6 Thermal analysis of building-integrated solar components

The substitution of classical building materials by 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. By using ventilated double facades with photovoltaics as an example, a methodology can be developed for thermal characterisation which 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 photovoltaics) 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_o. 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solar components, in particular warm facades, the assumption of a solar element surrounded by outside air is no longer applicable.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), partly transparent photovoltaic modules are, for example, often only separated by further glazings from the room (with room air temperature T_i) for architectural reasons.

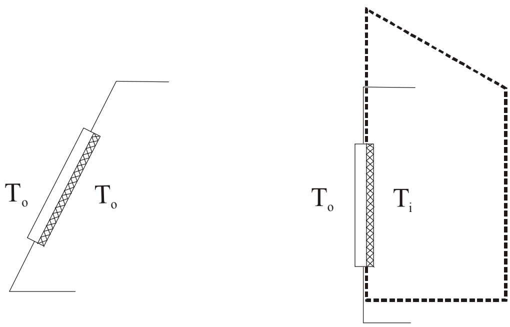


Figure 6.1: Free-standing and building-integrated collectors.

Due to this thermal coupling, heat gains for the space occur which in winter contribute to heating energy cover, but which in summer can cause overheating problems. With photovoltaic modules in a 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 pre-heating outside air. At the same time, transmission heat losses of the room can be recovered by the heated gap air.

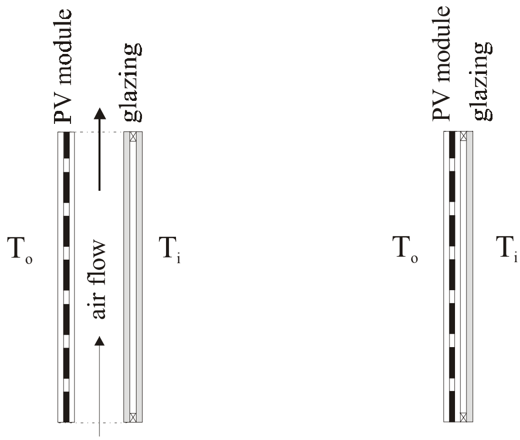


Figure 6.2: PV module in a back-ventilated cavity facade with double glazing on the room side and a PV module integrated in an double glazing construction.

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. Subsequently, a more detailed thermal model of a back-ventilated photovoltaic facade is developed which describes heat flows and temperature conditions in conventional double facades. With this model, the monthly thermal energy gains of a PV cavity facade can be calculated.

6.1 Empirical thermal model of building-integrated photovoltaics

Solar modules are generally characterised by high optical absorption coefficients in the wavelength area of short-wave solar radiation. In current-generating photovoltaic modules, however, only a small part of the absorbed irradiance, about 10–15%, is converted into electricity, with mainly heat being produced. This heat can be used for heating and actively cooling buildings, 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 transition of the module rear are 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 (see Table 6.1):

Table 6.1: Installation situations of building-integrated solar elements.

No.	*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

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.

$$\left(T_{Module} - T_o\right)\Big|_{1000W/m^2} = \frac{\Delta T}{\Delta G} \times 1000 \frac{W}{m^2} \tag{6.1}$$

The gradient obtained as an average of the test reference years varies from a minimum of 0.019 K/(W/m^2) for a free-standing module, up to 0.052 K/(W/m^2) for a facade without back-ventilation, so at 1000 W/m^2 irradiance, module temperatures of 19–52 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 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 free-standing module. In the most unfavourable version – the non back-ventilated facade – 7.5–10% less electricity is produced annually than by the free-standing module, due to temperature effects alone. The electrical energy losses are calculated relative to the annually produced energy of a free-standing module.

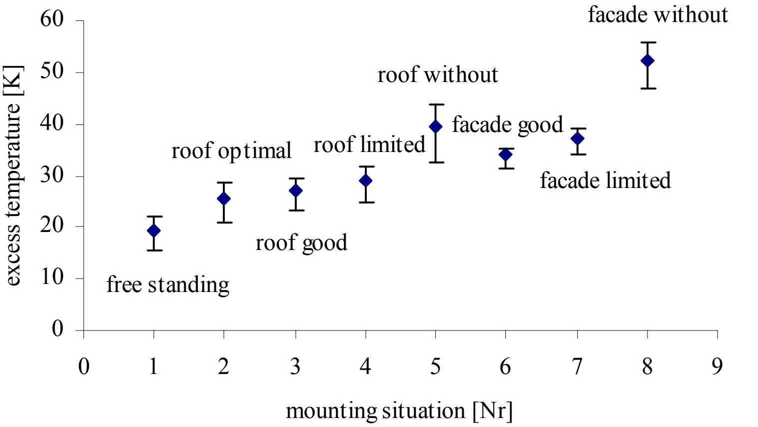


Figure 6.3: Temperature rise of a PV module in different building integration solutions with optimal, good and limited ventilation or without ventilation.

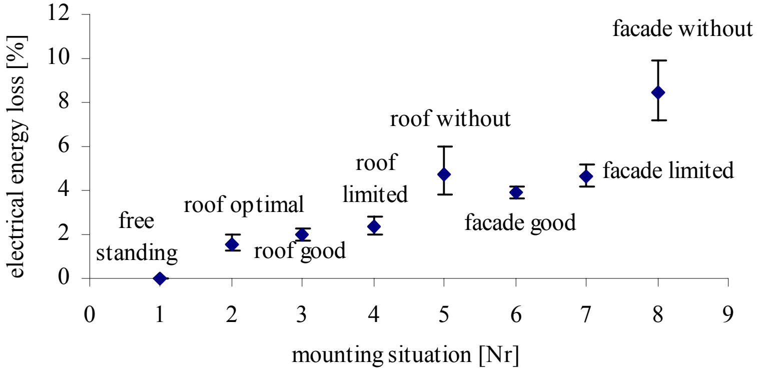


Figure 6.4: Yearly electrical energy loss of a PV module in different building integration solutions.

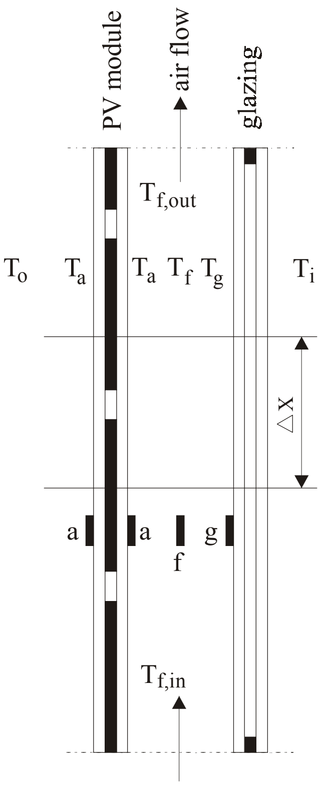
6.2 Energy balance and stationary thermal model of ventilated double facades

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 has to be created to analyse the effective heat transfer and thermal use of the module waste heat. This model must permit different temperatures as a boundary condition of the integrated solar component (ambient temperature and room temperature), and take 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 (Vollmer, 1999).

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 via free convection or fan-operated. In cavity facades, the back-ventilation gap has dimensions of between about 0.1–1 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-throughflowed 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 4 mm + 6 mm glass, 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]
h_r: radiant heat transfer coefficient [W/m^{2}K]
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]
U_f: U-value between the PV module (node *a*) and environment [W/m^{2}K]
c: heat capacity of air [J/kgK]
ρ: density of air [kg/m³]
$\dot{V}$: flow rate [m^3/s]
L: gap height [m]

Figure 6.5: 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 $\dot{Q}_{el}$ is deducted in the balance from the absorbed irradiance.

$$G\alpha_{PV} - U_f\left(T_a - T_o\right) - h_{ca}\left(T_a - T_f\right) - h_r\left(T_a - T_b\right) - \dot{Q}_{el} = 0 \qquad (6.2)$$

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.

$$c\rho\dot{V}\frac{dT_f}{dx} = h_{ca}b\left(T_a - T_f\right) + h_{cb}b\left(T_b - T_f\right) \qquad (6.3)$$

For the gap width b, the unit length $b = 1$ m is used here.

Node *b*:
The solar irradiance transmitted by the PV module, with the transmission coefficient τ_{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.

$$G\left(\tau_{PV}\alpha_b\right) - h_r\left(T_b - T_a\right) - h_{cb}\left(T_b - T_f\right) - U_b\left(T_b - T_i\right) = 0 \qquad (6.4)$$

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 λ_{PV} [W/mK], and from the outside heat transfer coefficient h_a [W/m^{2}K]. The U-value of the rear U_b consists of the thermal resistance of the gap rear R_b [m^{2}K/W] (e.g. 0.3 m^{2}K/W for a non-coated double glazing), and of the internal thermal resistance between the surface and room $1/h_i$:

$$U_f = \frac{1}{\frac{s_{PV}}{\lambda_{PV}} + \frac{1}{h_a}} \qquad (6.5)$$

$$U_b = \frac{1}{R_b + \frac{1}{h_i}} \qquad (6.6)$$

Equations (6.2) and (6.4) 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 (6.3).

$$T_a = \frac{(B\alpha_{PV} + h_r \tau_{PV} \alpha_b)G + h_r U_b T_i + BU_f T_o + (Bh_{ca} + h_r h_{cb})T_f - B\dot{Q}_{el}}{AB - h_r^2} \tag{6.7}$$

$$T_b = \frac{(A\tau_{PV} \alpha_b + h_r \alpha_{PV})G + AU_b T_i + h_r U_f T_o + (Ah_{cb} + h_r h_{ca})T_f - h_r \dot{Q}_{el}}{AB - h_r^2} \tag{6.8}$$

From this results a differential equation for the fluid temperature T_f, which can be solved by separation of the variables.

$$c\rho\dot{V}\frac{dT_f}{dx} = D_1 T_i + D_2 T_o - D_3 \dot{Q}_{el} + D_4 G - D_5 T_f \tag{6.9}$$

The general solution is

$$T_f(x) = \frac{1}{D_5}\left(D_1 T_i + D_2 T_o - D_3 \dot{Q}_{el} + D_4 G - Ce^{-Zx}\right) \tag{6.10}$$

with the integration constant C dependent on the boundary condition and the constants D_1 to D_5 given by

$$D_1 = \frac{h_{ca} h_r U_b + h_{cb} A U_b}{AB - h_r^2} \qquad D_2 = \frac{h_{cb} h_r U_f + h_{ca} B U_f}{AB - h_r^2}$$

$$D_3 = \frac{h_{ca} B + h_{cb} h_r}{AB - h_r^2}$$

$$D_4 = \frac{h_{ca} h_r \tau_{PV} \alpha_b + h_{ca} B \alpha_{PV} + h_{cb} h_r \alpha_{PV} + h_{cb} A \tau_{PV} \alpha_b}{AB - h_r^2}$$

$$D_5 = h_{ca} + h_{cb} - \frac{\left(2h_{ca} h_{cb} h_r + h_{ca}^2 B + h_{cb}^2 A\right)}{AB - h_r^2}$$

with $A = U_f + h_{ca} + h_r$ $\quad B = U_b + h_{cb} + h_r$ $\quad$ and $\quad Z = \frac{D_5}{c\rho\dot{V}}$

The integration constant C results from the boundary condition at the gap inlet; here the ambient temperature T_o:

Boundary condition: with x = 0 $\quad T_f = T_o$

$$C = D_1 T_i + (D_2 - D_5)\, T_o - D_3 \dot{Q}_{el} + D_4 G \tag{6.11}$$

Thus the special solution results

$$T_f(x) = \left(1 - e^{-Zx}\right) \frac{D_1 T_i + D_2 T_o - D_3 \dot{Q}_{el} + D_4 G}{D_5} + T_o e^{-Zx} \tag{6.12}$$

The mean fluid temperature of the entire flow channel is obtained by integrating Equation (6.12) over the entire gap length *L*.

$$\begin{aligned} \overline{T}_f &= \frac{1}{L}\int_0^L T_f(x)\,dx \\ &= \frac{D_1 T_i + D_2 T_o - D_3 \dot{Q}_{el} + D_4 G}{D_5}\left(1 + \frac{1}{LZ}\left(e^{-ZL} - 1\right)\right) + \frac{T_o}{LZ}\left(1 - e^{-ZL}\right) \end{aligned} \tag{6.13}$$

With the mean fluid temperature, the mean absorber and rear temperatures of the gap can also be calculated, using Equations (6.7) and (6.8).

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.

6.2.1 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 parametres. For most applications, the assumption of a constant heat transfer coefficient is sufficiently exact. Euro standard EN 832 states that a mean thermal resistance of $1/h_i = 0.13$ m^{2}K/W 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), 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 (2002) 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 were 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.

$$\mathrm{Re} = \sqrt{\mathrm{Re}_{free}^2 + \mathrm{Re}_{for}^2} \tag{6.14}$$

The free convection proportion results from the temperature-induced density variation over the gap height *L*.

$$\mathrm{Re}_{free} = \sqrt{\frac{Gr}{2.5}} \tag{6.15}$$

$$Gr = \frac{g\beta' L^3 \left|\left(\overline{T}_f - \overline{T}_{a,b}\right)\right|}{\nu^2} \tag{6.16}$$

$$\mathrm{Re}_{for} = \frac{\mathrm{v}L}{\nu} \tag{6.17}$$

With flow conditions over an individual plate, the boundary layer flow at the lower panel edge can begin laminarly and after a certain length become turbulent, the transition point being about Re = 2×10^5 (Merker and Eiglmeier, 1999). The mean Nußelt number is formed from a laminar and a turbulent proportion.

$$Nu = \sqrt{Nu_{lam}^2 + Nu_{turb}^2} \tag{6.18}$$

For Prandtl numbers between $0.6 < \mathrm{Pr} < 10$, integration over the local Nußelt number produces the laminar proportion:

$$Nu_{lam} = 0.664\sqrt{\mathrm{Re}}\sqrt[3]{\mathrm{Pr}} \tag{6.19}$$

The turbulent proportion is calculated from an empirical correlation derived from numerical integration of the boundary layer equation:

$$Nu_{turb} = \frac{0.037\,\mathrm{Re}^{0.8}\,\mathrm{Pr}}{1 + 2.443\,\mathrm{Re}^{-0.1}\left(\mathrm{Pr}^{\frac{2}{3}} - 1\right)} \tag{6.20}$$

with $$\mathrm{Pr} = \frac{\nu c \rho}{\lambda}$$

From the mean Nußelt number, the heat transmission coefficients result as usual in $h_c = \dfrac{Nu\,\lambda}{L}$.

Example 6.1

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 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} m²/s	16.60×10^{-6}
Heat conductivity of air λ:	0.02758 W/mK	0.02554
Density ρ:	1.095 kg/m³	1.130
Heat capacity c:	1008.25 J/kgK	1007.75
heat expansion coefficient β':	0.00314 K^{-1}	0.0032
Re_{for}:	111 134	117 486
Gr:	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 come the heat transfer coefficients h_{ca} = 3.66 W/m²K, h_{cb} = 3.75 W/m²K and h_r = 3.07 W/m²K.

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 6.2

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 W/m²
Ambient temperature	T_o = 10°C
Room air temperature	T_i = 20°C
Wind velocity	v_w = 3 m/s
Flow velocity in the gap	v = 0.3 m/s
Absorption coefficient of the PV module	α_{PV} = 0.8
Transmission coefficient of the PV module	τ_{PV} = 0.1

Absorption coefficient of the back glazing	$\alpha_b = 0.05$
Layer thickness of the PV module	$s_{PV} = 0.01$ m
Heat conductivity of the PV module	$\lambda_{PV} = 0.8$ W/mK
Heat resistance of the back glazing	$R_b = 0.18$ m²K/W
Heat transfer coefficient inside	$h_i = 8$ W/m²K
Electrical efficiency	$\eta_{el} = 0.12$
Emission coefficient	$\varepsilon = 0.88$

Solution:

The temperatures from Example 6.1 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	$h_{ca} = 5.1$ W/m^{2}K
Heat transfer coefficient glazing fluid	$h_{cb} = 3.5$ W/m^{2}K
Heat transfer coefficient for radiation	$h_r = 2.9$ W/m^{2}K
Thus a mean fluid temperature results	$T_f = 19.6$°C.

If the mean fluid temperature is inserted into Equations (6.7) and (6.8), this produces for the

mean PV module temperature	$T_a = 40.9$°C
glazing temperature on the gap side	$T_b = 26.5$°C.

The fluid outlet temperature after 6.5 m height is $T_{f,out} = 27.2$°C,
i.e. the PV facade has warmed up the ambient air by 17.2°C.

The temperature distribution over the facade height for the above boundary conditions clarify 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.

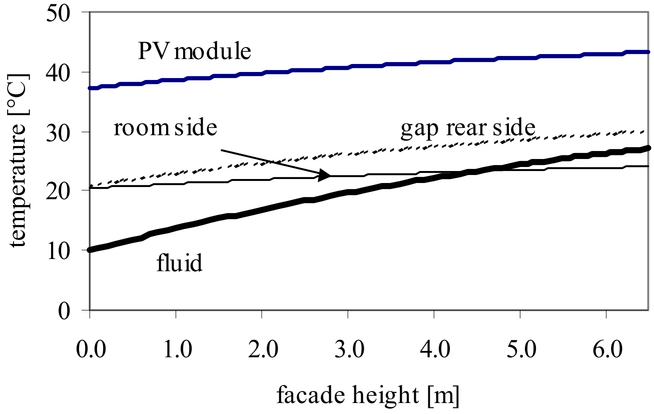


Figure 6.6: Rise in temperature of the ambient air entering through a back-ventilated PV facade. The boundary conditions are 800 W/m^2 irradiance, 3 m/s wind velocity, 10°C outside temperature, 20°C room air temperature and a flow velocity in the gap of 0.3 m/s.

6.3 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 (Versluis *et al.*, 1997).

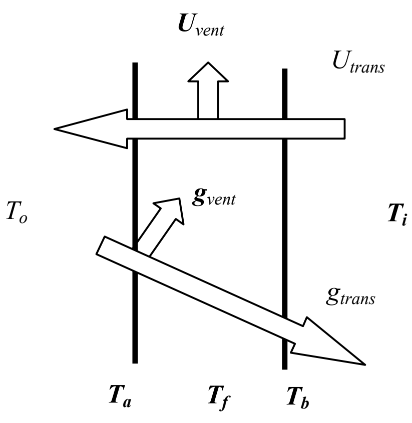


Figure 6.7: Characteristic values for the thermal characterisation of a back-ventilated facade.

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 which 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.

$$\dot{Q}_{trans} = U_{trans}\left(T_i - T_o\right) - g_{trans} G \tag{6.21}$$

$$\dot{Q}_{vent} = U_{vent}\left(T_i - T_o\right) + g_{vent} G \tag{6.22}$$

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 $\overline{T}_b$. For this definition, the heat flow from the room $U_b(T_i - T_b)$ is normalised to the temperature difference between the room and the environment.

$$U_b(T_i - \overline{T}_b) = U_{trans}(T_i - T_o) \tag{6.23}$$

$$U_{trans} = \frac{U_b(T_i - \overline{T}_b)}{(T_i - T_o)} \tag{6.24}$$

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%.

$$g_{trans} = \frac{A_{glass}}{A_{module}} \tau_{PV-glass} \tau_{gap\ rear} = 0.15 \times 0.9 \times 0.8 = 0.108 \tag{6.25}$$

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.

$$\dot{Q}_{vent} = h_{ca}(\overline{T}_a - \overline{T}_f) + h_{cb}(\overline{T}_b - \overline{T}_f) \tag{6.26}$$

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,

$$\dot{Q}_{vent} = \left(\frac{1-e^{-ZL}}{ZL}\right)(T_i D_1 - T_o\ (D_5 - D_2)) + \left(\frac{1-e^{-ZL}}{ZL}\right)\left(GD_4 - \dot{Q}_{el} D_3\right) \tag{6.27}$$

with L describing the height of the facade.

In agreement with Equation (6.22), $\dot{Q}_{vent}$ 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}:

$$U_{vent} = \frac{\left(\frac{1-e^{-ZL}}{ZL}\right)\left(T_i D_1 - T_o\ (D_5 - D_2)\right)}{T_i - T_o} \tag{6.28}$$

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.

$$g_{vent} = \left(\frac{1-e^{-ZL}}{ZL}\right)\left(D_4 - \frac{\dot{Q}_{el} D_3}{G}\right) = \left(\frac{1-e^{-ZL}}{ZL}\right)\left(D_4 - \eta_{el} D_3\right) \tag{6.29}$$

Example 6.3

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.

$U_{trans} = -2.14$ W/m^{2}K

$U_{vent} = 1.07$ W/m^{2}K

$g_{vent} = 0.178$

Thus the ventilation gains at an irradiance of 800 W/m^2 amount to

$$\dot{Q}_{vent} = U_{vent}\left(T_i - T_0\right) + g_{vent} G = \underbrace{1.07 \frac{W}{m^2 K}(20-10)K}_{10.7} + \underbrace{0.178 \times 800 \frac{W}{m^2}}_{142.4} = 153.1 \frac{W}{m^2}$$

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:

$\dot{Q}_{trans} = -97.4$ W/m^2.

6.4 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 EN832. 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

$$\bar{g}_{vent} = \frac{\sum_{j=1}^{hours\ per\ month} g_{vent,j} G_j}{\sum_{j=1}^{hours\ per\ month} G_j} \tag{6.30}$$

The useful energy from heat recovery is calculated by the temperature difference-weighted $\bar{U}_{vent}$ -value.

$$\bar{U}_{vent} = \frac{\sum_{j=1}^{hours\ per\ month} U_{vent,j} \left(T_i - T_{o,j}\right)}{\sum_{j=1}^{hours\ per\ month} \left(T_i - T_{o,j}\right)} \tag{6.31}$$

The entire transmission heat loss of the room is expressed by a mean heat transfer coefficient $\bar{U}_{trans}$.

$$\bar{U}_{trans} = \frac{\sum_{j=1}^{hours\ per\ month} U_{trans,j} \left(T_i - T_{o,j}\right)}{\sum_{j=1}^{hours\ per\ month} \left(T_i - T_{o,j}\right)} \tag{6.32}$$

Since in the monthly energy balance procedure only solar gains are taken into account, $\bar{U}_{vent}$ can also be deducted 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}). When active solar cooling is used, the useful 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.

$$Q_{vent} = \overline{g}_{vent} G_m + \overline{U}_{vent} \left(\overline{T}_i - \overline{T}_o \right) n_h \tag{6.33}$$

The mean ventilation g-value $\overline{g}_{vent}$ 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.

$$Q_{trans} = \overline{U}_{trans} \left(\overline{T}_i - \overline{T}_o \right) n_h - \overline{g}_{trans} G_m \tag{6.34}$$

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 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).

Table 6.2: Climatic boundary conditions, ventilation gains and transmission heat losses of the back-ventilated south-facing facade in Barcelona.

Month	G_m [kWh/m²]	T_o [°C]	n_h [-]	$\overline{g}_{vent} = \eta_{th}$ [-]	$\overline{U}_{vent}\left(\overline{T}_i - \overline{T}_o\right)n_h$ [kWh/m²]	$\overline{U}_{trans}\left(\overline{T}_i - \overline{T}_o\right)n_h$ [kWh/m²]	Q_{trans} [kWh/m²]	Q_{vent} [kWh/m²]
January	82	9.75	744	0.142	8.05	11.00	2.16	19.67
February	88	9.95	672	0.130	7.03	9.30	–0.20	18.50
March	106	11.26	744	0.128	6.98	8.08	–3.37	20.58
April	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
June	80	20.06	720	0.122	0.17	–3.69	–12.32	9.90
July	88	23.65	744	0.118	–2.29	–9.06	–18.61	8.15
August	100	23.46	744	0.129	–2.20	–9.60	–20.36	10.64
September	108	21.26	720	0.134	–0.71	–6.84	–18.48	13.75
October	107	17.01	744	0.130	2.26	–0.52	–12.12	16.25
November	90	12.70	720	0.137	5.42	6.16	–3.57	17.73
December	84	10.75	744	0.146	7.10	9.25	0.12	19.45
Sum/ Mean value	1107	15.7	8760	0.13	40	22	–98	184

In Stuttgart the average thermal efficiency is somewhat higher at 15%. Despite the high U-value of the non-coated double glazing, 4 W/m^{2}K, 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 1.5-1.8 W/m^{2}K. 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.

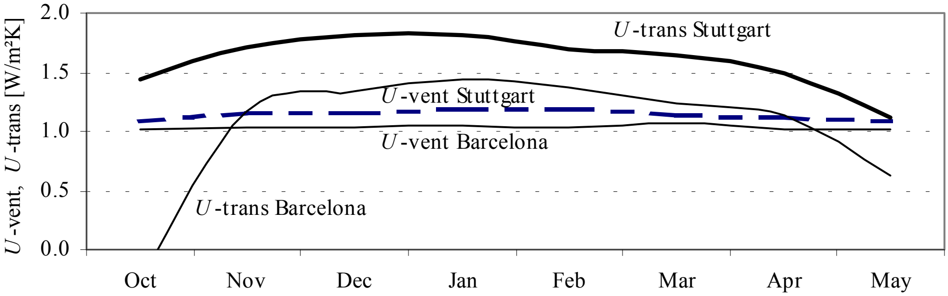


Figure 6.8: Mean monthly component characteristic values of the back-ventilated photovoltaic facade with conventional double glazing as the gap rear (U = 4 W/m^{2}K) in Stuttgart and Barcelona.

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 too 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 83–93 kWh/m^{2}a in Stuttgart, depending on the gap rear construction, the solar ventilation gains of a vitreous cavity facade are only 15 kWh/m^{2}a. 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 250–285 kWh/m^2, and must in all cases be controlled with shading devices. With commercialg-values for external sun protection of about 20%, overheating problems in the summer must be expected in such all-glass facades.