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CASE

The Performance Evaluation of Ventilated Windows in the Simultaneous Improvement of Energy Efficiency and Indoor Air Quality in Office Buildings: A Case Study

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Abstract: Energy efficiency and indoor air quality (IAQ) are two crucial required features in a building. Simultaneous improvement of energy efficiency and IAQ in a building can pave the way for obtaining a green building certification. This paper examined the performance of the airflow windows' supply and exhaust operating modes in energy-saving and providing IAQ criteria. The analytical zonal model coupled with the airflow network model was used to evaluate the system's thermal performance and the induced airflow. The simulation was done for an office building located in Shiraz, Iran. The results showed that the energy performance of ventilated windows is positive in nine months of the year. Compared to a conventional double-glazed window, the maximum energy savings is about 10%, which occurs in August. It is predicted that using ventilated windows in office buildings in Shiraz can improve the window's thermal performance by an average of about 5%. The results also showed that ventilated windows could provide the fresh air needed for the building in 250 days of the year to achieve the desired IAQ index (based on ASHRAE 62.1 standard). Furthermore, the effects of glass aspect ratio, airflow channel thickness, and the size of inlet/outlet openings on energy efficiency and IAQ of the suggested window were studied. Results indicated that in the climatic conditions of Shiraz, the exhaust operating mode is much more efficient than the supply mode.

Keywords: Ventilated window, Airflow window, Energy efficiency, Indoor air quality

1. Introduction

Indoor Air Quality (IAQ), as an essential part of Indoor Environmental Quality (IEQ), has a significant effect on the health and productivity of the building occupants. Poor indoor air quality has been connected to Sick Building Syndrome (SBS), lower productivity, which can be harmful to vulnerable groups such as children, young adults, the elderly, or those suffering chronic respiratory and cardiovascular diseases [1].

Keeping the air exchange rate between outdoors and indoors at an acceptable level is an effective solution to improve IAQ. The required exchange rate can be provided by a mechanical ventilation system or natural ventilation through doors, windows, and purpose-designed openings. However, providing the air exchange rate to improve

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IAQ through the existing methods often increases energy consumption [2]. According to documents produced by international organizations such as International Energy Agency (IEA) [3], ASHRAE [4], and CEN [5], regulations and advice, and guidelines were provided to target the IAQ/energy efficiency issue. Thus, finding strategies for achieving a satisfactory balance between good indoor air quality and rational energy use is a priority for researchers and designers.

Ventilated window is a novel design to solve this issue. Its structure allows exchanging air between outdoors and indoors while reducing the building cooling and heating loads resulting from the window [6]. This feature also makes the ventilated window an attractive passive system for net-zero energy or green buildings.

While most of the researchers working on the ventilated window focused on evaluating thermal performance and energy efficiency [7-12], The IAQ and energy consumption analysis has been the subject of limited studies. Gosselin and Chen [6] carried out experimentally validated computational fluid dynamics (CFD) simulations on a proposed dual airflow window to optimize the window design. Although they claimed that their proposed ventilated window improves IAQ, no quantitative results in IAQ parameters were reported. Zhang et al. [13] studied the influence of ventilated windows on indoor $PM_{2.5}$ and CO_2 concentration and the overall energy consumption. They reported that the increase of energy consumption due to indoor air quality improvement is inevitable for their specific proposed ventilated window.

Since ventilated window proved itself as an energy-efficient system applied in the building, it is necessary to evaluate its efficiency to provide the required IAQ and recognize the parameters influencing the interaction between IAQ and energy efficiencies. A practical and straightforward approach to assess the IAQ is the determination of the minimum ventilation rate. ASHRAE 62.1 [4] is a well-known reference to follow this approach. The present work aimed at assessing the performance of the naturally ventilated window in energy-saving and provision of IAQ requirement according to ASHRAE 62.1 [4]. First, the annual energy saved by the naturally ventilated window was calculated for a case study.

Furthermore, the adequacy of the induced airflow rate to provide the required fresh air was evaluated according to ASHRAE 62.1 standard [4]. In addition, a parametric study focusing on geometric factors was carried out, and the variation of each two aspects of interest was described. It is hoped that the present results can provide a more detailed picture of the benefit of using the ventilated windows in practice.

2. Description of the Ventilated Window

The subjected naturally ventilated window is composed of two parallel glass panes separated by an air cavity. As shown in Figure 1, air enters the channel through the inlet below and exits through the outlet above. Depending on the inlet and outlet of the airflow duct, four different operating modes can be defined for ventilated windows: supply mode, exhaust mode, indoor air curtain mode, and outdoor curtain mode. In the two modes of "supply" and "exhaust", the indoor and outdoor environments are connected, but these two environments are separate in the other two modes.

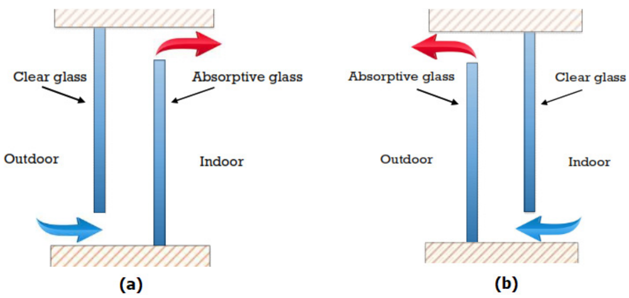

Figure 1. Schematic of the ventilated window: a) supply mode, b) exhaust mode.

For the exhaust mode, which is suitable for summer, the exterior sheet is a single pane of heat-absorbing glass, and the interior one is clear glass. However, for the supply mode used in winter, the sort order of glasses is reversed (see Figure 1). The supply operating mode can preheat the inlet air and compensate for part of the heating load of the indoors. In the naturally ventilated window, wind and buoyancy force due to the temperature difference of glass panes and air are driving forces of flow in the channel. The performance of a naturally ventilated window is the function of outdoor conditions such as velocity and orientation of wind, outdoor temperature, and solar irradiation. Thus, the induced airflow rate changes hourly and is uncontrolled.

3. Mathematical Modelling

3.1 Thermal Modelling

The present work uses the zonal model coupled with the airflow network model to predict the thermal performance of the ventilated window. In the zonal model, the ventilated window is partitioned into "n" equal sections along the channel, called "zone". As shown in Figure 2, three nodes represent the glass panes and the channel airflow temperatures in each zone. The thermal energy balance equation for each node is written to calculate the nodal temperatures. The energy balance equations are formulated under several simplifying assumptions, common in the most relevant studies [9,14].

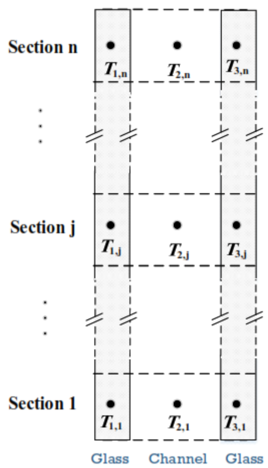

Figure 2. Schematic of the ventilated window discretization [10].

The energy flows of the nodes of section j are shown in Figure 3.

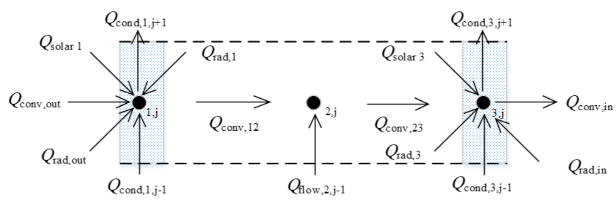

Figure 3. The energy flows of the nodes in each zone [10].

In each zone, writing the energy balance equations for nodes 1, 2, and 3 yields the following equations:

$$\dot{Q}_{conv,out} - \dot{Q}_{conv\,1,2} + \dot{Q}_{cond\,1,j-1} - \dot{Q}_{cond\,1,j+1} + \dot{Q}_{rad\,1} + \dot{Q}_{rad,out} + \dot{Q}_{solar\,1} = 0 \quad (1)$$

$$\dot{Q}_{conv\,1,2} - \dot{Q}_{conv\,2,3} + \dot{Q}_{flow\,2,j-1} = 0 \quad (2)$$

$$\dot{Q}_{conv\,2,3} - \dot{Q}_{conv,in} + \dot{Q}_{cond\,3,j-1} - \dot{Q}_{cond\,3,j+1} + \dot{Q}_{rad\,3} + \dot{Q}_{rad,in} + \dot{Q}_{solar\,3} = 0 \quad (3)$$

$\dot{Q}_{conv,out}$, $\dot{Q}_{conv,in}$, $\dot{Q}_{conv\,1,2}$ and $\dot{Q}_{conv\,2,3}$ are the convective heat transfer that occurred on the window's exterior, interior, and channel surfaces, respectively and are given by Equations (4) and (5) and (6):

$$\dot{Q}_{conv,out} = h_{conv,out} A_j (T_{1,j} - T_{out}) \quad (4)$$

$$\dot{Q}_{conv,in} = h_{conv,in} A_j (T_{3,j} - T_{in}) \quad (5)$$

$$\dot{Q}_{conv\,1,2} = \dot{Q}_{conv\,2,3} = h_{conv\,1,2} A_j (T_{1,j} - T_{2,j}) = h_{conv\,2,3} A_j (T_{2,j} - T_{3,j}) \quad (6)$$

In these equations, the convective heat transfer coefficients of $h_{conv,out}$, $h_{conv,in}$, $h_{conv1,2}$ and $h_{conv2,3}$ are calculated according to the conditions of airflow on the window surfaces. Section 3.2 provided more details on their calculations.

$\dot{Q}_{rad,out}$ and $\dot{Q}_{rad,in}$ are the radiative heat transferred between the window and surrounding surfaces and are obtained by Equations (7) and (8):

$$\dot{Q}_{rad,out} = h_{rad,out} A_j (T_{sky} - T_{1,j}) \quad (7)$$

$$\dot{Q}_{rad,in} = h_{rad,in} A_j (T_{mr} - T_{3,j}) \quad (8)$$

The radiative heat transfer coefficients between surfaces and indoors and outdoors are calculated as follows [9]:

$$h_{rad,in} = \sigma \varepsilon_4 (T_{mr}^2 + T_{3,j}^2)(T_{mr} + T_{3,j}) \quad (9)$$

$$h_{rad,out} = 0.5\sigma\varepsilon_1 (T_{sky}^2 + T_{1,j}^2)(T_{sky} + T_{1,j})(T_{sky} - T_{1,j}) / (T_{1,j} + T_{out}) \quad (10)$$

In Equation (7), T_{sky} is the sky temperature given by [9]:

$$T_{sky} = 0.0552 T_{out}^{1.5} \quad (11)$$

Also, T_{mr} in Equation (8) is the mean radiant temperature of surfaces surrounding the interior glass surface, which is assumed to be equal to indoor air temperature [15].

$\dot{Q}_{rad1}$ and $\dot{Q}_{rad3}$ are the total of the long-wave radiative heat exchanges between channel-side surfaces of the interested glass pane and the channel-side surfaces of the other zones. Their mathematical formulation are as follows:

$$\dot{Q}_{rad,\,1} = \sum_{i=1}^{n} \frac{\sigma A_{1,j}(T_i^4 - T_{1,j}^4)}{\frac{1}{F_{1i}} + \frac{1-\varepsilon_i}{\varepsilon_i} + \frac{1-\varepsilon_1}{\varepsilon_1}} \quad (12)$$

$$\dot{Q}_{rad,\,3} = \sum_{i=1}^{n} \frac{\sigma A_{3,j}(T_i^4 - T_{3,j}^4)}{\frac{1}{F_{3i}} + \frac{1-\varepsilon_i}{\varepsilon_i} + \frac{1-\varepsilon_3}{\varepsilon_3}} \quad (13)$$

where ε is the emissivity of the glass surface. F_{1i} and F_{3i} are the view factors of surfaces exposed to each other.

In Equations (1) and (3), $\dot{Q}_{solar\,1}$ and $\dot{Q}_{solar\,3}$ are the solar radiation absorbed by the outer and inner glass panes, respectively. These terms include two components of direct and diffuse absorption (Equations (14) and (15)).

$$\dot{Q}_{solar\,1} = I_{dir}\alpha_{dir\,1} + I_{diff}\alpha_{diff\,1} \quad (14)$$

$$\dot{Q}_{solar\,3} = I_{dir}\tau_{dir\,1}\alpha_{dir\,3} + I_{diff}\tau_{diff\,1}\alpha_{diff\,3} \quad (15)$$

I_{dir} and I_{diff} are the direct and diffuse solar irradiations obtained from the climate data of the studied location. a_{dir} and a_{diff} are the direct and diffuse absorption coefficients of the glass panes and $\tau_{dir\,1}$ and $\tau_{diff\,1}$ are the direct and diffuse transmission coefficients of the outer glass pane.

$\dot{Q}_{cond}$ is the conductive heat transfer between two adjacent zones of glass pane which is obtained readily from Fourier law in a general form of Equation (16).

$$\dot{Q}_{\text{cond i,j}} = \frac{kA_i(T_i - T_j)}{|y_i - y_j|} \tag{16}$$

$y_i - y_j$ is the distance between two adjacent nodes of the glass pane.

Finally, $\dot{Q}_{\text{flow 2,j}-1}$ is the flow energy change through the zone "j" and is given as follows:

$$\dot{Q}_{\text{flow 2,j}-1} = \dot{m}C_p(T_{2,j} - T_{2,j-1}) \tag{17}$$

C_p is the specific heat capacity of the airflow and $\dot{m}$ is the mass flow rate of channel air. In the forced flow, $\dot{m}$ is determinant, while in natural convection, its magnitude is the function of various parameters such as the temperature of the glass pane and flow and geometric channel dimensions. The present work calculates the mass airflow rate by the airflow network model [16]. This procedure determines the mass flow rate based on pressure distribution along the channel. The details of the airflow network model can be followed in reference [17]. In this situation, the mass and energy balance equations are coupled. First, the mass balance equations are solved using known boundary conditions, and the calculated airflow rates were then inserted into Equations (1) and (3) to calculate the temperatures.

3.2 Convective Heat Transfer Coefficients

The accuracy of zonal model output strongly depends on heat transfer coefficients used in the energy balance equations. Therefore, it is essential to apply proper and validated equations estimating actual thermal and hydrodynamic conditions. As climate conditions such as temperature, velocity, and orientation of wind, and solar radiation are not controllable and change hourly, airflow condition on surfaces of the window is varied and different types of flow (laminar, turbulent, and natural, forced) can occur. Thus, in this work, Nusselt correlations for the possible type of flows were calculated as follows.

Nusselt correlation for the exterior surface of the window, which is valid for both laminar and turbulent regimes, are given as [18].

$$\overline{\text{Nu}}_{mix} = \sqrt[3]{\overline{\text{Nu}}_n^3 + \overline{\text{Nu}}_f^3} \tag{18}$$

In this equation, $\overline{Nu}_n$ is the average Nusselt number of the natural convection and is calculated from the correlation presented by Churchill and Chu [18].

$$\overline{\text{Nu}}_n = \left(0.825 + 0.325\text{Ra}_H{}^{\frac{1}{6}}\right)^2 \tag{19}$$

$\overline{\text{Nu}}_f$ is the averaged Nusselt number of forced convection [19]:

$$\overline{\text{Nu}}_f = \frac{H}{k}(4.7 + 7.6v_s) \tag{20}$$

where v_s is the characteristic velocity of air flowing on the glass surface and is the function of the local wind velocity.

When the surface is windward, v_s can be obtained as [19]:

$$v_s = \begin{cases} 0.5 & ; \quad v \le 2\ \text{m/s} \\ 0.25v & ; \quad v > 2\ \text{m/s} \end{cases} \tag{21}$$

and, for a leeward surface:

$$v_s = 0.3 + 0.05v \tag{22}$$

where v is the local wind velocity obtained by meteorological measurements.

The convective heat transfer on the internal side primarily occurs by natural convection, and rarely by mixed and forced convection. Thus, Nusselt correlation of natural convection for the interior surface can be calculated from Equation (19).

Nusselt correlations used for laminar and turbulent regimes are separated for the channel side of the ventilated window. For laminar flow, the correlation presented by Bar-Cohen [20] was applied (Equation (23)) and for turbulent flow, the correlation suggested by Badr et al. [21] was used (Equation (24)).

$$\overline{Nu}_n = \left(\frac{576}{\text{Ra}_b} + \frac{2.873}{\sqrt{\text{Ra}_b}}\right)^{-0.5} \tag{23}$$

$$\overline{Nu}_n = 0.64\ (\text{Ra}_b)^{0.27} \tag{24}$$

3.3 Calculation of Total Heat Gain

To determine the thermal performance of windows, the total heat gain is an illustrative parameter. The total heat gain for the ventilated window includes three parts (Equation (25)); one part is the heat transfer due to the temperature difference between the interior glass surface and indoors, and one part is the heat transfer due to solar irradiation directly transmitted to the indoor space. The third part appears for the supply mode of the ventilated window in which the outdoor air is preheated while passing through the window channel before entering the room. Equations (26) to (28) define these three parts of the total heat gain. T_{outlet} in Equation (28) is the air temperature exit the channel entering the room.

$$Q_{\text{gain}} = Q_{\text{solar}} + Q_{\Delta\text{T}} + Q_{\text{preheat}} \tag{25}$$

where

$$Q_{\text{solar}} = \int_{t_1}^{t_2} \sum_{j=1}^{n} A_j(I_{\text{dir}}\tau_{\text{dir}} + I_{\text{diff}}\tau_{\text{diff}})_j\, dt \tag{26}$$

$$Q_{\Delta T} = \int_{t_1}^{t_2} \sum_{j=1}^{n} A_j \left(h_{\text{rad,in}} + h_{\text{conv,in}}\right)_j \left(T_{3,j} - T_{\text{in}}\right) dt \tag{27}$$

$$Q_{\text{preheat}} = \int_{t_1}^{t_2} \dot{m} C_p (T_{\text{outlet}} - T_{\text{in}}) dt \tag{28}$$

In the present work, energy saving is the result of the difference between the total heat gain of the equivalent double-glazed window and the ventilated window.

3.4 Requirements for Indoor Air Quality

According to standard ASHRAE 62.1 [4], the design zone outdoor airflow, i.e., the outdoor airflow provided to the zone by the supply air distribution system, shall be determined according to Equation (29).

$$\dot{V}_{oz} = \frac{\dot{V}_{bz}}{E_z} \tag{29}$$

$\dot{V}_{bz}$ is the breathing zone outdoor airflow, which is the design outdoor airflow required in the breathing zone of the occupiable space or spaces in a zone. This parameter is determined as:

$$\dot{V}_{bz} = R_p P_z + R_a A_z \tag{30}$$

where P_z and A_z are zone population and zone floor area. R_P and R_a are outdoor airflow rates required per person and per unit area respectively and determined from tables given in standard ASHRAE 62.1 [4].

E_z is Zone Air Distribution Effectiveness and defined as a measure of how effectively the zone air distribution uses its supply air to maintain the acceptable air quality in the breathing zone. The value of E_z is listed in table 6-2 of standard ASHRAE 62.1 [4].

4. Validation of the Zonal Model

In order to verify the zonal model coupled with airflow network model, the experimental data reported by Chow et al. [14] are used. Chow et al. carried out an experiment of a ventilated window in Hong Kong for three consecutive summer days. The studied window is 1.95 m in height and 0.88 m in width composed of an outer absorptive glass pane and an inner clear glass pane. The cavity thickness is 0.035 m.

Figure 4 compares the calculated and the measured temperatures of the absorptive and clear glass panes. The graphical plots indicate that the zonal model can soundly predict changes in glass temperature. However, it is found under-estimation in all hours of simulation. It is probably due to unpredictable weather conditions and the effect of the window configuration, such as louvers installed for intake and exhaust and the frame. The mean and maximum deviations of the temperatures of absorptive and clear glasses are (0.42 and 2.3) ℃ and (0.23 and 1.2) ℃ , respectively.

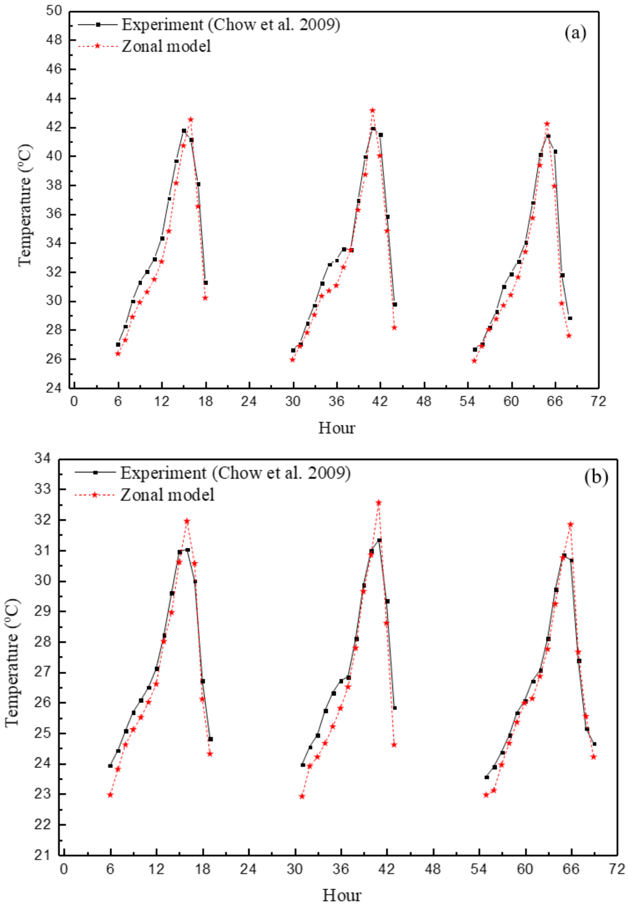

Figure 4. Comparison of glazing surface temperature of the present zonal model with the results reported by Chow et al. [14]: a) absorptive glass pane, b) clear glass pane.

5. Results and Discussion

Consider a 6 m × 5 m ×3 m office room. This room carries a 3 m^2 window at the center of the external wall. According to Iranian national building regulation, this considered area of the window-to-floor ratio of 0.1. This regulation states that rooms should be provided with natural lighting through windows as much as possible. Therefore, the total window area should be less than 10% of the floor area in each space. The window in the current study was taken as facing south to have a maximum solar thermal load. The indoor temperature setpoints are 20 °C in winter (from November to April next year) and 24 °C in summer (from April to October). According to standard ASHRAE 62.1 [4], Rp is 2.5 L/s for each person, and Ra is 0.3 L/sm^2. Considering the occupant density of 5 persons per 100 m^2, the breathing zone outdoor airflow is calculated at 12.75 L/s.

Assuming supply is drawn in on the opposite side of the room from the exhaust, the zone air distribution effectiveness, Ez, is equal to 0.8 for both winter and summer operations. Therefore, in the current study, the overall airflow rate required to meet IAQ is 15.9 L/s. In this work, the WINDOW software (Version 7.2) was used to acquire the glass radiative properties required for the zonal model. Table 1 shows the normal-incident optical characteristics of the window glasses.

Table 1. The normal-incident optical characteristics of the window glasses.

	α	*ρ*	*τ*	*ε*
absorptive glass	0.508	0.054	0.438	0.84
clear glass	0.091	0.075	0.834	0.84

The selected city for simulation was Shiraz, located in the south of Iran. The data library reported by Iran Meteorological Organization was the source of the required weather data, including direct and diffuse solar radiation, dry bulb temperature, and velocity and direction of the wind. Also, calculations were done hourly and under quasi-steady conditions.

5.1 The Thermal Performance of Natural Ventilation via Ventilated Window

As stated in section 2, natural ventilation via the ventilated window is possible for both modes of supply and exhaust. The chosen city to study was Shiraz, the city of Iran with a hot semi-arid climate. As ventilated windows are more efficient in the day than night due to solar radiation and the working time of offices is usually from morning to afternoon, the time interval of the study is limited to 6:00-18:00. The window's aspect ratio is 1.5, and the channel thickness is 10 cm. The simulation was conducted for all hours of the year. The simulation results of monthly energy saved are shown in Figure 5. Blue and red columns, respectively, indicate the results of supply and exhaust modes.

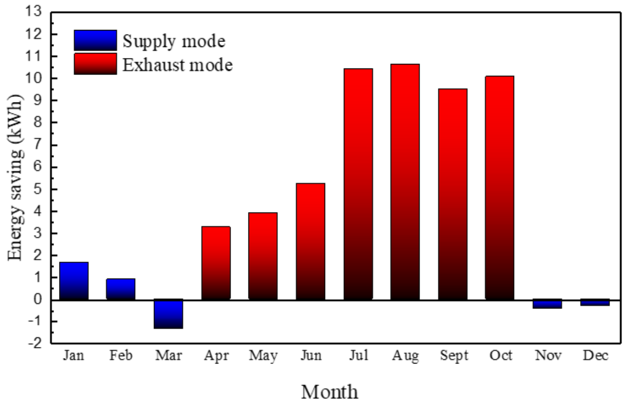

Figure 5. The monthly energy saved for the naturally ventilated window.

As shown in Figure 5, the exhaust mode is considerably more energy-efficient than the supply mode for Shiraz. The energy-saving in the three months of March, November, and December is negative, which is meant that the energy consumption increases in these months. However, this increase in energy consumption is negligible, so that the percentage of increase is less than 2% (see Table 2). In addition, exhaust mode has better performance in three months of summer than in spring. It is due to the higher temperature and solar irradiance in the summer months rather than the spring months. The annual energy saving obtained using the naturally ventilated window is 53.3 kWh.

In addition to energy-saving efficiency, the ventilated window has proved itself a helpful system in providing airflow required for IAQ. Figure 6 shows the airflow rate passed through the ventilated window in the first five days of July. After sunrise, the air starts to flow in the channel, as can be seen. As the hour reaches noon, the flow rate increases and then falls to zero near sunset. Considering

Table 2. The percentage of the monthly energy saving compared to the double-glazing window.

Month	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Energy saving (%)	5.4	1.8	-1.9	4.3	5.6	6.3	9.6	9.7	7.8	7.3	-0.53	-0.63	5.3

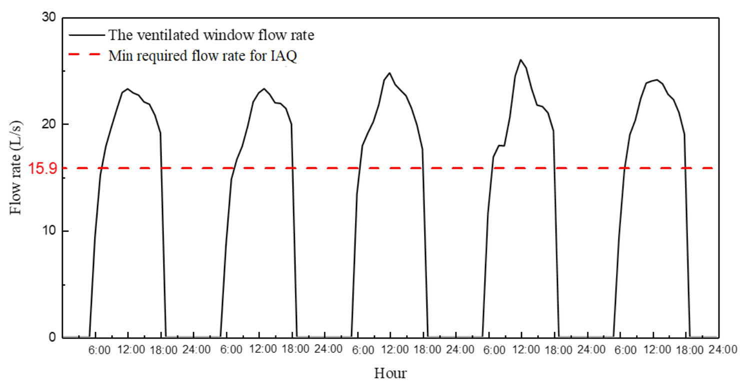

Figure 6. The airflow rate passed through the ventilated window in the first five days of July.

the 15.9 L/s as the minimum airflow rate required to provide fresh air of the office space, the ventilated window can induce the required airflow from morning to afternoon, covering offices' working hours. It should be mentioned that the inlet and outlet of the channel are considered closed during the night, and the window acts as a double-glazed window.

In order to have a better view into the system efficiency in IAQ subject, the histogram of the hours in which the airflow rate is higher than that required during a year is plotted as Figure 7.

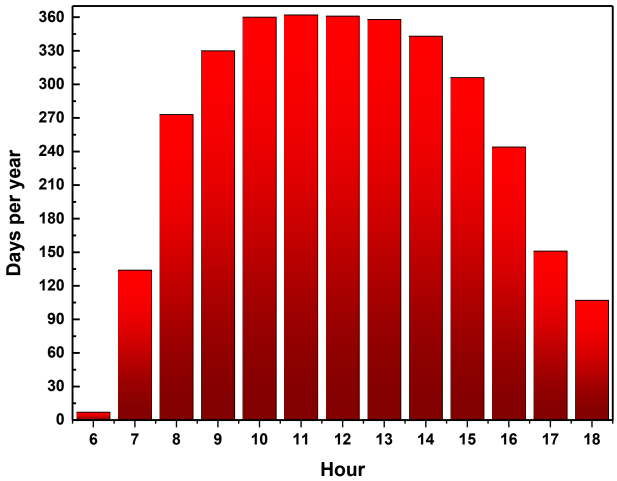

Figure 7. The histogram of the hours when IAQ requirement met during a year.

Midday hour, 12:00, experiences the most days of the year for providing minimum fresh air. It is evident that the solar radiation exposure to the southern window is the maximum around noon and the natural ventilated window has the maximum efficiency in generating the airflow. In addition, there are over 240 days between 8:00 and 16:00 when the generated airflow is higher than that required for IAQ.

5.2 Parametric Study of the Ventilated Window

Geometry has a primary role in the thermal performance of the ventilated window. The proper geometrical design of the ventilated window can efficiently improve its functionality. In the present study, the effect of the geometry characteristics (including the channel thickness, the thickness and aspect ratio of the glass, and the inlet and outlet size) on the thermal performance and IAQ was investigated. The outputs considered for evaluation are:

1) The annual thermal energy saved by the ventilated window is representative of thermal performance.

2) The number of days during the year in which the ventilated window provides the minimum airflow rate during the working time of the office (8:00 -16:00) as representative of IAQ.

Figure 8 shows the effect of the glass aspect ratio on the two above outputs. As can be seen, the thermal energy saved decreases as the aspect ratio increases. Considering the glass area constant, increasing aspect ratio increases the glass height and decreases its width, consequently decreasing the channel Rayleigh number. When the channel Rayleigh number reduces, the channel airflow carries less thermal energy. Thus, the more solar energy absorbed by the inner glass is transferred indoors. Changing the aspect ratio from 0.5 to 2 lessens the energy saving by 13%. A similar trend can be seen for the IAQ index. A higher aspect ratio weakens the buoyancy force to induce airflow to the channel.

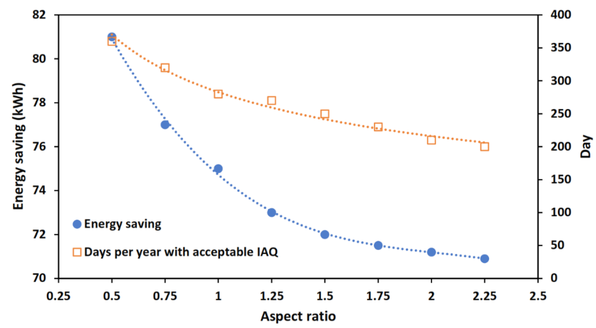

Figure 8. The effect of glass aspect ratio on energy saving and IAQ.

The effect of the channel thickness is described in Figure 9. As shown, increasing the channel thickness results in the increase of all outputs. From the energy and airflow points of view, the channel thickness has the determining effect on generating buoyancy force. It is explainable mathematically in channel Rayleigh number definition where channel thickness appears with fourth power ($Ra_b = \frac{g\beta\Delta T}{\vartheta\alpha}\frac{b^4}{H}$). Since natural convection relies directly on the Rayleigh number, the increase of channel thickness leads to increasing the energy-saving and airflow rate.

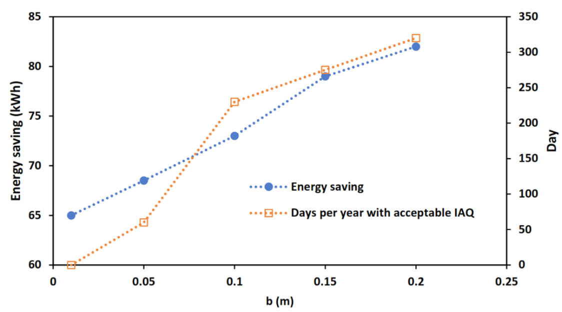

Figure 9. The effect of channel thickness on energy-saving and IAQ.

Inlet and outlet size is another factor influencing the window performance. In the present study, both inlet and outlet sizes are assumed the same. As depicted in Figure 10, energy-saving is augmented as opening height increases. It is expected that the airflow passes more easily through the channel when the local flow resistance of the outlet and inlet is reduced by increasing the inlet and outlet height. Openings act as valves and control the airflow rates. To provide airflow rate required for IAQ, it is necessary to determine a minimum size for openings. As can be seen in Figure 10, the opening size below 5 cm does not meet the minimum airflow rate of 15.9 L/s.

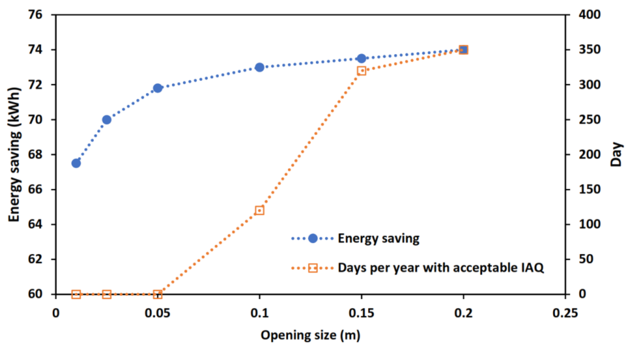

Figure 10. The effect of opening size on energy saving and IAQ.

6. Conclusions

The present work examined the thermal performance of the naturally ventilated window and its capability to provide adequate fresh air flow rate according to indoor air quality standards. Zonal model coupled with airflow network model was used to simulate the ventilated window's thermal performance and airflow rate. The simulation was done for Shiraz, and the monthly energy-saving and the hourly airflow rate induced by the ventilated window were evaluated. It was found that the summer performance of the ventilated window is more efficient than the winter performance. Compared to the double-glazed window, the maximum percentage of energy-saving is obtained in August with 9.7%, while the ventilated window deteriorated in March with 1.9% more heat gain. An annual energy saving of 5.3% is achieved.

The airflow rate induced naturally by the ventilated window changes hourly so that the flow starts with sunrise and increases to its maximum value around noon and then falls as reaching sunset. Considering 15.9 L/s as the minimum airflow rate to meet the IAQ standard of ASHRAE 62.1 and the working time of the office between 8:00 and 16:00, the ventilated window can supply the required fresh air for 259 days of the year.

Furthermore, a parametric study was done to study the effect of the channel thickness, window aspect ratio, and opening size on the thermal performance and index of the number of days with minimum airflow rate covering the office working time. It was found that the increase in the channel thickness and opening ratio leads to the energy-saving and IAQ index whilst the aspect ratio increase causes the energy-saving and IAQ index to reduce.

Declaration of Interests

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Nomenclature

A	Area (m^2)	Subscripts	
C_p	Specific heat (J/kgK)	conv	Convection
E_z	Zone Air distribution effectiveness	b	Channel thickness
F	View factor	diff	Diffuse
h	Convective heat coefficient (W/m^2K)	dir	Direct
I	Solar radiation (W/m^2)	rad	Radiation
k	Conductivity (W/mK)	cond	Conduction
$\dot{m}$	Mass flow rate (kg/s)	H	Channel height
Nu	Nusselt number	in	Indoor
$\dot{Q}$	Heat flux (W)	out	Outdoor
Ra_b	Channel Rayleigh number $\frac{g\beta\Delta T}{\vartheta\alpha}\frac{b^4}{H}$	D_h	Hydraulic diameter
Ra_H	Rayleigh number $\frac{g\beta\Delta TH^3}{\vartheta\alpha}$	mr	Mean radiant
T	Temperature (℃)	f	Forced convection
$\dot{V}$	Volume flow rate (m^3/s)	n	Natural convection
		mix	Mixed convection
Greek symbols		s	Surface
α	Absorption coefficient	bz	Breathing zone
σ	Stefan-Boltzmann's constant	oz	Outdoor zone
ε	Emissivity		
τ	Transmission coefficient		
ρ	Reflection coefficient		

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