1. Introduction

Energy consumption in buildings is one of the most important challenges towards a sustainable society.\[1,2\] The building sector accounts for almost one third on the global final energy consumption.\[3\] Across Europe, about 30% of the final energy is used for space heating and cooling. If no counter-actions are undertaken, this share will increase by more than 50% by 2050.\[3\] At the same time, there has been an accelerating trend of designing residential as well as nonresidential buildings with large glazing size.\[4\] Aside overall appeal, this has been driven largely by comfort considerations such as, e.g., providing a spacious feeling, creating a visual and psychological connection between interior and exterior environment for users, and reducing energy consumption for artificial lightning.\[5\] However, challenges also arise, most prominently, a possible increase in the building’s energy demand due to higher heating load because of higher thermal transmission as compared to opaque façades, and higher cooling load because of enhanced solar heat gain. Such an effect would be counterproductive in view of the further sustainable development goals (SDGs).

Several routes are therefore being considered in order to overcome this issue. Many of those target transformation of the passive glazing into a functional, smart component of the building envelope, which could contribute to energy conservation. For example, smart windows have been brought forward which include chromogenic functions, such as thermochromic, electrochromic, photochromic, and gasochromic materials.\[6–9\] Such devices enable a dynamic regulation of glazing transmittance, absorptance, and reflectance, often in order to reduce the cooling energy demand.

In a previous report, we introduced a novel approach towards large-area fluidic windows which implement a liquid heat storage for active cooling or heating.\[10\] Such windows are manufactured from a rolled capillary sheet laminated with a thin cover glass sheet, whereby the internal capillary channels are used for fluid transport. This turns the glazing into a thermo-active building system (TABS).\[10\] Figure 1 shows the design concept of such a capillary glass panel and a photograph of a prototype device. By replacing normal glass sheet in a state-of-the-art triple glazing with the capillary glass panel, different application cases can be realized, for example, as a heating or cooling panel for indoor air-conditioning, as shown in Figure 1.
(right).

Compared to conventional all-air-based systems, the fluidic window device operates at a lower temperature difference between the circulating fluid and the target space for air-conditioning, thus, higher energy efficiency can be achieved when using with a heat pump. Moreover, if combined with a ventilation system, the ventilation system only needs to meet the air-exchange rate requirements and a lower air flow rate is required. Consequently, the draught rate can be reduced, and a higher thermal comfort is expected.

We previously considered the functionality of such devices as a decentral heat pump, whereby auxiliary photovoltaic energy harvesting was used for driving a heat pump operating between two capillary devices facing indoors and outdoors, respectively. It was demonstrated that such a system’s efficiency strongly depends on the window-to-floor ratio of a given model room, reaching competitive performance at a value of about 0.4. However, when implemented in a more straightforward way as a triple glazing on building level with only a single capillary pane (Figure 1, denoted use-case (B) in Ref. [13]), it remained unclear how the building’s glazing area affects the device energy performance. On one hand, larger glazing area requires lower temperature difference and improves the system efficiency. On the other hand, enlarging the glazing area will also increase the cooling energy consumption due to higher solar heat gain. Moreover, it is interesting to see whether the requirements on the thermal comfort can be met when using the device for room heating and cooling under different window-to-wall ratios. Therefore, this report focuses on a qualitative and quantitative analysis of the impact of the building’s window-to-wall ratio on the triple glazed fluidic window energy efficiency and consumption. By way of example, the results and methodology of this analysis may provide more general guidelines for smart windows device integration into buildings for architects and HVAC engineers from the energy perspective.

![Figure 1. Design concept of the capillary glass panel (left) and application case for indoor heating and cooling (right). The application case corresponds to use-case (B) in Ref. [13].](image)

2. Simplified Analysis

To show how energy conservation can be achieved with the proposed device during the heating period and how the window-to-wall ratio affects the energy consumption, we first consider two cases: (A) a room with a triple glazed window and a convector and (B) a room with a triple glazed fluidic window, as shown in Figure 2. Here, we assume that the glazing’s $U$-value for both cases is identical. We used an equivalent electrical resistance network to present the steady-state behavior of the room and the triple glazed fluidic window. The room model derives from the European standard BS EN 12 831. To describe the thermal behavior of the fluidic window, we adopt the resistance model proposed by Fraaß, Koschenz, and Lehmann. Moreover, we introduce the over-temperature as the difference of the temperature to the outdoor ambient temperature $\vartheta = \vartheta_1 - \vartheta_0$ for simplification.

2.1. Steady-State Analysis for Heating

2.1.1. Heating Energy Demand of the Triple Glazed Fluidic Device

For the case A, the heating energy demand of the convector can be determined as follows

$$\dot{Q}_h = \frac{\vartheta_1}{R_f} \cdot \frac{\vartheta_1}{R_w} \cdot \frac{\vartheta_1}{R_V}$$  \hspace{1cm} (1)

where $\vartheta_1$ is the room over-temperature in K, $R_f$ is the thermal resistance of the window in K $\cdot$ W$^{-1}$, $R_w$ is the thermal resistance of exterior wall in K $\cdot$ W$^{-1}$ and $R_V$ is the thermal resistance for ventilation loss in K $\cdot$ W$^{-1}$. $R_f + R_w$ is the total thermal resistance of the window ($R_t = R_f + R_w$), as shown in Figure 2A. For comparison, we split this resistance into two parts $R_t$ and $R_V$. The meaning of the two thermal resistances will be discussed later. The $\dot{Q}_h$ is also known as the heating load of the room.

The thermal resistances $R_t$, $R_w$, and $R_V$ are determined as follows

$$R_f = \frac{1}{U_f \cdot A_f} = \frac{1}{U_f \cdot a_f \cdot A_e}$$  \hspace{1cm} (2)

$$R_w = \frac{1}{U_w \cdot A_w} = \frac{1}{U_w \cdot (1-a_f) \cdot A_e}$$  \hspace{1cm} (3)

$$R_V = \frac{1}{0.34 \cdot n \cdot V_e}$$  \hspace{1cm} (4)

where $U_f$ is the $U$-value of the window in W K$^{-1}$ m$^{-2}$, $A_e$ is the total room envelop area, $A_w$ is the window area, $a_f$ is the ratio of the window area to the envelop area (also known as window-to-wall area), $U_w$ is the $U$-value of the exterior wall in W K$^{-1}$ m$^{-2}$, $V_e$ is
the room air volume, \( n \) is the air exchange rate in h\(^{-1}\). 0.34 is the product of the air density and the air heat capacity in Wh m\(^{-3}\). For simplification, we assume that the air density and heat capacity remain constant.

In case B, the room is heated with the triple glazed fluidic window. With the equivalent electrical network in Figure 2B, two important variables, mean conductive layer temperature \( t_m \) and the heating energy demand \( Q_f \) can be determined. The concept of a mean conductive layer temperature \( t_m \) comes from the international standard BS EN ISO 11 855.[16] It describes the average temperature of the fluid layer in the triple glazed fluidic window. Therefore

\[
\dot{Q}_f = \frac{\vartheta_m}{R_a} + \frac{\vartheta_m - \vartheta_i}{R_i} \tag{5}
\]

\[
\vartheta_m = \vartheta_i \left(1 + \frac{R_i}{R_w} + \frac{R_i}{R_V}\right) \tag{6}
\]

where the \( R_i \) is the resistance between the conductive layer and indoor air in K W\(^{-1}\), \( R_a \) is the resistance between the conductive layer and outdoor air in K W\(^{-1}\). Applying the window-area related resistances \( r_i \) and \( r_a \) in K m\(^2\) W\(^{-1}\), we can rewrite the resistances \( R_i \) and \( R_a \) into

\[
R_i = \frac{r_i}{a_i A_e} \tag{7}
\]

\[
R_a = \frac{r_a}{a_i A_e} \tag{8}
\]

As a result, we can simplify the Equation (5) into

\[
\dot{Q}_f = \frac{\vartheta_m}{R_a} + \frac{\vartheta_m - \vartheta_i}{R_i} = \frac{\vartheta_i}{R_a} \left(1 + \frac{R_i}{R_w} + \frac{R_i}{R_V}\right) + \frac{\vartheta_i}{R_a} \tag{9}
\]

Dividing Equation (9) with 1 and applying (7) and (8) yields

\[
\frac{\dot{Q}_f}{Q_n} = 1 + \frac{r_i}{r_a} \tag{10}
\]

Equation (10) reveals an interesting fact that the ratio between the heating energy demand of the triple glazed fluidic window \( Q_f \) and it of the reference convector case \( Q_n \) is constant and only dependent on the window-area related resistances \( r_i \) and \( r_a \). This ratio is higher than 1 due to the extra heat loss from the triple glazed window to the exterior environment. This means that the heating energy demand of the triple glazed fluidic window is higher than the room’s heating load. To reduce the heating energy demand of the device, one may consider enlarging the \( r_a \), which means that the triple glazed fluidic window requires thermal insulation. This can be accomplished with help of low-e coating.

The \( r_i \) is the sum of two partial resistances (1) The thermal resistance between the conductive layer and the window inner surface. This resistance is negligible in our

![Figure 2. A) room with convector and its equivalent resistance network B) room with triple glazed fluidic window and the equivalent resistance network.](image-url)
case, since the cover glass of the capillary fluidic glass is less than 1 mm thick, which yields a resistance value of less than 0.001 K m² W⁻¹.

(2) The thermal resistance between the window inner surface and indoor environment; this resistance is the reciprocal of the heat exchange coefficient. European standard BS EN 673 suggests that the heat exchange coefficient between the window inner surface and indoor environment can be assumed as 77 W m⁻² K⁻¹.[17] This yields a resistance of \( r_i = 0.13 \) K m² W⁻¹.

For example, if the \( U \)-value of the proposed device is 0.75 W m⁻² K⁻¹, with the following relationship

\[
U_f = \frac{1}{r_i + r_a} \tag{11}
\]

we can determine the resistance \( r_a = 1.2 \) K m² W⁻¹. With this configuration, the heating energy demand of the triple glazed fluidic device would be 10.8% higher than that of the reference convector case.

2.1.2. Mean Conductive Layer Temperature, Energy Efficiency and Primary Energy Consumption

One advantage of the proposed fluidic window is that it requires low temperature for heating once enough glazing area is available. Understanding this effect requires a deeper insight into Equation (6). For simplification, we define an equivalent room depth \( d_e \) as the ratio of the room air volume to the total room envelop area \( V_c / A_c \). Inserting Equations (2) and (8) back to six yields

\[
\vartheta_m = \vartheta_i \left[ (1 - r_i U_w) + \frac{r_i}{a_e} \left( U_w + 0.34 n d_e \right) \right] \tag{12}
\]

Equation (12) reveals an important fact that the required mean conductive layer over-temperature \( \vartheta_m \) is proportional to the equivalent room depth and inversely proportional to the window-to-wall ratio \( a_c \). To illustrate how the required mean conductive layer temperature behaves under various conditions, we consider a test room with different equivalent depth and equipped with the fluidic window. The design room temperature is 20 °C. The air exchange rate \( n \) is 0.5 h⁻¹. The \( U \)-value of the exterior wall \( U_w \) is 0.28 W m⁻² K⁻¹, as required by the German Energy Saving Ordinance.[18] The thermal resistance between the indoor air and the conductive layer \( r_i \) is 0.13 K m² W⁻¹. Figure 3A presents the required mean conductive layer temperature for −10 °C outdoor temperature for different window-to-wall ratio \( a_c \) and equivalent depth \( d_e \). −10 °C outdoor temperature represents the typical design outdoor temperature for cities in Germany.

The mean conductive layer temperature can be approximately seen as the inner surface temperature, since the device cover glass is very thin and thus the thermal resistance between the conductive layer and the inner surface can be neglected. An inner surface temperature higher than 35 °C should be avoided, because it results in thermal discomfort. With this consideration, we come to one important criterion that rooms with very low window-to-wall ratio and high equivalent room depth are not suitable for the proposed concept. For the 5 m × 3.5 m × 2.5 m test room, the minimum window-to-wall ratio is 0.27.

To estimate system efficiency, we adopt the simplified heat pump model proposed in the European standard BS EN 13612-2.[19] In this model, the heat pump coefficient of performance (COP) is based on the ideal efficiency and the Carnot efficiency \( \eta \). The air-to-water heat pump coefficient of performance can be calculated as follows

\[
\text{COP} = \frac{\vartheta_i + \Delta t_i + 273.15}{(\vartheta_i + \Delta t_i) - (\vartheta_i - \Delta t_i)} = \frac{\vartheta_i + \Delta t_i + 273.15}{\vartheta_i + \Delta t_i + \Delta t_a} \tag{13}
\]

where \( \eta \) is the Carnot efficiency, \( \Delta t_i \), and \( \Delta t_a \) are the temperature difference between the system inlet temperature and the condensing temperature, and the outdoor air temperature and the evaporating temperature, respectively. For a modern air-to-water heat pump, the Carnot efficiency is around 0.4, and the \( \Delta t_i \) and \( \Delta t_a \) can be assumed to be 5 and 10 K.[19, 20] For simplification, we assume that the circulating fluid is heated by an air-to-water heat pump, and the inlet temperature \( t_i \) is 3 K higher than the mean conductive layer temperature, based on the fact that the water inlet outlet temperature difference of a radiant heating system often varies between 5 and 10 K.[21] Combining Equations (12) and (13) yields an estimation of COP. Figure 3B shows the COP for different window-to-wall ratio at 5 °C outdoor temperature and equivalent depths for different window-to-wall ratios. These results show despite higher energy efficiency at higher window-to-wall ratio, the improvement is relatively limited. For the 5 m × 3.5 m × 2.5 m test room, the expected COPs for the window-to-wall ratio of 0.5 and 0.9 are 3.42 and 3.56, respectively. This corresponds to an efficiency improvement of 6%.

The primary energy demand for both case A and B in Figure 2 is estimated as follows

\[
Q_p = f_p \cdot \frac{Q}{f_c} \tag{14}
\]

where \( f_p \) is the primary energy factor, \( Q \) is the heating energy demand, \( f_c \) is the transfer coefficient for the site energy. The convector in case A in Figure 2 normally requires a high system temperature and thus is often combined with a boiler. For a natural gas boiler, the \( f_c \) is 1.11 and \( f_p \) is 1.1 in Germany.[18, 22] For the triple glazed fluidic window, the \( f_c \) is the COP value and \( f_p \) is the primary energy factor for electricity 1.8.[18]

Figure 3C shows the primary energy demand of the reference convector case A and of the fluidic window. For the room with very low window-to-wall ratio, the primary energy demand of the triple glazed fluidic window is higher than the convector due to bad heat pump efficiency. Recall that the minimum window-to-wall ratio is 0.27 for the 5 m × 3.5 m × 2.5 m test room to avoid thermal discomfort. In the range of 0.27–1, the primary energy demand of the fluidic window becomes lower than the convector and the difference becomes larger with increasing window-to-wall ratio, as shown in Figure 3D. With these results, we confirm again that buildings with not too low window-to-wall ratio are suitable for the triple glazed fluidic window device because of both thermal comfort and energy conservation.
Furthermore, we can observe that the electricity demand of the fluidic window reaches the minimum near the lowest allowed window-to-wall ratio, then starts to increase. Because the heating energy demand increases linearly and the COP increases nonlinearly with the window-to-wall ratio, one may expect that higher window-to-wall ratio results in higher electricity demand during the heating period.

2.2. Simplified Analysis for Cooling

2.2.1. Fluid Inlet Temperature and Cooling Load

Similar as the heating period, the circulating fluid temperature during the cooling period cannot be randomly selected, because a surface temperature below the dew point will cause condensation problem. Thus, the cooling capacity from the fluidic glazing device is limited. An empirical fluid inlet temperature to avoid condensation is 16 °C. For the cooling case, it also needs to be studied whether the cooling capacity with 16 °C inlet temperature can fully cover the cooling load.

For the cooling load, it can be expected that the cooling load correlates approximately linearly with the window-to-wall ratio, as shown in Figure 4A. Because of other heat source such as the heat dissipation of people, electronic devices, and lightning, the intercept of this line should be positive. On the other hand, the maximum cooling capacity is also linear to the window-to-wall ratio, but with an intercept of zero. The intersection point of these two lines is an important criterion for the cooling system design. If the window-to-wall ratio is lower than the intersection point, then the device maximum cooling capacity is not able to cover the required cooling energy demand. Therefore, we come to a similar conclusion as for the heating case: a too low window-to-wall ratio is not suitable for the proposed device.

Accurate cooling load calculation requires a time-dependent room or building simulation. Therefore, we conducted a room simulation for the 5 m × 3.5 m × 2.5 m test room with our simulation model. We adopted the wall structure of room type medium from German standard VDI 2078, which represents a room with intermediate thermal mass. The room temperature setpoint is 25 °C. The internal heat gain and schedule of the room is adopted from the DOE prototype office building middle. The peak heat gain per floor area dissipated

![Figure 3. A) required mean conductive layer temperature for various room depth and window-to-wall ratio for design case B) estimated heat pump COP for outdoor temperature at 5 °C. C) Primary energy demand of the convector (case A) and triple glazed fluidic window (case B). D) Primary energy demand difference between case A and case B.](image-url)
from the equipment, lightning, and people are 8.1, 9.7, and 5.4 W m\(^{-2}\), respectively. The glass three of the triple glazed device is assumed to be solar reflective glass and the glass 1 is assumed to be low-e glass. This configuration yields a solar heat gain coefficient (SHGC) of 0.25. We varied the window-to-wall ratio from 0.4 to 0.9. We discovered that the room with a window-to-wall ratio of smaller than 0.4 cannot be maintained at 25 °C during the cooling period. The cooling load for other window-to-wall ratios are shown in Figure 4B.

### 2.2.2. Energy Demand for Cooling

To see how the energy conservation can be achieved during the cooling period with the triple glazed fluidic window analytically, we adopted the monthly energy balance method.\(^{[25,26]}\) The main concept can also be presented in form of a similar equivalent electrical network as for heating, as shown in Figure 5. The difference between the monthly energy balance method and the method for the heating case analysis is that the monthly energy balance method calculates the energy demand over a period instead of instantaneous demand. With this simplification, the impact of the thermal mass does not need to be considered and the time-dependent problem can be simplified into a steady-state one.

Again, we consider two cases: (A) room with convector and (B) room with the triple glazed fluidic window. Assume that the windows in both rooms have the same solar heat gain factor. With the Kirchhoff’s law, we can determine the heating energy demand for both cases. For case A

\[
Q_n = \frac{\dot{q}_i}{R_f} + \frac{\dot{q}_i}{R_s} + \frac{\dot{q}_v}{R_V} - Q_b
\]

where \(Q_n\) is the hourly average cooling energy demand during in observed period in W, \(Q_b\) is the hourly average heat gain in W. Meanwhile, for case B

\[
Q_f = \frac{\dot{q}_m - \dot{q}_i}{R_i} + \frac{\dot{q}_m}{R_a}
\]

\[
\dot{q}_m = -Q_b R_i + \dot{q}_v \left( 1 + \frac{R_i}{R_v} + \frac{R_i}{R_s} \right)
\]

Inserting Equation (17) back into Equation (16), we can determine the hourly average cooling energy demand of the fluidic window \(Q_b\). Dividing \(Q_b\) with \(Q_n\) returns the same result as Equation (10). It means that the cooling energy demand of the fluidic window is higher than the convector and the ratio is constant.

The heat gain \(Q_g\) of a room consists of three parts: solar heat gain through glazing, solar heat absorption through opaque walls, and internal heat gain from people, device, and lighting.\(^{[25]}\) suggests estimating the hourly heat gain with the utilization factor \(\eta\)

\[
Q_g = \eta \left( g \cdot A_f \cdot I + r_{se} \cdot U_w \cdot A_s \cdot (\alpha I - F h_s \Delta t_e) + q_i \cdot A_n \right)
\]

where \(g\) is the solar heat gain coefficient of the window, \(I\) is the hourly mean solar radiance intensity in W m\(^{-2}\), \(r_{se}\) is the outer thermal resistance of the exterior wall, \(\alpha\) is the outer surface absorptance of the exterior wall, \(F\) is the view factor between the wall and the sky, \(h_s\) is the transfer coefficient for radiative heat exchange, \(\Delta t_e\) is the temperature difference between the ambient air and the sky, \(q_i\) is the hourly mean internal heat gain per floor area and \(A_n\) is the floor area.

However, for conventional cooling systems like convectors, a low inlet temperature such as 6 °C is often required. For the proposed fluidic window, we consider the inlet temperature to be 16 °C, as discussed previously. Because of higher inlet temperature, the heat pump efficiency for cooling is enhanced. The heat pump efficiency for cooling can be estimated as follows

\[
EER = \frac{t_v - \Delta t_i + 273.15}{(t_s + \Delta t_s) - (t_v - \Delta t_i)} = \frac{t_v - \Delta t_i + 273.15}{-\dot{q}_v + \Delta t_i + \Delta t_s}
\]

With the system efficiency and the cooling energy demand, we can determine the primary energy demand for cooling. Again, we consider the 5 m × 3.5 m × 2.5 m test room. Assume that the mean indoor temperature is 25 °C, the mean outdoor temperature is 30 °C, the utilization factor is 0.8, the hourly mean internal heat gain per floor area is 5 W m\(^{-2}\), the hourly mean solar radiance intensity is 127 W m\(^{-2}\), which is the mean solar radiance intensity on the south in Berlin in August.\(^{[27]}\) Figure 6 shows the estimated mean primary energy demand for both cases. It can be seen that for different window-to-wall
ratios, the primary energy consumption of the proposed triple glazed fluidic window is lower than the convector due to higher inlet temperature and higher efficiency. Moreover, we can see that the primary energy consumption increases linearly with the window-to-wall ratio.

3. Building Simulation

In the previous section, we derived expressions for the required mean conductive layer temperature, the heating and cooling energy demand and the system efficiency of triple glazed window, and conducted a theoretical comparison to conventional HVAC systems. The results show that compared to conventional heating and cooling systems, using the triple glazed fluidic window will reduce the heating and cooling energy consumption effectively. In this section, we concentrate on studying the impact of the building’s window-to-wall ratio on the triple glazed fluidic window’s energy consumption. As discussed in the previous section, the primary energy consumption of the fluidic window becomes higher with larger window-to-wall ratio for both heating and cooling. However, a larger window-to-wall ratio also results in higher solar heat gain in winter, which compensates a part of the heating energy demand. To see how the window-to-wall ratio affects the device heating energy consumption and to evaluate the annual primary energy consumption, a long-period building simulation is required.

3.1. Simulation Setup

We select the prototype office building middle from the US Department of Energy (DOE) as our model object. This prototype building consists of three floors. Each floor has a dimension of 50 m × 33 m × 2.74 m. To simplify the modeling, DOE proposed to divide each floor into five thermal zones: four exterior zones with different orientations facing south, east, north, and west and one interior zones. Figure 7A presents the 3D visualization of this building. Figure 7B shows the time-dependent total heat gain per floor area of a week. The detailed values

Figure 5. A) room with convector and its equivalent resistance network. B) room with triple glazed fluidic window and the equivalent resistance network.

Figure 6. Mean primary energy demand per floor area.
can be found on the website of US DOE. Furthermore, this office building is assumed to be in Berlin. For the simulation, we used the Test Reference Year 2017 data from the Germany’s National Meteorological Service.

In the simulation study, we considered the exterior zones are air-conditioned with the proposed triple glazed fluidic device. The glass 1 is assumed to be a solar reflective glass and the glass 3 is assumed to be a low-e glass in order to reduce solar heat gain in summer and thermal transmission loss in winter. Their physical indicators are listed in Table 1. The interior zone is air-conditioned with an all-air based system. Enough fresh air needs to be supplied to the zones to meet the requirement on the air quality. German standard DIN EN 15 251 recommends a fresh air flow rate of 1.7 l s$^{-1}$ m$^{-2}$ per floor area. This corresponds to an air exchange rate of 2.23 h$^{-1}$. To avoid thermal discomfort, the supply air is assumed to be preheated to 16 °C during the heating period and precooled to 26 °C during the cooling period. If the outdoor air temperature is higher than 16 °C during the heating period or lower than 26 °C during the cooling period, then the outdoor air will not be pre-handled.

To enhance the system efficiency, we used an outdoor-temperature-dependent inlet temperature during the heating period. Furthermore, we consider a bivalent operation for the heat pump: if the outdoor ambient temperature drops below the bivalent point, the heat pump will be switched off and a secondary heater (e.g., boiler) will be switched on to ensure the high energy efficiency of the system. If the outdoor ambient temperature rises above the bivalent point, the heat pump will be switched on again. The thumb-up value for the bivalent point is 0–5 °C. For the simulation study, we select 0 °C as the bivalent point. For the simulation, a dry free cooler is also included to improve the cooling period efficiency. We assume that if the free cooler is able to supply 16 °C water, the cooling energy required by the system should be provided by the free cooler, otherwise the compressor will be switched on. For the switching between the heating and cooling mode, we assume that if the exhaust air temperature of the building is higher than 23 °C, then the heat pump will be in cooling mode, otherwise in heating mode.

We adopted the adaptive room temperature setpoint suggested by the German standard DIN EN 15 251. Figure 8 shows the adaptive room temperature setpoint. The DIN EN 15 251 suggests that the room temperature setpoint as well as the comfort temperature range should be dependent on the outdoor ambient air temperature $t_a$. The setpoint should not be lower than 22 °C and not be higher than 26 °C. Therefore, the room temperature setpoint $t_i,0$ can be expressed with the following equation

$$t_i,0 = \frac{22 + 26}{2} + \frac{22 - t_a}{2}$$

![Figure 7](image_url) A) DOE prototype office building middle. B) Heat gain curve. C) HVAC system. D) Chiller with free cooling.

| Physical properties | Clear glass | Low-e glass | Solar reflective glass |
|---------------------|-------------|-------------|-----------------------|
| Thickness [mm]      | 6           | 6           | 6                     |
| Thermal conductivity [W m$^{-1}$ K$^{-1}$] | 1           | 1           | 1                     |
| Transmittance front [£] | 0.85       | 0.6         | 0.3                   |
| Transmittance back [£]  | 0.85       | 0.6         | 0.3                   |
| Reflectance front [£] | 0.08       | 0.25        | 0.45                  |
| Reflectance back [£]  | 0.08       | 0.2         | 0.55                  |
| Emissivity front [£] | 0.837      | 0.05        | 0.837                 |
| Emissivity back [£]  | 0.837      | 0.837       | 0.05                  |
\[ t_{o,0} = \max(\min(18 + 0.25 t_o, 26), 22) \]

We considered five different window-to-wall ratios during the simulation study: 0.5, 0.6, 0.7, 0.8, and 0.9. Since our major focus is the energy performance of the triple glazed fluidic device and the HVAC system configuration is the same for different window-to-wall ratio, the energy consumption of the interior zone and the ventilation system can be considered the same and will not be included in the results discussion part. The description of the modeling methods for building, heat pump and free cooler can be found in the appendix. The implementation of the thermal models of the building and the HVAC system is conducted in Wolfram Mathematica.

### 3.2. Simulation Results

**Figure 9A** presents the operation temperature curves of the exterior zones for the window-to-wall ratio of 0.5. The results show that the temperature of all the exterior area can be regulated within a comfortable range. For all other window-to-wall ratio, the results are similar. **Figure 9B** presents the thermal comfort evaluation method according to DIN EN 15 251. The x-axis of the graph is the outdoor air temperature and the y-axis of the graph is the indoor operative temperature. DIN EN 15 251 specifies that the operative and the corresponding outdoor air temperature must be within the range marked with dashed line for 95% of the occupied time throughout the whole year. The simulation results for all window-to-wall ratios satisfy the requirement. For the reason of space, the temperature curves for other window-to-wall ratios will not be presented here.

**Figure 10A** shows the energy generation of different heaters and coolers for the triple glazed device energy consumption. The heating energy generation from the boiler increases slightly with the window-to-wall ratio. This behavior is as expected, since higher window-to-wall ratio leads to higher energy demand in cold days. The heat pump heating energy generation for different window-to-wall ratios does not differ from each other significantly. However, to keep the building indoor temperature in the comfortable range, the heat pump needs to generate more cooling energy due to the higher solar heat gain. **Figure 10B** shows the heat pump efficiency for both heating and cooling. As analyzed in the previous section, we expected limited improvement with larger window-to-wall ratio. Compared to the SCOP of the case with the window-to-wall ratio equal 0.5, 0.9 only shows an efficiency improvement of 4.5%. The system efficiency for cooling also shows similar behavior.

**Figure 10C** shows the site energy consumption of the triple glazed device per floor area. The heating energy consumption does not vary much with different window-to-wall ratio. The highest increase is only at 5%. However, the specific cooling energy increase significantly with higher window-to-wall ratio. With the SCOP and SEER value of the heat pump as well as the energy generation from the heat pump and cooler, the primary energy consumption can be determined. Hereby, we assumed the boiler to be a natural gas condensing boiler. **Figure 10D** shows the device primary energy consumption. The results show that the device primary energy consumption becomes higher with higher window-to-wall ratio.

### 4. Further Considerations

The simulation results indicate that from the energy perspective, buildings with lower window-to-wall ratio are advantageous for the implementation of the triple glazed device. However, if there is an on-device energy harvesting system with enough efficiency, such as a photovoltaic module, this relationship might change. The electricity generation of photovoltaic modules can be estimated as follows

\[ E = \eta \cdot I \cdot A_e \cdot \alpha \]

where \( \eta \) is the photovoltaic efficiency, \( I \) is the accumulated solar radiation intensity. For simplification, we assume that the photovoltaic temperature behavior for different window-to-wall ratios is similar and thus the annual average efficiency remains constant. Therefore, the electricity generation is approximately proportional to the window-to-wall ratio. **Figure 11** shows the electricity generation and device energy consumption. If the efficiency is high enough, the slope of the electricity generation becomes steeper than the energy consumption. In this case, the total energy consumption will decrease with higher window-to-wall ratio. If the transfer efficiency exceeds some critical value, the electricity generation will match the energy consumption at the window-to-wall ratio of 1. In this situation, the triple glazed device with on-device photovoltaic becomes self-sufficient. If further enhancing the transfer efficiency, the device generated more energy than it consumes from certain window-to-wall ratio, and thus even becomes a plus energy device.

### 5. Conclusions

We investigated the impact of a building's window-to-wall ratio on the energy consumption of a façade using triple glazed...
fluidic windows in terms of user thermal comfort. For this, we conducted a simplified analysis and a theoretical comparison between the triple glazed fluidic window and the conventional heating and cooling systems. Our analysis shows that the proposed device consumes less primary energy as compared to a conventional air-conditioning system such as convector.

In order to determine the relationship between the device energy consumption and the building's window-to-wall ratio accurately, we set up a detailed model for building thermal simulation. The simulation study for a model building in Berlin indicates that the requirement on the thermal comfort can be met with different window-to-wall ratios from 0.5 to 0.9. We found that the proposed device consumes less energy if the building's window-to-wall ratio is lower. However, this relationship might change if there is an on-device energy harvesting system such as a transparent photovoltaic module. In

Figure 9. A) temperature curve of the exterior zones for the window-to-wall ratio of 0.5. B) Thermal comfort evaluation according to DIN EN 15 251 for the window-to-wall ratio of 0.5.
In this case, a larger window-to-wall ratio becomes advantageous from the energy perspective. With appropriate efficiency and high window-to-wall ratio, the proposed device attains self-sufficiency or even a plus-energy performance. We suggest that the obtained guidelines for window and building design can be transferred to any type of smart window with known performance on single (laboratory) device level in order to judge the potential of large-scale implementation.

6. Experimental Section

Thermal Building Model: To simulate the heat conduction through opaque walls, the procedure proposed by Beuken was followed.[31] This model based on semidiscretization of 1D heat equation, also known as the method of lines. The discretized equation can be presented as an electrical RC-network, as shown in Figure 12A. The heat transfer through the glass facade was also determined with an equivalent electricity network, as shown in Figure 12B. This model derived from the German standard VDI 6007 part 2 and the European standard EN 673.[17,32] It had been detailed described in the previous report.[13] In this study, the heat capacity of the glass was neglected. This assumption was acceptable, since the time constant of the glass façade was very small compared to it of the wall and ceiling. The thermal behavior of the capillary glass panel can be determined with the equivalent R-network proposed by Fraaß, Koschenz, and Lehmann.[14,15]

For the boundary condition of the models, the radiative and convective heat exchange was considered separately. Hereby, constant convective heat transfer coefficient was assumed between the internal surface and the room air. The assumed coefficients corresponds to the international standard ISO 13 791.[33] For the vertical components, the coefficient was assumed to be 2.5 W m$^{-2}$ K$^{-1}$. For the horizontal components, the coefficient was assumed to be 5 W m$^{-2}$ K$^{-1}$ for the heat flow upwards and 0.7 W m$^{-2}$ K$^{-1}$ for the heat flow downwards. For the exterior surfaces, the convective heat transfer coefficient was assumed to be dependent on the wind speed $w$

$$h_{c,a} = 4 + 4 \cdot w$$  \hspace{1cm} (22)$$

The radiant heat exchange between indoor surfaces were determined with transfer matrix method.[34] For this method, accurate view factors
between surfaces were required. The view factor of surface 1 to surface 2 \( \Phi_{1 \rightarrow 2} \) was calculated as follows

\[
\Phi_{1 \rightarrow 2} = \frac{1}{A_1} \int_A \int_A \cos(\vartheta_1) \cos(\vartheta_2) \frac{dA_1}{r^2} dA_2
\]

(23)

where \( \vartheta_1 \) and \( \vartheta_2 \) are the angle between the normal vector of the surface element \( dA_1 \) and \( dA_2 \), \( r \) is the distance between two surface elements, \( A_1 \) is the area of surface 1. The view factor calculation in Equation (23) required double surface integral (DSI), which needed high computing effort. However, the double surface integral can be converted into double contour integral (DCI) by applying the Stokes’ theorem

\[
\Phi_{1 \rightarrow 2} = \frac{1}{2 \pi A_1} \oint_{\partial A_1} \oint_{\partial A_2} \ln(r) \, ds_1 ds_2
\]

(24)

where \( s_1 \) and \( s_2 \) are vector element of the contour of surface 1 and 2.\[15\]

By converting double surface integral into double contour integral, the computational time can be largely reduced.

Special care needs to be taken when calculating the view factor between two surfaces that share a common edge. Equation (24) shows singularity issue if \( r \) is zero. To avoid this problem, we adopt the analytical solution of the double contour integration along the same line

\[
\Phi = \frac{1}{2 \pi A_1} \oint_{L_1} \oint_{L_2} \ln(r) \, ds_1 ds_2 = \frac{1}{2 \pi A_1} L_1^2 (1.5 - \ln L_c)
\]

(25)

where \( L_c \) is the length of the line \( L \).\[16\]

Variable Speed Air-to-Water Heat Pump and Chiller Model: The simplified second order equation fit model was adopted to determine the air-to-water heat pump and chiller performance under full load operation

\[
\dot{Q}_f = a_1 t_s + a_2 t_1 + a_3 t_2 + a_4 t_3 + a_5 t_4 + a_6 t_5 + a_7 t_6
\]

(26)

\[
P_f = b_1 + b_2 t_1 + b_3 t_2 + b_4 t_3 + b_5 t_4 + b_6 t_5 + b_7 t_6
\]

(27)

where \( t_s \) is the leaving water temperature of the heat pump/chiller, \( t_1 \) is the entering air temperature, \( \dot{Q}_f \) is the full load heating/cooling capacity of the heat pump or chiller, \( P_f \) is the full load power input, \( a_i \) and \( b_i \) are empirical constants that can be acquired from nonlinear model fit with manufacturer’s catalogue data. In this study, the catalogue data from DAIKIN were selected.\[37\]

To estimate the part load performance, the simplified estimation method proposed in the international standard ISO 13612-2 was used.\[19\]

The part load performance was characterized by the part load factor \( f_x \), which is a correction factor for the coefficient of performance (COP) and energy efficiency ratio (EER) under part load condition. It is assumed that the COP/EER value under part load operation was dependent on the part load ratio \( x \), which defined as the ratio of required heating or cooling capacity \( \dot{Q} \) to the full load capacity of the heat pump or chiller plant \( \dot{Q}_{\text{max}}(t_s, t_1) \) at the same operation condition

\[
x = \frac{\dot{Q}}{\dot{Q}_{\text{max}}(t_s, t_1)}
\]

(28)

It is suggested if the heat pump or chiller has inverter control, then the part load factor can be assumed to remains constantly at 1 if the part load ratio is above 0.25.\[19\] Otherwise, the heat pump will work in ON–OFF mode. Therefore

\[
\text{if } x \geq 0.25 : f_x = 1
\]

(29)

\[
\text{if } x < 0.25 : f_x = \frac{4x}{3.6x + 0.1}
\]

(30)

As a result, the energy efficiency (EER and COP) can be determined as follows

\[
\text{COP} = \text{COP}_{f} \cdot f_x
\]

(31)

\[
\text{EER} = \text{EER}_{f} \cdot f_x
\]

(32)
where \( \text{COP}_f \) and \( \text{EER}_f \) is the heat pump/chiller efficiency full load efficiency.

Free Cooler: To enhance the energy efficiency, modern chillers were often equipped with free cooling function. Free cooling was useful during the trans-seasonal period, when the outdoor temperature was not very high, but the indoor environment already required cooling due to high heat gain. In this case, the chiller compressor was not switched on. Instead, the low temperature of the outdoor air can be directly used to extract the heat from the HVAC water system. Figure 7D shows a chiller with free cooling function. To estimate the cooling capacity of the free cooler, the free cooler was considered as a simple air-to-water heat exchanger with constant temperature at the air side. Thus, the heat exchange efficiency \( \varepsilon \) can be determined with

\[
\varepsilon = 1 - e^{-\text{NTU}}
\]

where NTU is the number of transfer unit. The number of transfer unit depends on various factors such as mass flow rate, water and air temperature, etc.[38] suggests neglecting other factors and only considering the impact of mass flow rate.[39] discussed and validated this assumption. Therefore, the number of transfer units can be fitted with the manufacturer’s catalogue data using the following nonlinear model.

\[
\text{NTU} = k_1 \left( \frac{m_w}{m_{w,\text{ref}}} \right)^{k_2}
\]

where \( m_{w,\text{ref}} \) is the water mass flow rate at the reference condition, \( k_1 \) and \( k_2 \) are the empirical constants that can be determined with the nonlinear fit. During the simulation, the DR35 dry cooler was selected.[40]

The electrical consumption of the free cooler is assumed to remain constant at 1.2 kW.[40]

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Conflict of Interest

The authors declare no conflict of interest.

Keywords

air-conditioning, heating ventilation, smart windows, thermo-active building system, triple-glazed fluidic windows, window-to-wall ratio

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