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NRCC-41020

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January 1998

A version of this document is published in / Une version de ce document se trouve dans:
ASHRAE Transactions, 104, (pt. 1), pp. 1-10, 1998

SF-98-13-3

Comparison of the Monthly Thermal Performance of a Conventional Window and a Supply-Air Window

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ABSTRACT

A computer program was developed to simulate the performance of a conventional window and a supply-air window in the cold climate of Ottawa, Canada. It was found that the latter window design lead to higher monthly net heat gains, especially during the winter when it is the most beneficial. This increase is due mainly to a reduction in the conductive heat loss rather than an increase in the solar heat gain. The results also support the fact that the supply-air window can be employed continuously to satisfy the ventilation requirement of the space with a small penalty in the cooling load during the summer. The temperature on the room side of the inner pane is only slightly lower in the case of the supply-air window, which indicates that the two window designs studied will result in the same comfort level inside the space.

INTRODUCTION

During the heating season, it is possible to decrease the heating load of the indoor space by employing measures that can reduce the heat loss through the building envelope to the outdoors. In order to achieve this, there has been a considerable effort throughout the years to employ materials with higher thermal resistance in the construction of the exterior walls of building envelopes. In addition, new and more efficient window designs have appeared on the market. For instance, it has been suggested that allowing air to flow between the panes of a multiple-glazing window would improve the thermal performance of the fenestration unit. Such windows are known as airflow windows.

One variation of these windows, referred to as the exhaust-air window, allows for indoor air to flow between the inner two panes of the window. In the cooling season, the airflow helps reduce the cooling load when the heat picked up from the window panes is discharged outside. In the heating season, the heat lost through the outer pane of the window comes mostly from the exhaust airflow, which helps reduce the transmission loss through the fenestration. In addition, the exhaust airflow helps maintain the inner pane temperature close to the room temperature, resulting in better thermal comfort.

The supply-air window allows for outdoor air to flow between the outer two panes of the window and into the building. The airflow helps reduce the heating load when the heat picked up by the air from the window finds its way back inside the space. In addition, the airflow between the panes helps satisfy the outdoor air requirement of the space. In the cooling season, the supply-air window may increase the cooling load when the heat picked up from the hot panes during a sunny summer day is delivered to the space. The cold outdoor airflow between the panes leads to lower inner pane temperature that may lead to reduced thermal comfort. This study was done to compare the thermal performance of a triple-glazed supply-air window to the thermal performance of a conventional triple-glazed window.

Throughout the years, there have been several studies that dealt with the thermal performance of the supply-air window (Barakat 1987; Wright 1986; Yuill 1987a, 1987b). Barakat (1987) performed an experimental study to assess the performance of the supply-air window during the heating season in Ottawa, Ontario, Canada. A triple-glazed window was installed on the south wall of a two-room test unit that was continuously monitored. An identical adjacent unit fitted with a conventional double-glazed window was monitored as a control. It was found that the supply airflow recovered a large fraction of the heat loss, which represented 50% of the energy required to heat the ventilation air. The U-factor of the supply-air window based on the average winter outdoor temperature

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and the total heat loss during this season was found to be 0.5 W/m^2·K (0.088 Btu/h·ft^2·°F). The overall reduction in purchased energy compared to a double-glazed or triple-glazed window was 25% and 20%, respectively. This study was carried for only a period of four months during the heating season, which did not allow for any conclusions to be made about the performance of the window during the cooling season.

Ferguson and Wright (1984) developed the computer program VISION to analyze the heat transfer through singly and multiply glazed windows. Wright (1986) was able to modify this program to simulate the performance of a supply-air window. The author then used the modified version of VISION to derive U-factors and shading coefficients for the center glass region at specific weather conditions for a variety of window designs. The U-factors were obtained when the window was exposed to zero solar radiation at an outdoor temperature of −18 °C (−0.4 °F), an indoor temperature of 21 °C (69.8 °F), and a cloud cover of 0.5. It was reported that a triple-glazed supply-air window had a U-factor of 1.22 W/m^2·K (0.21 Btu/h·ft^2·°F) compared to a value of 1.84 W/m^2·K (0.32 Btu/h·ft^2·°F) for a conventional triple-glazed window. Under the previous conditions, Wright reported that the temperature of the inner pane of the supply-air window was 1 °C (1.8 °F) lower than that of the conventional triple-glazed window. The equivalent shading coefficient of the supply-air window was very close to that of the simple triple-glazed window, equal to 0.78. As indicated earlier, this work studied the performance of the supply-air window under specific weather conditions. No attempt was made to look at how the performance of the supply-air window varied throughout the year.

Yuill (1987a) conducted an extensive study of supply-air windows for the ministry of Energy, Mines, and Resources Canada. The authors performed a full simulation of the airflow of the supply-air window by solving the full Navier-Stokes equations with the appropriate boundary conditions. The flow was assumed to be developing hydrodynamically and thermally. It was indicated that the airflow had to be maintained in the laminar regime, ReD_h < 2000, so that the heat reaching the outer pane was minimized. In order to maintain hydrodynamic stability, it was found that the ratio of the Grashof number to the Reynolds number had to be less than 24. The latter two conditions put an upper limit on the spacing between the panes and a lower limit on the mass flow rate through the supply air channel. The authors indicated that with the required ventilation rates and the window sizes available, it was possible to have a supply-air window with a channel flow that is laminar and hydrodynamically stable.

The energy savings during the heating season obtained through the use of the supply-air window over a conventional triple-glazed window were estimated by Yuill (1987a) for a typical house in Winnipeg. The heat load was predicted using the degree-day method with a supply-air window U-factor that was solved for an outdoor temperature of −18 °C, an indoor temperature of 21 °C, and a cloud cover of 0.5. The authors reported annual reductions of about $CAN 70, based on the Manitoba electricity cost of $CAN 0.03/kWh, in the heating bill using an airflow channel gap of 10.7 mm (0.42 in.). Similar to the study by Wright (1986), this study did not look into the variation of the performance of the supply-air window from one month to the next. Instead, it relied on a simulation of the fenestration heat transfer under specific weather conditions to predict the seasonal performance of the unit.

The present study uses a simulation program similar to the one developed by Wright (1986) and an hourly weather file for the city of Ottawa to study the monthly variation in the thermal performance of the supply-air window. Monthly net heat gains are obtained for a supply-air window with three panes and compared to those for a conventional triple-glazed window.

COMPUTER PROGRAMS

Conventional Triple-Glazed Window

The performance of triple-glazed windows is simulated using a program identical to the VISION program developed by Ferguson and Wright (1984), which is reproduced to perform the present study. The relevant heat transfer variables employed in the problem formulation are shown in Figure 1, which also shows the relevant dimensions of the window employed in the simulation. Every pane is represented by a node i and is characterized by a temperature T_i, a room-side radiosity Jd_i, and a weather-side radiosity Ju_i. For each of the three panes, it is possible to write an energy balance equation given by

$$Ju_{i-1} + Jd_{i+1} - Ju_i - Jd_i + h_{i-1}(T_{i-1} - T_i) + h_i(T_{i+1} - T_i) + S_i = 0 \qquad (1)$$

where h_{i-1} and h_i are the heat transfer coefficients on the room side and the weather side of the pane, respectively, and S_i is

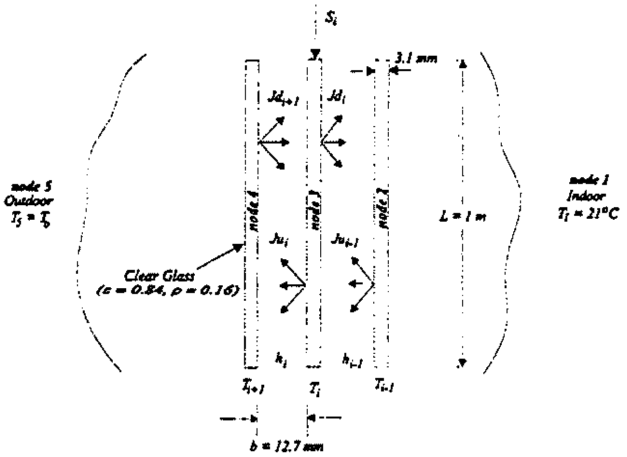

Figure 1 Heat transfer variables for the conventional triple-glazed window.

the solar radiation absorbed by node i. The total longwave radiosity of node i on the room side and the weather side, respectively, are given by

$$Jd_i = \varepsilon\sigma T_i^4 + (1 - \varepsilon - \tau)Ju_{i-1} + \tau Jd_{i+1}, \qquad (2)$$

$$Ju_i = \varepsilon\sigma T_i^4 + (1 - \varepsilon - \tau)Jd_{i+1} + \tau Ju_{i-1}, \qquad (3)$$

where ε and τ are the longwave emissivity and transmissivity of glass. Equations 1, 2, and 3, combined with correlations for the heat transfer coefficients, can be solved using an iterative procedure for specific indoor and outdoor conditions. Initial estimates for the node temperatures are chosen between the indoor and the outdoor temperatures. It was found that the procedure converges within few iterations.

The net heat gain, *NHG*, accounting for conductive gains or losses and solar gains through the window is evaluated, in W/m^2, for every hour of the year based on the following expression:

$$NGH = Jd_2 - Ju_1 + h_i(T_2 - T_1) + S_1 \qquad (4)$$

The monthly net heat gain, in MJ/m^2, is obtained by summing all the hourly contributions given by Equation 1.

Supply-Air Window

The computer program described earlier for the conventional-triple-glazed window is modified in order to accommodate the airflow through the outer channel. Figure 2 shows the relevant heat transfer variables in the case of the supply-air window along with the dimensions employed in the simulation. In order to account for the heat transfer within the airflow channel, different forms of the energy balance equation are used for the middle and outer panes represented by nodes 3 and 4. For the middle pane, node 3, the energy equation becomes

$$Jd_4 + Ju_2 - Jd_3 - Ju_3 + h_2(T_2 - T_3) - \bar{q}_a + S_3 = 0 \qquad (5)$$

and for the outer pane we have

$$Jd_5 + Ju_3 - Jd_4 - Ju_4 + h_4(T_o - T_4) - \bar{q}_b + S_4 = 0 \qquad (6)$$

where q_a is the heat transferred from the middle pane to the airstream and q_b is the heat transferred from the airstream to the outer pane.

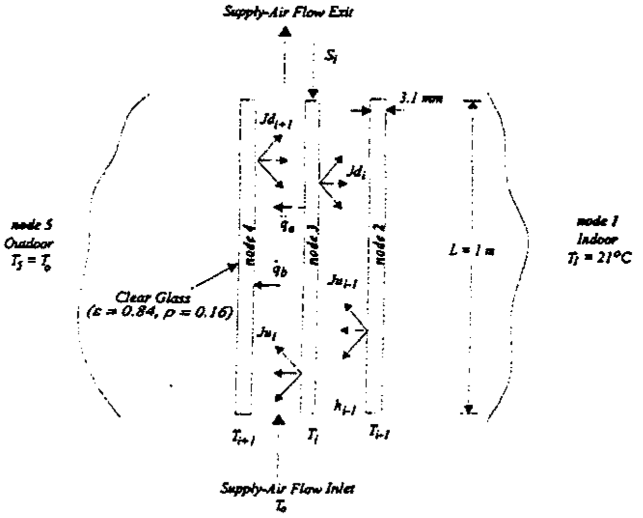

Figure 2 Heat transfer variables for the triple-glazed supply-air window.

The supply airflow is modeled based on the work of Hatton and Turton (1962). It is assumed that the flow is laminar and hydrodynamically fully developed with unequal wall temperatures. The authors provide expressions for the local Nusselt number along the length of the channel. These expressions are integrated to derive expressions for the average heat flux on the room-side and the weather-side walls of the channel. As a result, the following equations are obtained for q_a and q_b:

$$\bar{q}_a = \frac{2K(T_3 - T_{wm})}{bL} \left[\begin{array}{c} x + 2k\sum_1^{\infty} \dfrac{3(\partial Y_{En}/\partial y)_{y=1}}{8\lambda_{En}^3(\partial Y_{En}/\partial\lambda_{En})_{y=1}} \\ \exp(-8\lambda_{En}^2 x/3) - \\ 2\sum_1^{\infty} \dfrac{3(\partial Y_{On}/\partial y)_{y=1}}{8\lambda_{On}^3(\partial Y_{On}/\partial\lambda_{On})_{y=1}} \\ \exp(-8\lambda_{On}^2 x/3) \end{array} \right]_0^L \qquad (7)$$

$$\bar{q}_b = \frac{2K(T_3 - T_{wm})}{bL} \left[\begin{array}{c} x - 2k\sum_1^{\infty} \dfrac{3(\partial Y_{En}/\partial y)_{y=1}}{8\lambda_{En}^3(\partial Y_{En}/\partial\lambda_{En})_{y=1}} \\ \exp(-8\lambda_{En}^2 x/3) - \\ 2\sum_1^{\infty} \dfrac{3(\partial Y_{On}/\partial y)_{y=1}}{8\lambda_{On}^3(\partial Y_{On}/\partial\lambda_{On})_{y=1}} \\ \exp(-8\lambda_{On}^2 x/3) \end{array} \right]_0^L \qquad (8)$$

where

$$T_{wm} = \frac{T_3 + T_4}{2} \qquad (9)$$

$$K = \frac{T_o - T_{wm}}{T_3 - T_{wm}} \qquad (10)$$

and b is the spacing between the panes of the channel, L is the height of the window, and k is the thermal conductivity of air. The eigenvalues, λ_{En} and λ_{On}, and the derivatives appearing in Equations 7 and 8 are provided by Hatton and Turton (1962).

The energy balance for the inner pane is identical to the one employed in the case of the conventional window given by Equation 1. The longwave radiosity on the room side and the weather side of each of the panes are still given by Equations 2 and 3. These equations, along with Equations 5 through 10, are combined with correlations for the heat transfer coefficient on the room side, inside the sealed cavity, and on the weather side of the window and then solved using an iterative technique based on Newton's method (Burden and Faires 1985). These correlations are obtained from the VISION3 reference manual (UW 1992). Initial estimates for the node temperatures are again chosen between the outdoor and the indoor temperatures. The program is found to converge easily within few iterations.

In the case of the supply-air window, the expression for *NHG*, in W/m^2, is modified to account for the contribution of the airflow and it is given by

$$NGH = Jd_2 - Ju_1 + h_1(T_2 - T_1) + \bar{q}_a - \bar{q}_b + S_1 \tag{11}$$

where q_a is positive when gained by the airstream and q_b is positive when gained by the outer pane, which is in agreement with the model formulated by Hatton and Turton (1962). The hourly net heat gains are summed up with the appropriate conversions to obtain the monthly net heat gains in MJ/m^2.

The present analysis is based on the assumption that the supply-air window is employed to satisfy all the ventilation requirement of the space. It can be shown that the mass flow rate per unit projected area of the window to satisfy this ventilation requirement is given by

$$\dot{m} = \rho H \frac{A_f}{A_p} ACR \tag{12}$$

where ρ is the density of air, H is the floor to ceiling height of the indoor space, A_f is the floor area, A_p is the projected window area, and ACR is the air change rate. It also can be shown that the Reynolds number based on the hydraulic diameter is given by

$$\mathrm{Re}_{Dh} = \frac{2\dot{m}L}{\mu} \tag{13}$$

where $\dot{m}$ is the mass flow rate and μ is the dynamic viscosity of air. Using the following values for H, A_f/A_p, and ACR:

$$H = 2.44 \text{ m (8 ft)}$$

$$\frac{A_f}{A_p} = 10$$

$$ACR = 0.5 \text{ air changes per hour}$$

The values of the Reynolds number and the mass flow rate are found to be 525 and 0.00433 kg/s (34.3 lbm/h) per unit aperture area of the window, respectively.

RESULTS AND DISCUSSION

Using the computer programs described above, along with an hourly weather file for the city of Ottawa, the monthly net heat gains, *NHG*, are determined for windows facing east, west, north, and south. A positive *NHG* indicates that the window is a net heat gainer, and a negative *NHG* indicates that it is a net heat loser. The contribution of the solar gain and the conductive loss or gain toward the net heat gain is obtained for the north- and the south-facing windows. The results also allow for an assessment of the effect of employing the supply-air window concept on the inside temperature of the inner pane during the heating and the cooling seasons.

Effect of the Window Orientation on the Monthly Heat Gain

Figure 3 shows the variation of the monthly heat gain for the conventional triple-glazed window for four different orientations. As expected, the north-facing window, which receives the least amount of beam solar radiation, has the lowest monthly *NHG* throughout the whole year. This window is a net heat loser from October to March and a net heat gainer during the rest of the year. The east- and the west-facing windows, with very comparable monthly heat gains, are net heat losers only from November to February. During the summer months, the latter two windows have net heat gains that exceed those of the south-facing window, which is a net heat gainer throughout the whole year. These findings are in agreement with the results of a study performed by Barakat (1980).

Figure 4 contains the same information as that contained in Figure 3 for the supply-air window. The variations of the *NHG* for the different window orientations have the same shape here as in the case of the conventional window, with the exception that the *NHG*s for the supply-air window are systematically higher than those shown in Figure 3. In order to allow for a better comparison of the monthly *NHG* of the conventional triple-glazed window and of the supply-air window, Figures 5 and 6 have been constructed to show the *NHG* values for each orientation together on the same graph.

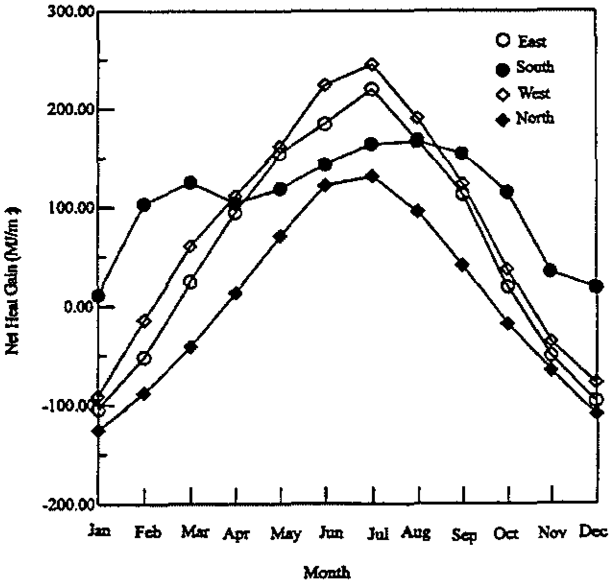

Figure 3 *Effect of window orientation on the monthly net heat gain for the conventional triple-glazed window.*

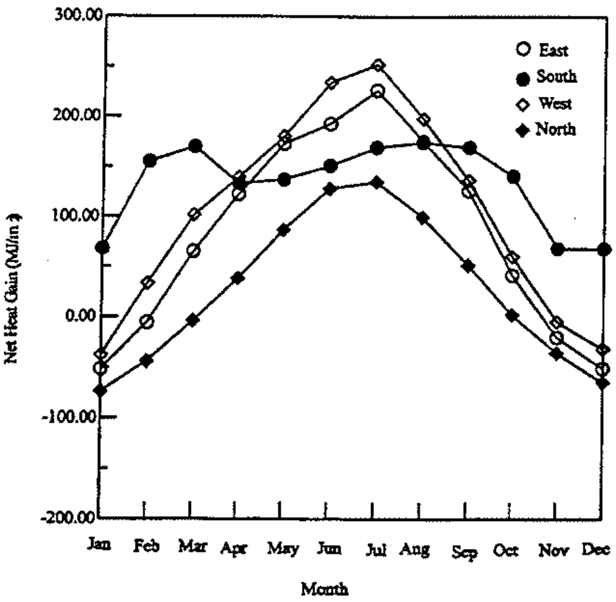

Figure 4 *Effect of window orientation on the monthly net heat gain for the supply-air window.*

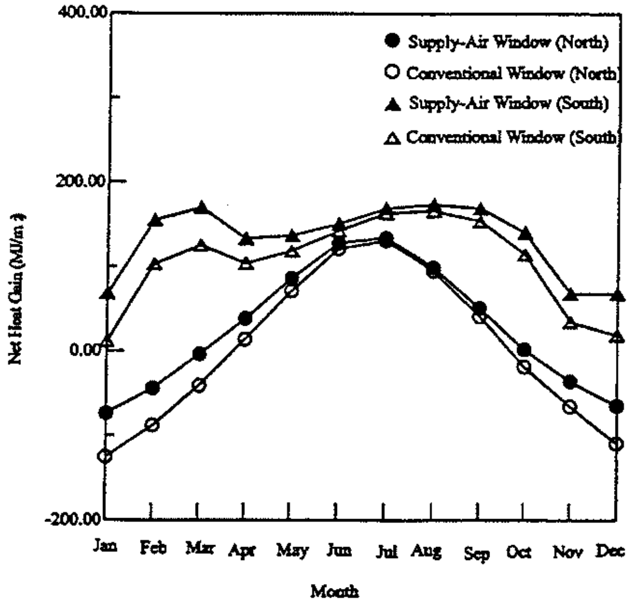

Figure 5 *Comparison of the monthly net heat gain of the conventional window and the supply-air window (north and south orientations).*

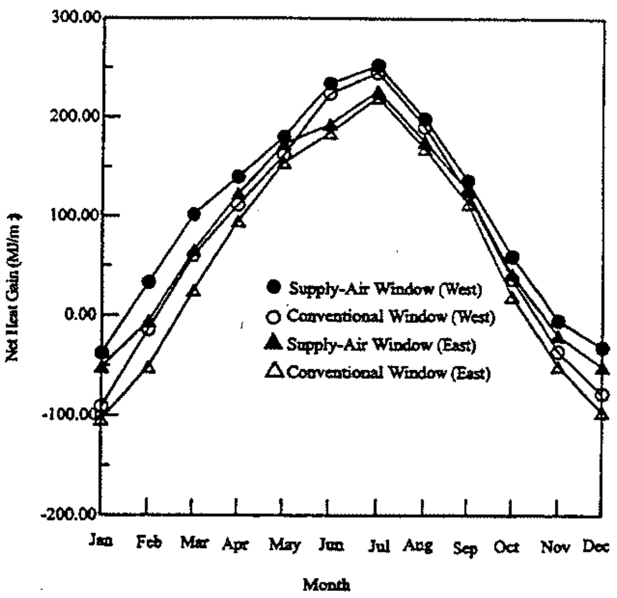

Figure 6 *Comparison of the monthly net heat gain of the conventional window and the supply-air window (east and west orientations).*

Both Figures 5 and 6 show that the difference in the *NHG* values between the conventional window and the supply-air window is the highest during the winter months of December and January and it decreases until it reaches a minimum during the summer months of June and July. For the north-facing window, the increase in the *NHG* varies from 51 MJ/m^2 (4.59 kBtu/ft^2) in January to 3 MJ/m^2 (0.26 kBtu/ft^2) in July, whereas for the south-facing window, it is 57 MJ/m^2 (5 kBtu/ft^2) in January and 6 MJ/m^2 (0.53 kBtu/ft^2) in July. These results are very important because they indicate that the supply-air window can lead to significant increases in the heat gain during the heating season when it is needed the most, while satisfying the ventilation requirement of the space. In addition, during the summer months, it appears that the supply-air window can still be used to satisfy the outdoor air requirement with a small penalty in the cooling load. An accurate assessment of the effect of this on the annual energy consumption of a certain building is only possible through a detailed energy analysis, which must take into consideration all the factors that contribute to the energy balance of the space.

The net heat gain varies continuously throughout the whole day. Figures 7 and 8 show an example of such a variation for a north- and a south-facing window, respectively, on a clear day in January. The *NHG* is the lowest at night when the solar radiation is absent. As the available solar radiation starts to increase in the morning, so does the *NHG* until it reaches a maximum around noon. Then it starts decreasing until it reaches a minimum when the available solar radiation is zero again. The *NHG* for the supply-air window is higher than that for the conventional triple-glazed window throughout the whole day for both window orientations.

For the south-facing window, the difference between the net heat gains of the two window designs is about 24 W/m^2 (7.6 Btu/h·ft^2) during the early morning hours. As the available solar radiation starts increasing, so does the latter difference until it reaches a maximum of about 45 W/m^2 (14.26 Btu/h·ft^2) around noon. This increase indicates that the supply-air window is capable of recovering a portion of the solar energy absorbed by the panes and delivers it back to the space. This recovered energy would have been lost to the surroundings by

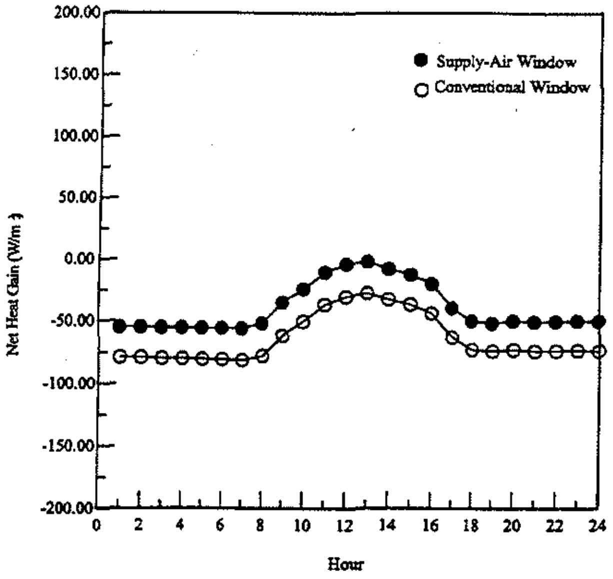

Figure 7 *Hourly variation of the net heat gain for a north-facing window during a clear day in January.*

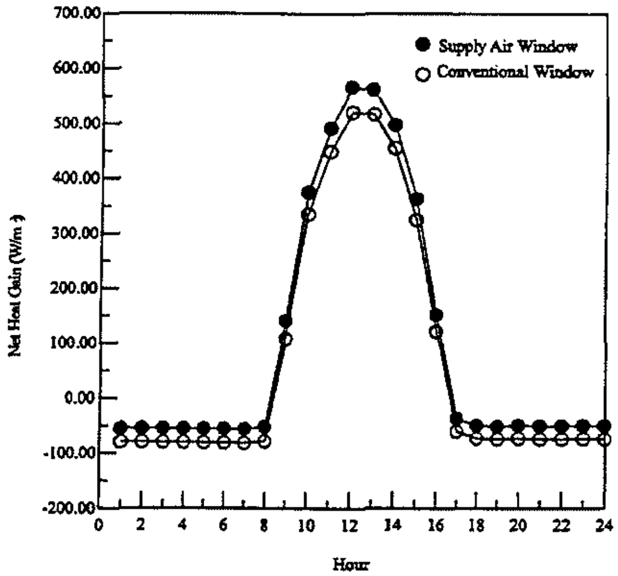

Figure 8 *Hourly variation of the net heat gain for a south-facing window during a clear day in January.*

radiation and convection in the case of the simple triple-glazed window. The contribution of the solar gain is much smaller for a north-facing than for a south-facing window. As a result, the difference between the net heat gains of the two window designs facing north is close to 24 W/m^2 (7.6 Btu/h.ft^2) throughout the whole day.

It is found that the net heat gain during the heating season, October through April inclusive, for a south-facing window is 511.8 MJ/m^2 and 802.7 MJ/m^2 (45 kBtu/ft^2 and 70.5 kBtu/ft^2) for the simple triple-glazed window and for the supply-air window, respectively. Therefore, the airflow window results in an increase in the heat gain by as much as 56% when compared to the conventional triple-glazed window. This extra heat gain through the window will not all show up as a decrease in the purchased heating energy of the space. Part of this energy is absorbed by the mass of the structure and then lost to the surroundings by convection and radiation, particularly if it leads to overheating.

Contributions of Solar Radiation and Conduction

The net heat gain accounts for the solar radiation that finds its way to the interior of the space and the conductive heat loss or gain. The programs described earlier are used along with the Ottawa weather file to assess the contribution of each of these components toward the monthly net heat gains. The conductive heat gain or loss is obtained by setting the solar radiation equal to zero in the weather file. The results are shown in Figure 9 for the supply-air window and the conventional triple-glazed window. This figure is applicable to any window orientation because it is obtained in the absence of any solar radiation.

The results indicate, for the location in question, that conduction leads to a net monthly heat loss throughout the whole year. As expected, this loss is maximum during the winter months when the outdoor temperature is the lowest, and it is the lowest in the summer months when the latter temperature is the highest. The supply-air window results in a decrease in the conductive loss for most of the year. This decrease is the highest during the coldest months reaching 50 MJ/m^2 (4.4 kBtu/ft^2) in January, which amounts to a 31% reduction in the

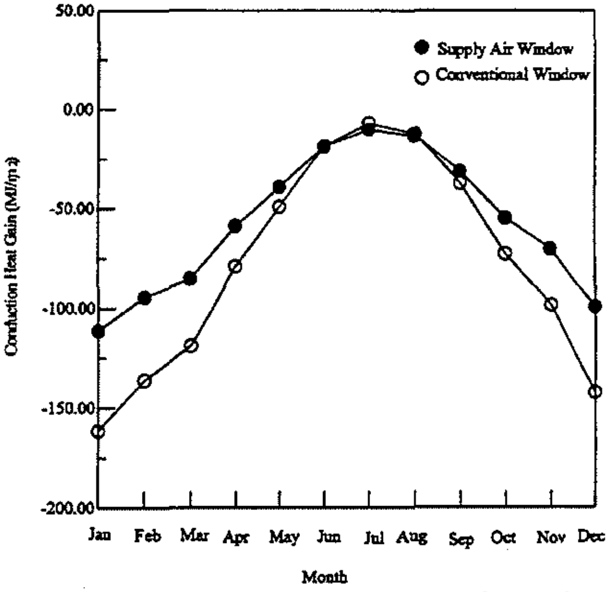

Figure 9 *Comparison of the monthly conductive heat gain for the conventional window and the supply-air window.*

conductive loss of a conventional triple-glazed window. During the months of June, July, and August the conductive loss of the supply-air window and that of the conventional triple-glazed window are both very close and small in magnitude.

Assuming that the heating season in Ottawa stretches from October to April, it is possible to find the net conductive heat loss during this period using the results shown in Figure 9. For the conventional triple-glazed window, this conductive heat loss is equal to 805 MJ/m^2 (70.8 kBtu/ft^2). Using the average outdoor temperature from the weather file during this period, −2.3 °C (27.8 °F), and the latter seasonal heat loss, it is possible to evaluate a U-factor of 1.88 W/m^2·K (0.33 Btu/h·ft^2·°F) for the simple triple-glazed window. In the case of the supply-air window, the total conductive heat loss during the heating season is only 571 MJ/m^2 (50.2 kBtu/ft^2), which results in a U-factor of 1.3 W/m^2·K (0.22 Btu/h·ft^2·°F) based on the same seasonal average outdoor temperature. These values are very similar to those reported by Wright (1986) who simulated a conventional window and a supply-air window with characteristics very similar to the ones described in this study. On the other hand, Barakat (1987) reported an experimental U-factor of only 0.5 W/m^2·K (0.088 Btu/h·ft^2·°F) for a supply-air window with an airflow channel with a pane spacing of 66 mm (2.6 in.). This U-factor was based on the average winter outdoor temperature for the city of Ottawa and the total heat loss through the window during this season. The larger pane spacing used in the latter study makes it less likely for the heat picked up by the thermal boundary layer, developing on the room side of the channel, to reach the outer pane of the supply-air window. As a result, less heat is lost to the outside, which explains the lower U-factor obtained by Barakat.

The contribution of solar radiation to the monthly net heat gain is simply the difference between the latter and the monthly conductive losses or gains. Figure 10 shows the variation of the monthly solar heat gain for a north- and a south-facing window. Contrary to the conduction effect, the difference in the solar heat gain between the supply-air window and the conventional triple-glazed window is relatively constant throughout the year for both window orientations. This difference varies from 1.4 MJ/m^2 to 6.6 MJ/m^2 (0.12 kBtu/ft to 0.58 kBtu/ft^2) and from 5.5 MJ/m^2 to 10.7 MJ/m^2 (0.48 kBtu/ft to 0.94 kBtu/ft^2) for the north-facing and the south-facing windows, respectively. The increase in the solar heat gain in the case of the supply-air window is due to the recovery by the airflow of some of the solar energy absorbed by the panes that would have otherwise been lost to the outdoors by radiation and convection. This absorbed solar radiation is higher for a south-facing window than for a north-facing window, which explains the larger increase in the monthly solar gain associated with a south-facing supply-air window.

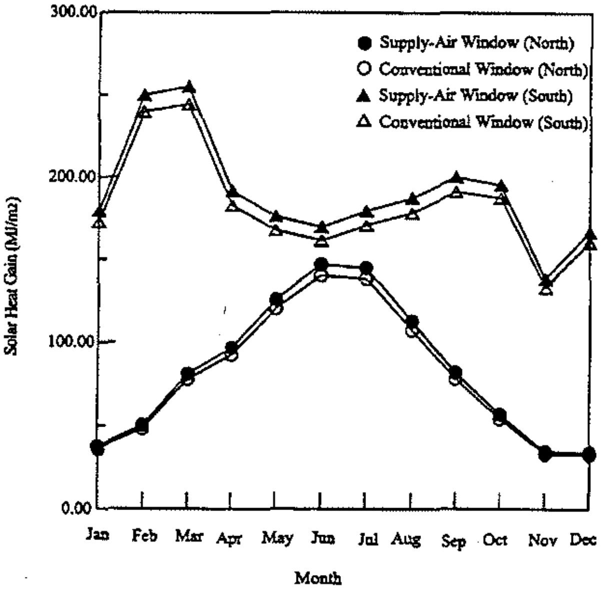

Figure 10 *Variation of the monthly solar heat gain for north- and south-facing windows.*

The results contained in Figure 10 indicate that the solar gains of the two types of windows are very comparable throughout the year. In fact, the ratio of the solar gain of the conventional triple-glazed window to that of the supply-air window varies from 0.95 to 0.96 for both window orientations. Therefore, the increase in the net heat gain associated with the supply-air window is attributed mainly to the reduction in the conductive losses rather than an increase in the solar gains.

Room-Side Temperature of the Inner Pane

In the case of a fenestration system with a low thermal resistance, the air temperature close to the window, in the cold season, can be substantially lower than that of the air in the middle of the room. Both the cold draft that this creates and the increase in the heat loss from the human body to the window by radiation add to the level of discomfort inside the space. During the summer months, this problem is not as severe due to the lower maximum temperature difference between the indoors and the outdoors. The difference between the simple triple-glazed window and the supply-air window, as far as maintaining acceptable comfort levels inside the space, is assessed by comparing their temperatures on the room side of the inner pane for a north- and a south-facing window during a clear winter and summer day in Ottawa.

During the winter, the supply-air window allows cold outdoor air to come in contact with the middle pane. Therefore, we expect the temperature on the room side of the inner pane to be colder in the case of the supply-air window than in the case of the conventional triple-glazed window. Figure 11 confirms this fact for a north- and a south-facing window during a clear day in January. For both windows, the pane temperature is lower at night than during the day when the window is heated by the available solar radiation. The south-facing window receives a greater amount of beam radiation than the north-facing window, which explains the higher jump

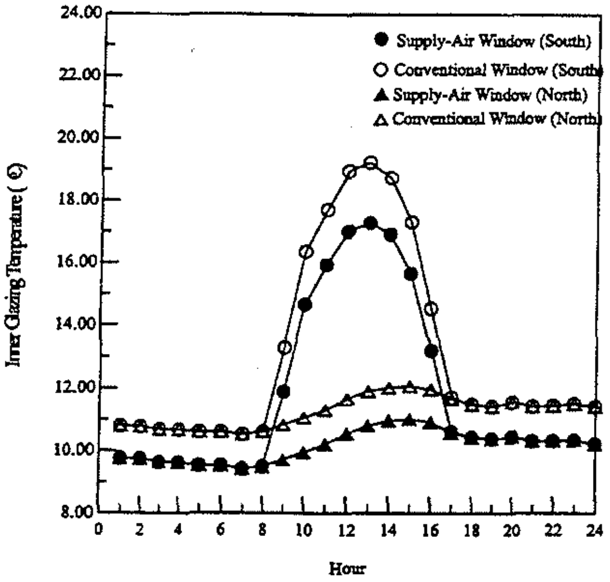

Figure 11 *Hourly variation of the inner glazing room-side temperature of a north- and a south-facing window on a clear day in January.*

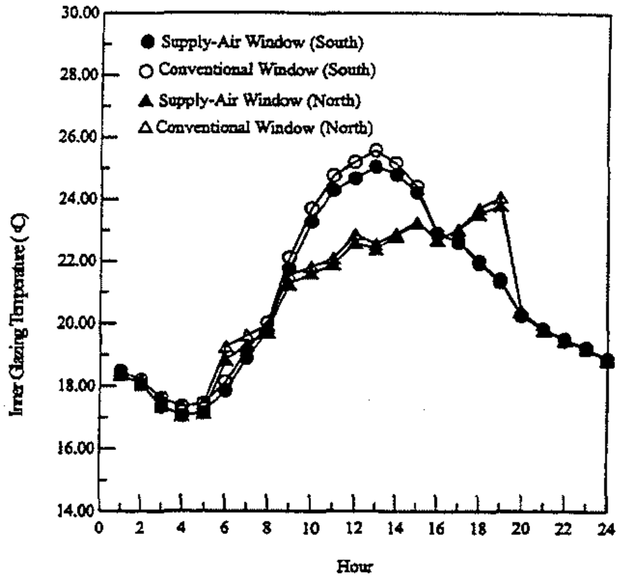

Figure 12 *Hourly variation of the inner glazing room-side temperature of a north- and a south-facing window on a clear day in June.*

in the pane temperature during the day associated with this window orientation. Throughout the whole day, the supply-air window results in a pane temperature that is from 1 °C to 2 °C (1.8 °F to 3.6 °F) less than that in the case of the conventional triple-glazed window. Therefore, the airflow window increases substantially the heat gain during the winter months with a very small penalty in the thermal comfort inside the space.

Elmahdy (1990) introduced the concept of the temperature index (TI) as a means of assessing the condensation potential of a fenestration system during the cold season. This index is the ratio of the difference between the temperature of the room side of the inner pane and that of the outdoors to the difference between the room-side and the weather-side air temperatures. Based on the fact that the room-side temperature of the inner pane of the supply-air window is about 1 °C less than that of the conventional window, the results presented by Elmahdy indicate that the TI of the former window design is only about 2% less than that of the latter. Therefore, under the same climatic conditions, the two fenestration systems studied would have about the same condensation potential.

In the summer, the temperature difference between the outdoors and the indoors is much smaller. Therefore, we expect the difference in the room-side pane temperature of the two window designs to be less in the summer than in the winter. This is confirmed by the results in Figure 12, which shows the variation of the pane temperature for a north- and a south-facing window during a clear day in June. Throughout the whole day, the difference in the pane temperatures of the two types of windows is about 0.2 °C (0.36 °F). The results also indicate that the pane temperature is the smallest at night and reaches a peak during the day when the outdoor temperature and the available solar radiation are the highest.

CONCLUSIONS

Based on the results obtained during the present study, it is possible to draw the following conclusions concerning the performance of the triple-glazed supply-air window and that of the conventional triple-glazed window:

1. The monthly net heat gain of the triple-glazed supply-air window is always higher than that of the conventional triple-glazed window. This increase in the *NHG* is the highest during the coldest months, December and January, and it decreases as we approach the months of June and July.
2. The supply-air window can be used during the heating season to satisfy the ventilation requirements and to decrease the space heating load. During the summer, it appears that the supply-air window can still be used to satisfy the fresh air requirement with a small penalty in the cooling load.
3. For a south-facing window, the difference between the hourly net heat gain of the supply-air window and that of the conventional triple-glazed window increases with an increase in the available solar radiation. The supply-air window enables the recovery of some of the solar energy absorbed by the panes that would have otherwise been lost to the outside in the case of the conventional triple-glazed window.
4. The supply-air window has a lower conductive loss than the conventional triple-glazed window. This difference is the largest during the coldest months of the year when the conductive losses are the highest.

5. The solar heat gain associated with the supply-air window is only slightly higher than that associated with the conventional triple-glazed window. The airflow through the outer panes helps recover some of the solar radiation absorbed by the panes that would have otherwise been lost to the outside by radiation and convection. As a result, this increase in the solar gain is higher for a south-facing window than for a north-facing window.
6. The combined effect of conclusions four and five is that the increase in the net heat gain associated with the supply-air window is due mainly to a reduction in the conductive heat loss rather than an increase in the solar gain.
7. The temperature of the inner pane of the supply-air window is slightly lower than that of the conventional window. Therefore, regardless of the season, both window designs should result in a similar comfort level within the indoor space.

NOMENCLATURE

A_p = projected area of the window (m^2)
ACR = air change rate
A_f = floor area of indoor space (m^2)
b = gap spacing inside the airflow channel (mm or in.)
h = heat transfer coefficient ($W/m^2 \cdot K$ or $Btu/h \cdot ft^2 \cdot {}^\circ F$)
H = floor to ceiling height of indoor space (m or ft)
Jd = room-side longwave radiosity (W/m^2 or $Btu/h \cdot ft^2$)
Ju = weather-side longwave radiosity (W/m^2 or $Btu/h \cdot ft^2$)
k = thermal conductivity of air ($W/m \cdot K$ or $Btu/h \cdot ft \cdot {}^\circ F$)
L = window height (m or ft)
$\dot{m}$ = mass flow rate inside the airflow channel (kg/s or lbm/h)
NHG = net heat gain (MJ/m^2 or $kBtu/ft^2$)
q_a = heat flux on the room-side wall of the airflow channel (W/m^2 or $Btu/h \cdot ft^2$)
q_b = heat flux on the weather-side wall of the airflow channel (W/m^2 or $Btu/h \cdot ft^2$)
$\mathrm{Re}D_h$ = Reynolds number
S = solar radiation absorbed by the node (W/m^2 or $Btu/h \cdot ft^2$)
T = temperature (°C or °F)
T_o = outdoor temperature (°C or °F)

Greek Symbols

ε = longwave emissivity of glass
μ = dynamic viscosity of air ($N \cdot s/m^2$ or $lbf \cdot h/ft^2$)
ρ = density of air (kg/m^3 or lbm/ft^3)
σ = Boltzmann constant ($W/m^2 \cdot K4$ or $Btu/h \cdot ft^2 \cdot {}^\circ F4$)
τ = longwave transmissivity of glass

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