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Improving Energy Savings of a Library Building through Mixed Mode Hybrid Ventilation †

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Abstract: In Canada both residential and commercial buildings often require intensive ventilation and air-conditioning to maintain occupant's thermal comfort and indoor air quality during the operational hours in cooling season. One way to reduce the cooling load consumption is utilizing mixed-mode cooling approach for space conditioning through natural ventilation. This paper presents the potential of control strategies for motorized window opening schedules to reduce the cooling load for a library building that was designed to be net-zero in terms of annual energy consumption (Canada's first institutional net-zero energy building in Varennes (near Montreal), Québec, Canada). Even though the building is located in cold climate zone, the performance study shows that the building is cooling dominated i.e., it has more cooling load than heating load. To achieve net-zero energy building status, the potential of mixed-mode cooling approach (natural ventilation combined with mechanical ventilation) during the cooling season needs to be investigated. Preliminary simulation results show that the mixed-mode cooling could achieve 10 to 20% energy savings based on hybrid ventilation (HV) with fixed schedules, whereas 65% savings based on HV with variable schedules. This paper also shows 47% reduction of cooling load could be achieved by applying shading control strategies over without shading control strategies.

Keywords: energy savings; mixed-mode cooling; natural ventilation (NV); hybrid ventilation (HV); thermal comfort

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1. Introduction

Mixed-mode (MM) cooling refers to a hybrid approach for space conditioning that utilizes both free cooling from natural ventilation and mechanical cooling from heating, ventilation and air-conditioning (HVAC) systems to reduce building energy use and maintain occupant's thermal comfort [1]. Natural ventilation, with favorable weather conditions, such as temperature, wind speed, and humidity, can reduce cooling load and can shift the peak energy demand [1,2]. In most MM building control strategies, hybrid ventilation (HV) with fixed schedules are employed to reach the desired zone set-point, where local climate and specific building features such as thermal mass, façade orientation, window to wall ratio are not considered and thus HV with fixed schedules may lead to increased cooling load or occupant discomfort. In comparison, hybrid ventilation (HV) with variable schedules based on local weather forecasts and occupancy patterns can reduce cooling load and occupant discomfort within future time horizon [3].

The investigated building (Canada's first institutional net-zero energy building (NZEB) in Varennes) produces electricity from a 110.5kWp building integrated photovoltaic with thermal recovery (BIPV/T) system, also has a geothermal heat pump system and significant indoor concrete thermal mass [4]. The early performance of the building shows that there is 33% difference between energy consumption (163 MWh per year) and energy production (110 MWh per year) [4]. Since the investigated building is cooling dominated, one of the recommendations to make the building net-zero energy is utilizing longer period of natural ventilation in cooling season and possibly open the motorized windows at a lower outdoor temperature.

The goal of this study is to develop hybrid ventilation control strategies to reduce the cooling energy consumption while maintaining occupant comfort.

2. Methodology

A thermal network model is developed in MathCAD for this study [5]. The simulated zone (shown in Figure 1) is excerpt from a real library building which is located in Varennes, Quebec. The zone area is approximately 590 m^2 with 5 inch concrete radiant slab. To condition the space, radiant slab for both heating and cooling is used in south, east and west perimeter zone of the building and an underfloor air distribution (UFAD) system is applied for the rest of the area. South façades are double glazed, low-e, argon-filled windows to increase passive solar gain and all other façades are triple glazed, low-e, argon-filled to minimize thermal losses. Window-to-wall ratios (WWR) in south, north, east, and west sides are 30, 10, 20, and 30 percent, respectively.

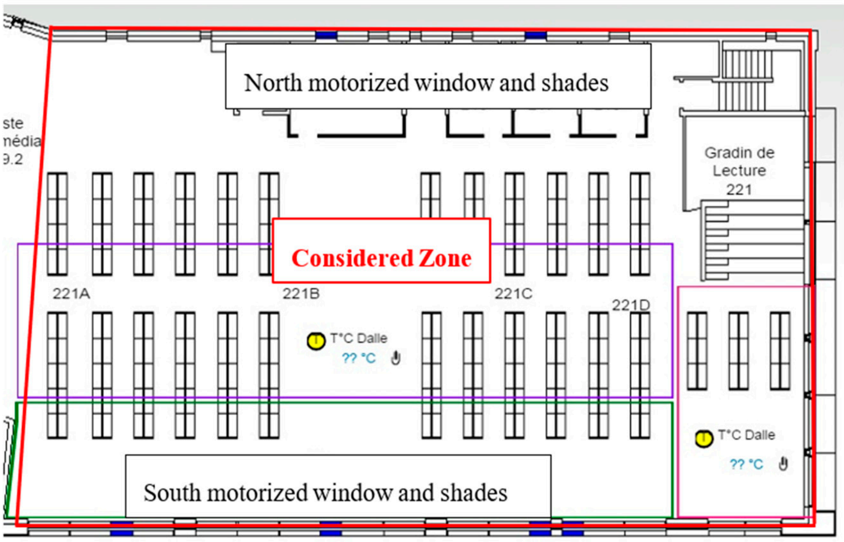

Figure 1. Considered zone for simulation.

Thermal Network

The thermal network model in Figure 2 is developed based on energy balance equations and solved by explicit finite difference method. In detailed thermal network model each surface is modeled separately, whereas in simplified thermal network model, surfaces with high thermal mass (such as concrete radiant floor) are modeled individually and surfaces with low thermal mass (such as gypsum wall surfaces) can be combined together. The simplified thermal network model can estimate the outputs with acceptable precision [6], which is adopted in this simulation. The radiant floor is discretized into two control volumes (node 3 and 4) and walls and roof are represented by one control volume (node 5). Both auxiliary heating and cooling (i.e., radiant floor and UFAD) systems are controlled through set-point temperatures and shown at node 4 and node 1, respectively. The set-point temperature of the radiant floor is maintained well above the dew point temperature to avoid condensation of the floor. Table 1 describes the variables used in Figure 2.

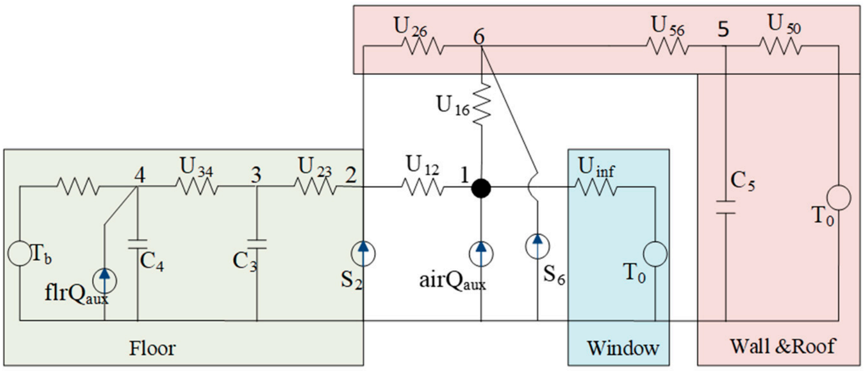

Figure 2. Thermal network model for simulation.

Table 1. Description for thermal network model.

No.	Description
Node #1	Room Air
Node #2	Interior surface of the radiant floor
Nodes #3 & #4	Inside radiant floor (discretized into two control volumes, C_3 and C_4)
Node #5	Inside walls and roof (considered as one control volume, C_5)
Node #6	Node at the interior surface of walls and roof (considered as one node)
U12, U16	Convective conductance between air node and interior surfaces
U26	Radiative conductance between interior surfaces
U23, U34, U56	Conductive conductance of surfaces due to thermal mass
Uinf	Conductance due to infiltration through windows
T0, Tb	Ambient and basement temperature
Tsp_air, Tsp_floor	Set-point temperature of room air and radiant floor
S2, S6	70% and 30% of total solar gain absorbed by the floor and the remaining surfaces
$AirQ_{aux}$, $FlrQ_{aux}$	Auxiliary heating/cooling source for room air and radiant floor

The convective conductances between room air and interior surfaces are calculated by Equation (1) [7]. The radiative conductances between interior surfaces are linearized and calculated using Equation (2) [7].

$$U_{1,i} = A_i h_{c,i} \tag{1}$$

$$U_{i,j} = A_i F^*_{i,j} 4\sigma \left(T_m\right)^3 \tag{2}$$

The general form of energy balance equations corresponding to node i at time interval p and $p + 1$ are: nodes with thermal capacitance (Equation (3)) and nodes without thermal capacitance (Equation (4)):

$$T_i^{p+1} = \frac{\Delta t}{c_i}\left[Q_i^p + \sum\nolimits_j U_{i,j}\left(T_j^p - T_i^p\right)\right] + T_i^p \tag{3}$$

$$T_i^{p+1} = \frac{Q_i^p + \left[\sum\nolimits_{j\neq i} U_{i,j} T_j^p\right]}{\sum\nolimits_{j\neq i} U_{i,j}} \tag{4}$$

The auxiliary cooling load, Q_{aux}, is calculated at each time step by applying proportional control as in Equation (5). The daily cooling load is the integration of Q_{aux} at each time step over 24 h interval.

$$Q_{aux}^{p+1} = k_p\left(T_{sp}^p - T_1^p\right) \tag{5}$$

where:

- A_i: area of surface i (m^2);
- $h_{c,i}$: convective heat transfer coefficient of surface i ($Wm^{-2}K^{-1}$);
- σ: Stefan-Boltzmann constant ($Wm^{-2}K^{-4}$);
- $F_{i,j}$*: radiative exchange factor between surfaces i and j;
- T_m: mean surface temperature (°C);
- Ui,j: thermal conductance between nodes i and j (WK^{-1})
- Q_i: heat at node i (due to internal heat gain, solar heat gain or auxiliary heat) (W)
- k_p: proportional control
- T_{sp}: set-point temperature (°C)
- T_i: temperature at node i (°C)
- C_i: thermal capacitance of node i (JK^{-1})
- Δt: simulation time step (s)

3. Model Verification

Simulation and measured data of room air temperature were compared to validate the model. The simulation is performed based on constant room set-point temperature (24 °C) with shades fully open. Combined convective and radiative heat transfer coefficients are considered constant with temperature between the interior surfaces and room air node, whereas radiative heat transfer coefficients are considered between interior surfaces. The results presented in Figure 3 show that there is good agreement between the measured and simulated room temperatures.

4. Simulation Strategies

Simulations are performed for an exposed floor (thermal mass) for seven consecutive days in summer to demonstrate the potential performance of HV with fixed and variable schedules in comparison with baseline simulations. Table 2 shows the summary of the different control strategies.

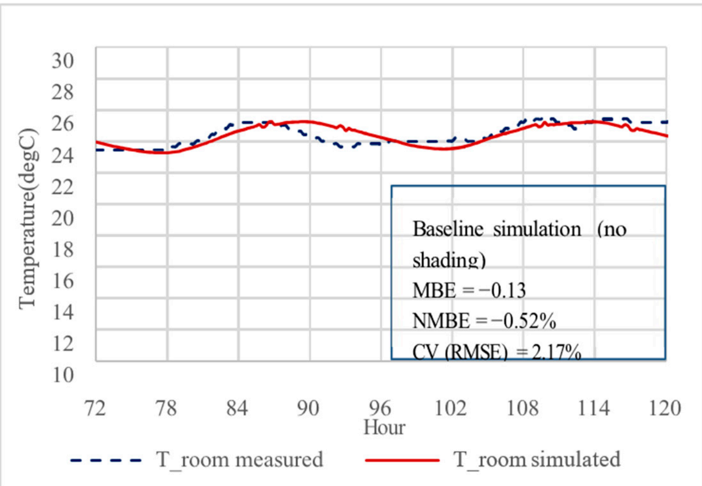

Figure 3. Model validation with measured data for baseline simulation (shades fully open).

Table 2. Different control strategies for simulations.

	Cooling Mode	Set-Point Temperature T_{sp}	Motorized Window (Open/Closed)
Baseline (no shading)	100% mechanical cooling	T_{sp_air} = 24 °C T_{sp_floor} = 23 °C	CLOSED
Baseline (shading 11.00 a.m.–4.00 p.m.)	100% mechanical cooling	T_{sp_air} = 24 °C T_{sp_floor} = 23 °C	CLOSED
HV with fixed schedules (4 ach)	Mixedmode cooling	T_{sp_air} = 24 °C T_{sp_floor} = 23 °C	CLOSED (9.00 a.m. to 6.00 p.m.); Otherwise OPEN
HV with fixed schedules (6 ach)	Mixedmode cooling	T_{sp_air} = 24 °C T_{sp_floor} = 23 °C	CLOSE (9.00 a.m. to 6.00 p.m.); Otherwise OPEN
HV with variable schedules (4 ach)	Mixedmode cooling	T_{sp} = 24 °C (8.00 a.m. to 4.00 p.m.) T_{sp} = 26 °C (remaining hr)	OPEN (T_{amb} ≈ 15 °C to 25 °C)
HV with variable schedules (6 ach)	Mixedmode cooling	T_{sp} = 24 °C (8.00 a.m. to 4.00 p.m.) T_{sp} = 26 °C (remaining hr)	OPEN (T_{amb} ≈ 15 °C to 25 °C)

Baseline simulations are performed at constant set-point temperature with full mechanical cooling with and without shading control. Motorized shading control is applied on south façade windows to reduce the solar gain from 11.00 a.m. to 4.00 p.m. which will eventually minimize the cooling load. HV with fixed schedules are performed at constant set-point temperature but allowing natural ventilation through motorized windows from 6.00 p.m. to 9.00 a.m. Windows are closed from 9.00 a.m. to 6.00 p.m. to avoid entering hot ambient air inside the building. Typical design of natural ventilation in buildings has an air change rate of around 5 ach [8]. In this case, natural ventilation with 4 and 6 air change per hour (ach) are considered and no significant difference between the two cases is found. In HV with variable schedules strategies, motorized windows are open when ambient air temperature is in the range of 15 °C to 25 °C during occupied hours (8.00 a.m. to 4.00 p.m.) while maintaining 24 °C set-point temperature in occupied hours and 26 °C for unoccupied hours. Tdew ≤ 13.5 °C and wind speed (Wspd ≤ 7.5 m/s) should also be addressed for permitting natural ventilation, but for simplicity these parameters are not considered in this simulation. In general, when motorized windows are open for natural ventilation, mechanical cooling system turns off and vice versa.

5. Results and Discussion

Simulation results are shown in Figures 4–7. Temperature graphs in Figures 4–6 are shown for 4th and 5th days since first three days simulated results are unstable. The coefficient of variation of the root-mean-square error (CV-RMSE) and normalized mean bias error (NMBE) provide additional information about the performance of the model.

Figure 4 shows the room temperature for baseline simulations for with/without shading control. Baseline simulations are performed at full mechanical cooling. Motorized shading control is applied to minimize solar gain from 11.00 a.m. to 4.00 p.m. There is no significant fluctuation in room temperature (less than 0.5 °C for most of the time and maximum 1.3 °C when the simulations are unstable at the beginning). Figure 7 shows cooling load would be reduced by more than 47% by applying shading devices during those hours over baseline simulation with no shading.

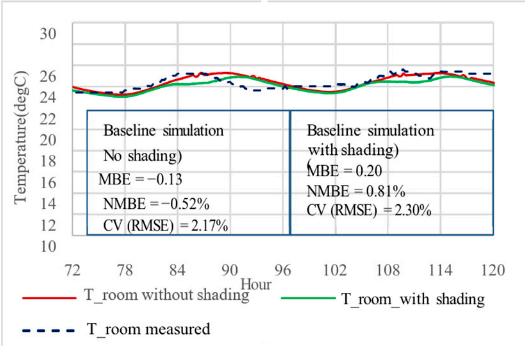

Figure 4. Simulated room temperature for baseline simulation with/without shading.

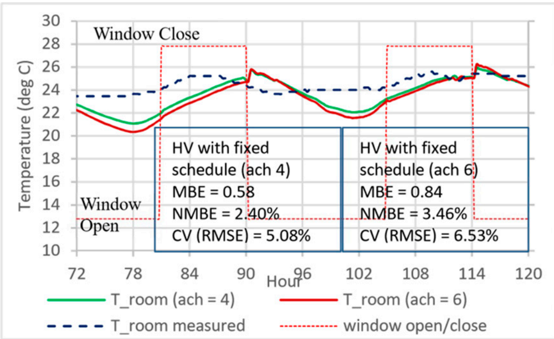

Figure 5. Simulated room temperature for HV with fixed schedules with 4 ach and 6 ach.

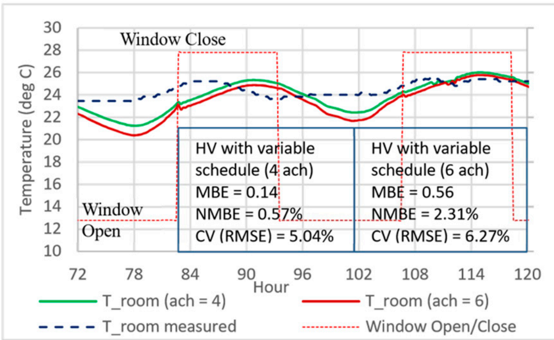

Figure 6. Simulated room temperature for HV with variable schedules with 4 ach and 6 ach.

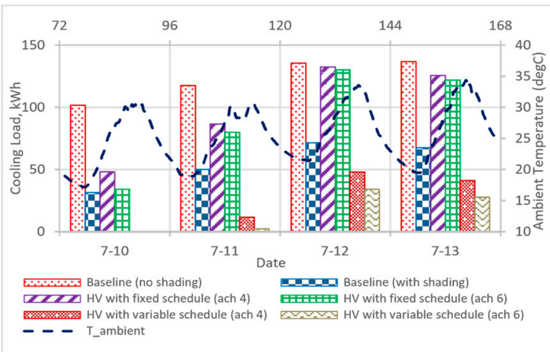

Figure 7. Cooling load simulation results for different control strategies.

Figure 5 shows the room temperature for HV with fixed schedules simulations with 4 ach and 6 ach. Natural ventilation is permitted from 6.00 p.m. to 9.00 a.m. when the ambient air temperature is relatively low. The window opening positions over time are shown in Figure 5. Figure 7 shows simulation results for HV with fixed schedules would achieve at least 10 to 20% energy savings with

4 ach and 6 ach over baseline simulation in any moderate temperature day. But in hot day, HV with fixed schedules show only 2 to 3% energy savings since window opening control is based on fixed hourly basis. In this case, hot ambient air (>30 °C) enters into the building and increases the cooling load. It is worth mentioning that natural ventilation with 4 ach and 6 ach have less impact on temperature variation and cooling load.

Room temperature for HV with variable schedules for both 4 ach and 6 ach are shown in Figure 6. In this case set-point temperature is kept 24 °C during occupied hour (8.00 a.m. to 4.00 p.m.) and 26 °C during unoccupied hour. Natural ventilation is allowed when the ambient temperature is within 15 °C to 25 °C. Figure 7 shows that the HV with variable schedules strategies would achieve more than 65% savings on energy consumption for both 4 ach and 6 ach over baseline simulation.

6. Conclusions

This paper presents the potential of hybrid ventilation (HV) strategies to reduce cooling load over one week period of July. Simulation results show that 10 to 20% savings on cooling loads could be achieved when decisions are made based on HV with fixed schedules and more than 65% savings for HV with variable schedules strategies. For shading control strategies, simulated results show that 47% savings on cooling loads could be achieved over baseline simulation with no shading. Combined convective and radiative heat transfer coefficients between surfaces and air node are considered constant with temperature in this simulation. In future, convective and radiative heat transfer coefficients should be modeled separately in perimeter zone (especially in south perimeter), where temperature varies at every time step. Also, humidity and wind speed for ambient air should be included in future model.

Author Contributions: S.S., A.K.A. and R.G.Z.: conceived and designed the simulations; S.S.: performed the simulation and analyzed the data; S.S.: wrote the paper; A.K.A. and R.G.Z.: review the paper.

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