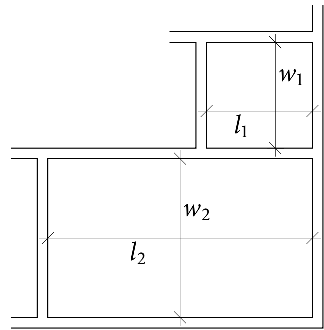
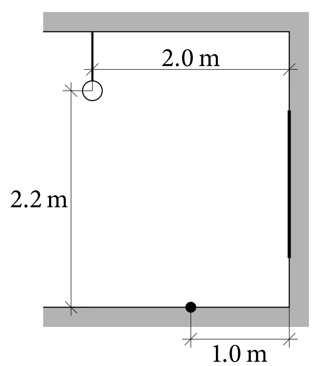

w_1
l_1
w_2
l_2

8 Illumination

Page 216, **Figure 8.1**: A supplementary multimedia content was added.

Page 242: New problem is added:

8.4 Isotropic light of luminous flux 1500 lm is located 2.2 m above the floor and 2.0 m from the vertical wall with the mirror, as shown in the figure. Calculate the illumination of the floor 1.0 m from the wall. Take into account only the direct light and the light reflected by the mirror. (23.7 lx)



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DOI 10.1007/978-3-319-57484-4

A Tables

Page 243–246: New symbols/names were added:

Symbol	Unit	Name
n	1/s	air change rate
n_{pr}	1/s	air change rate at the reference pressure difference
q_{pr}	m^3/s	air leakage rate at the reference pressure difference
γ	1	ratio of heat capacities
η	1	efficiency

M. Pinterić, *Building Physics*,
DOI 10.1007/978-3-319-57484-4

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