Evaluating the Performance of Natural Ventilation in Buildings through Simulation and On-site Monitoring

by

Haofan Cheng

B.Eng., Building Technology
Tsinghua University, 2011

Submitted to the Department of Architecture
in Partial Fulfillment of the Requirements for the Degree of

Master of Science in Building Technology
at the
Massachusetts Institute of Technology

June 2013

Signature of Author: ______________________________
Department of Architecture
9 May, 2013

Certified by: ______________________________
Leon R. Glicksman
Professor of Building Technology
Thesis Supervisor

Accepted by: ______________________________
Takehiko Nagakura
Associate Professor of Design and Computation
Chair of the Department Committee on Graduate Students

[this page intentionally left blank]

Evaluating the Performance of Natural Ventilation in Buildings through Simulation and On-site Monitoring

by

Haofan Cheng

Submitted to the Department of Architecture on May 10, 2013
in Partial Fulfillment of the Requirements for the Degree of
Master of Science in Building Technology

ABSTRACT

Natural ventilation in buildings is capable of reducing energy consumption while maintaining a comfortable indoor at the same time. It is important that natural ventilation is taken into consideration in the early design stage, probably through simulation program. However, existing simulation programs are limited because of their model assumption, simulation efficiency and user friendliness.

In this document a simulation program, CoolVent, is presented. It implements a multi-node, thermal and energy coupled model to simulate natural ventilation in buildings. The program is validated to be effective and efficient in simulation. Combined with the DOE building database, CoolVent evaluates the performance of different ventilation modes in different types of building and weather conditions.

The second half of this thesis includes a detailed on-site monitoring study of a naturally ventilated building in Boston. It details the post-occupancy system characteristics. The monitoring results have also been compared with CoolVent. The simulation results are verified by comparing with the monitored ones. The program is then used to evaluate the potential of improved operation.

Thesis Supervisor: Leon R. Glicksman
Title: Professor of Building Technology

[this page intentionally left blank]

ACKNOWLEDGEMENTS

First of all, I want to express my sincere gratitude toward my academic advisor, Professor Leon R. Glicksman. His deep understanding of the subject, earnest attitude toward scientific research and warm encouragement and support for me have been the most significant and valuable thing I experienced during my study at MIT and will continue to be an important part of my future.

I would like to thank all the faculties, staffs and students in the Building Technology Program. Especially, Professor Leslie K. Norford, for his generous advice and help on my application, course study and research experiment; Departmental Administrator Kathleen Ross, for her detailed help on all the logistic issues and care for me; my research group mates Alonso, Alejandra and Steve, for their help and advice on both the development of CoolVent and the monitoring experiment.

I want to thank my family and friends, both nearby and remote, for their love, support and friendship.

Thanks to the MIT Energy Initiative and US-China Clean Energy Research Center for providing me this research opportunity and financial support. It is my honor and luck to be at MIT.

BIOGRAPHIC NOTE

I completed my Bachelor of Engineering degree with honor in July 2011 at Tsinghua University. During that time I learned a comprehensive scope of courses concerning building science and technology. My undergraduate research experience covers thermal comfort, indoor air quality and on-site monitoring in buildings. I have attended exchange program in INSA Lyon and the Hong Kong Polytechnic University. Since coming to MIT in the fall of 2011, I have had the opportunity to work on a project concerning natural ventilation in buildings under the US-China Clean Energy Research Center (CERC). I am also a member of the MIT Energy Initiative. In the future I hope I can make contribution to the career of clean energy and sustainable development.

Please send questions or comments to the author at:

hfcheng@alum.mit.edu

TABLE OF CONTENTS

LIST OF TABLES AND FIGURES

CHAPTER 3 INTRODUCTION TO COOLVENT

CHAPTER 4 SIMULATION IN COOLVENT

1 INTRODUCTION

People spend approximately 90 percent of their life time in buildings [1]. Hence, it is of significant importance to maintain a healthy, comfortable and productive indoor environment. Building sectors consumes large amount of energy. In the United States, building operation consumes 39% of the total energy consumption [2], which makes it necessary to make building more energy efficient and environmentally friendly. Figure 1-1 shows the composition of US building energy consumption and its weight under the global context.

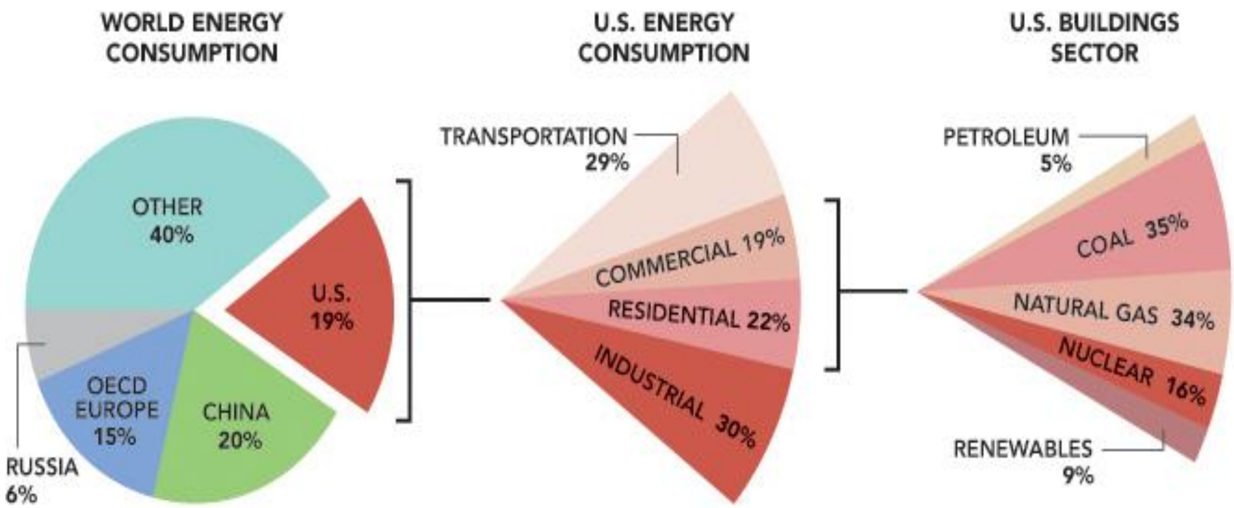


Figure 1-1. US building energy consumption composition

As a part of building operation, HVAC consumes a significant part of total building energy consumption. Natural ventilation has the potential of reducing energy consumption of the HVAC system and maintaining a suitable, sometimes even more productive [3] indoor environment. Figure 1-2 and Figure 1-3 show energy consumption of different sub-section of commercial and residential building. It shows that space cooling, refrigeration and ventilation consume 30.2% energy of commercial building and 21.5% for residential building and thus have a great potential of energy saving through taking advantage of natural cooling source.

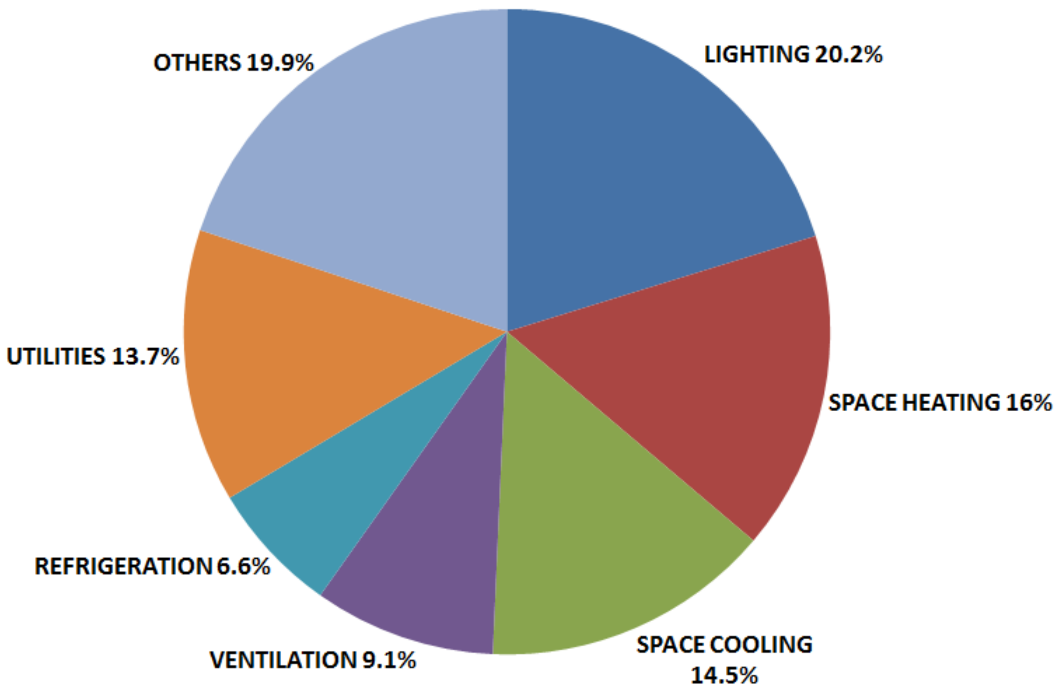


Figure 1-2. 2010 Commercial energy end-use splits (Primary energy)

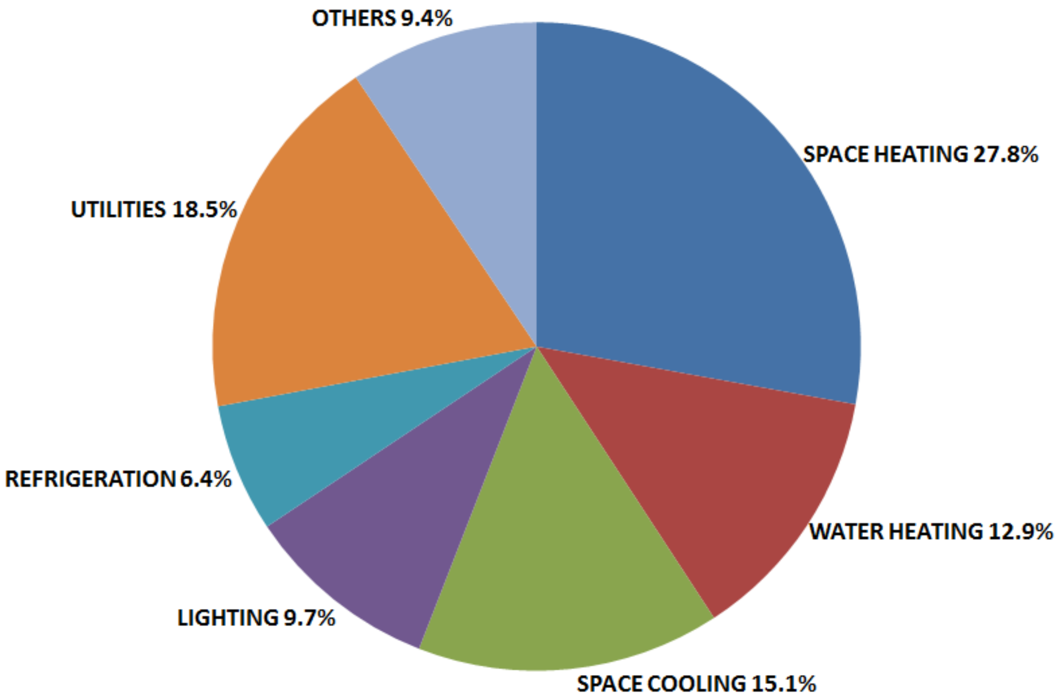


Figure 1-3. Residential energy end-use splits (Primary energy)

Hence, its importance has been recognized among the sustainable building design community. However, there are several existing problems impeding the widespread use of natural ventilation.

The first issue is that current building natural ventilation design lacks a quick and accurate simulation tool for the early design stage when many different plans have to be tried.

Another issue among the natural ventilation design community is although the building is designed with a promising future, its actual performance is uncertain. There have been some reports about existing naturally ventilation buildings and some problems have been pointed out. Hence, it is necessary to conduct field studies to evaluate the performance of those buildings.

The beginning section of this thesis is a literature review of ongoing related research. It is concentrated on existing research progress on energy saving potential of natural ventilation in buildings. Some common characteristics are summarized and limitations for future improvement are also pointed out. This serves as the foundation of thesis outline.

Chapter 3 presents the calculation method of CoolVent – an effective, efficient and user-friendly program simulating natural ventilation buildings. Firstly the interface is introduced. The interface is easy to use for designers who might not have a deep understanding of the physics of natural ventilation. Then several important equations are presented including the buoyancy and wind coupled airflow model, energy conservation model, air conditioning model and thermal mass model. The effectiveness and efficiency of this program are tested and verified.

Chapter 4 is simulation results in CoolVent. Several modes are proposed for simulation. The potential of natural ventilation on energy saving and indoor comfort are discussed. A supplementary mechanical cooling system is also simulated when outdoor weather condition is not suitable for natural ventilation. As an important part of natural ventilation, night cooling is studied. To optimize the use natural ventilation, a control algorithm is proposed.

Chapter 5 is the detailed monitoring results of Artists for Humanity Epicenter in Boston. One famous feature of the building is that it is a naturally ventilated building without mechanical air conditioning system. Comfort condition, night cooling effect, airflow and thermal comfort are monitored and analyzed. The building model is simulated under CoolVent. Some suggestions concerning better operation of the building are proposed.

2 LITERATURE REVIEW

In this section, several existing research papers concerning the energy saving potential of natural ventilation are selected and briefly presented. The objective of doing this is to summarize some of their common characteristics and thus discover potential areas that might be explored and improved by results presented in this thesis.

2.1 Existing Results

Yik [4] proposed an approach for simulating energy saving potential in residential building in Hong Kong. This model is composed of three parts: computational fluid dynamics program predicting wind pressure outside opening; a flow network nodal model calculating air flow rate; a thermal program predicting heat transfer and air conditioning. The building is operated under a hybrid mode: during a cooling period, the building is initially operated under natural ventilation; if natural ventilation cannot maintain a comfortable indoor environment, the building is switched into mechanical cooling mode which lasts until the end of that day. Simulation results show that natural ventilation could reduce air-conditioning energy by about 24%. Figure 2-1 shows energy saving potential of natural ventilation for different floors.

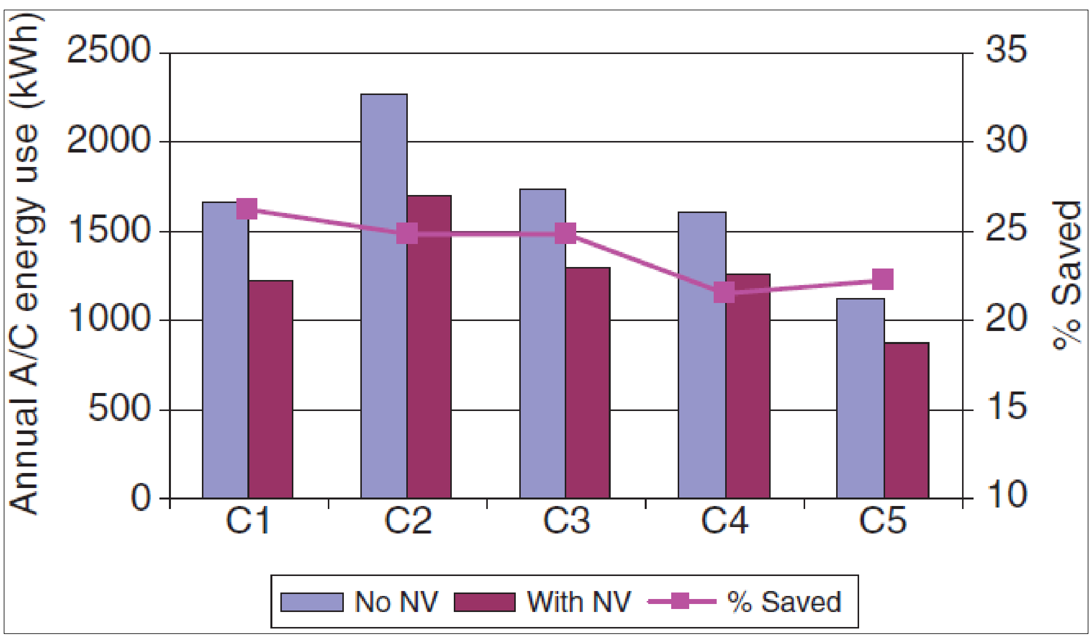
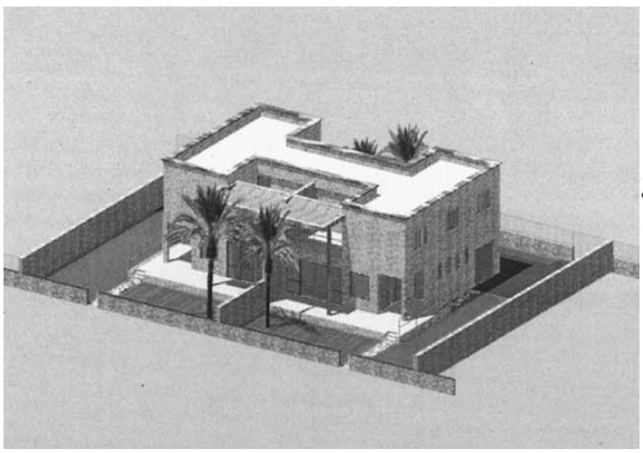


Figure 2-1. Energy consumption comparison for different sections of the building in Hong Kong

Cardinale [5] simulated the energy saving potential of natural ventilation in a traditional building in Italy. Important parameters of the model are building orientation, window position and exposure to solar radiation. Building materials are set to be uniform here to incorporate ordinary buildings. Simulation program used here is AIOLOS. The operation scheme is straightforward: the building will be open at night (6pm to 9am next day) and closed at daytime. Supplementary air conditioning system is used to maintain indoor environment at comfortable level. Results show that this strategy can save cooling load by 41%, 46% and 52% for three weather types in Italy.



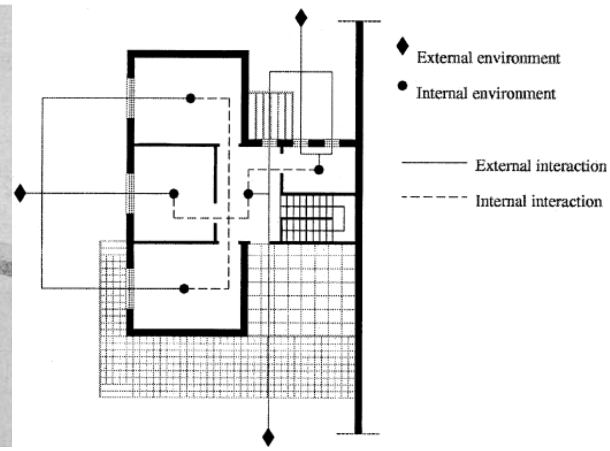


Figure 2-2. Building image and zone schematization

Zhao [6] simulated night ventilation assisted mechanical cooling strategy for large supermarkets in northeastern China, which is in a cold climate. Two models set up for this simulation are thermal storage of building thermal mass and indoor temperature model based on energy conservation. Using these models, total energy consumption is analyzed. Different operation strategies have been tried to find an optimal one. Results show that mechanical cooling and night ventilation are affecting each other, while two key parameters are operating period and air flow rate. The reduction of mechanical cooling load (or hours) resulted from night cooling is analyzed. Different operation strategies have been proposed under different weather conditions. Night cooling is estimated to save 3 kWh/m^2 annually in northeastern China.

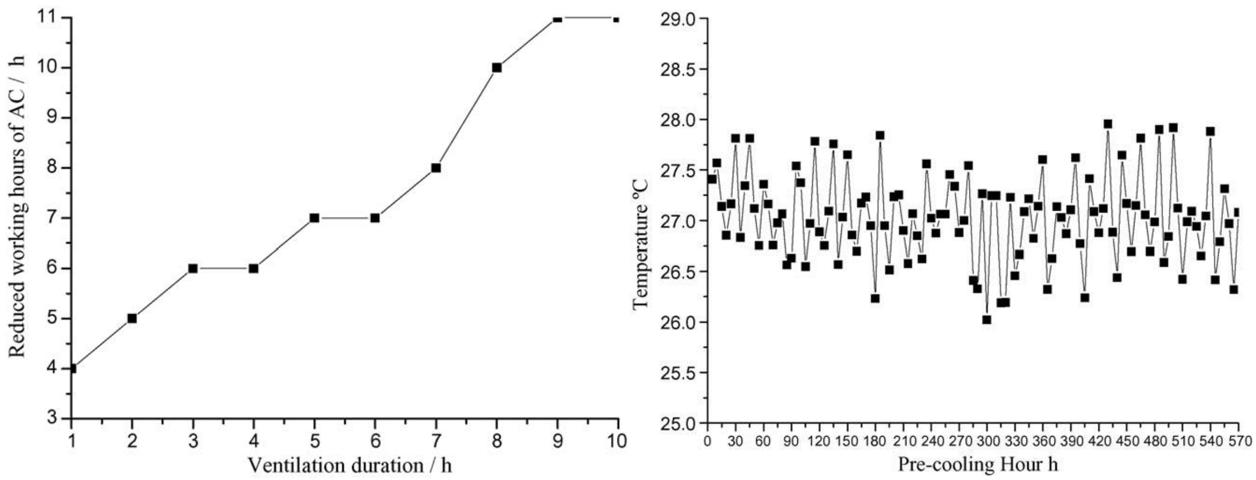


Figure 2-3. Relationship between the ventilation duration and the reduced working hours of active cooling & temperature change during the pre-cooling phase

Facing the fact the most studies on this field concentrate on theoretical aspects, Yang [7] conducted an experimental study on a building in Jacksonville in Florida. The objective of this study is to explore the impact of thermal mass on both peak cooling load and total cooling energy. Several measured parameters are air temperature and flow rate, thermal mass temperature and heat flux. Monitoring results show that the total cooling load at daytime is expected to be reduced by 18% while peak cooling load remains the same.

Luo [8] proposed an analytical model to evaluate the potential of natural ventilation for four different cities representing different weather types in China. The models used include a building model for energy conservation, buoyancy and wind force combined natural ventilation model, adaptive thermal comfort model and an ASHRAE model for indoor air quality. There are three steps to take this model into practice: calculate airflow rate and indoor air temperature based on the first two models; compare indoor air temperature and comfort limit; pick up the satisfactory points and derive natural ventilation potential. Figure 2-4 shows monthly natural ventilation hours.

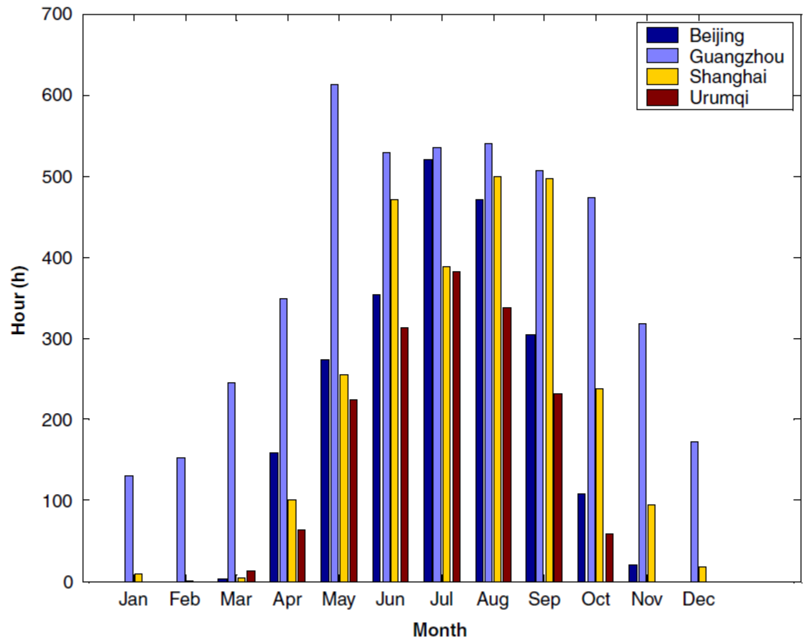


Figure 2-4. Monthly natural ventilation hours considering both thermal comfort and IAQ

Schulze [9] conducted a coupled airflow network and dynamic building simulation to evaluate the annual thermal comfort and energy saving potential of natural ventilation. Thanks to the cool weather type in Europe, both the indoor air quality and temperature could be maintained at a comfortable level. Control is needed to prevent excessive cooling. The estimated saving of net cooling energy is 13-22 kWh/m^2 in Stuttgart, 32-36 kWh/m^2 in Turin and 38-44 kWh/m^2 in Istanbul. Figure 2-5 below shows the simulation model.

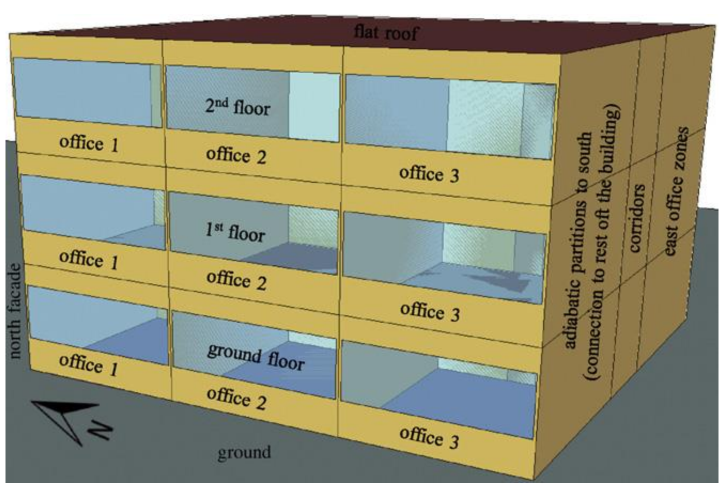


Figure 2-5. Simulation model of the office building

Based on the tentative conclusion that moving air can improve occupants' tolerance toward hot environment (upper limit for still indoor air is 26°C, which an equivalent temperature reduction of 1.1, 1.9 and 3.3°C can be obtained respectively at air velocity of 0.2, 0.4 and 1 m/s), Ayata [10] performed for different building models under different air velocities. Figure 2-6 below shows the distribution of air velocity. Based on this simulation results, it is concluded that natural ventilation cooling could satisfy comfort for 20% of the occupied time in a building in Bangkok.

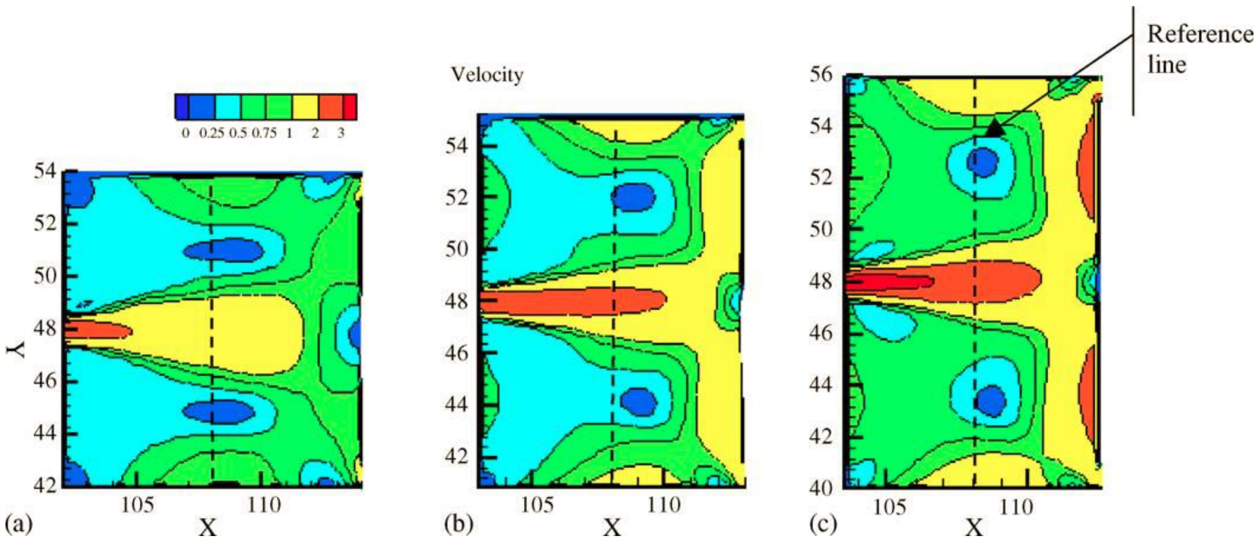


Figure 2-6. Air distribution inside a room shown in different colors

Ji [11] simulated the energy saving potential of natural ventilation on a building located in Hangzhou, a city in south China under sub-tropical climate. The building model is first simulated under CFD to evaluate the airflow under different weather types and openings. Then the results are used as input of IES Virtual Environment, a well-established tool for analyzing the dynamic responses of a building. A simple strategy controlling dampers openness based on indoor and outdoor temperature at different time is introduced. Based on this strategy, cooling loads under both hybrid mode (a combination use of natural ventilation and mechanical cooling) and mechanical cooling mode are compared. Results show that hybrid mode is expected to save 30-35% cooling energy from the mechanical mode. Figure 2-7 below shows a monthly comparison.

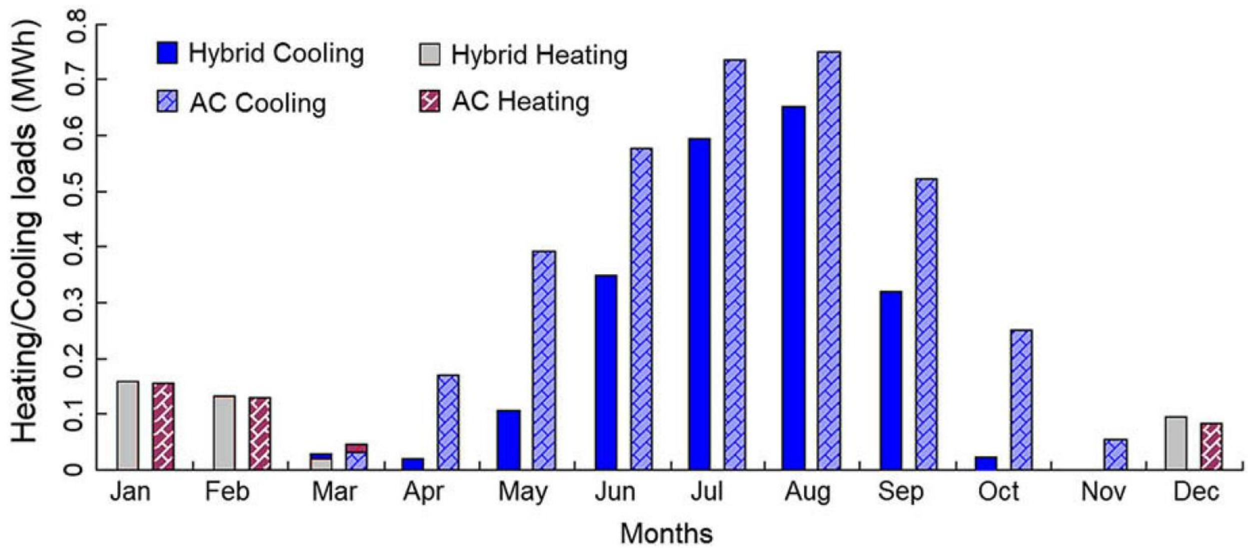


Figure 2-7. Annual heating and cooling loads of the space for space for AC and hybrid ventilation systems

Lomas [12] conducted simulation on a building operated under hybrid mode. The building is a library for a college located near Chicago. Combined simulation of thermal modeling and CFD is used under the likely operation of the building. Thermal comfort under this mode is evaluated by defining dry resultant temperature (DRT).Results show that when night cooling is used, internal DRT never exceeded 28°C, which is a significant improvement from the case when night cooling is not being used (value for this case is 395 h in total). This paper does not deal with air conditioning energy.

Olsen [13] presents simulation results from Energy Plus on a UK building under natural ventilation and hybrid mode. To simulate natural ventilation in Energy Plus, CFD simulation is firstly used to estimate the pressure and discharge coefficients. Then natural ventilation is simulated in Energy Plus. Results show that under pure natural ventilation mode, indoor temperature exceeds 25°C for 191 h and 28°C for only 40 h, which is quite close to a strict comfort standard. Control of the hybrid mode is very straightforward: the openness is kept to provide only the minimum airflow rate all the time. The hybrid system is estimated to save 22% energy from the optimal mechanical system and 42% energy from the existing building. Figure 2-8 below shows energy consumption under different cases.

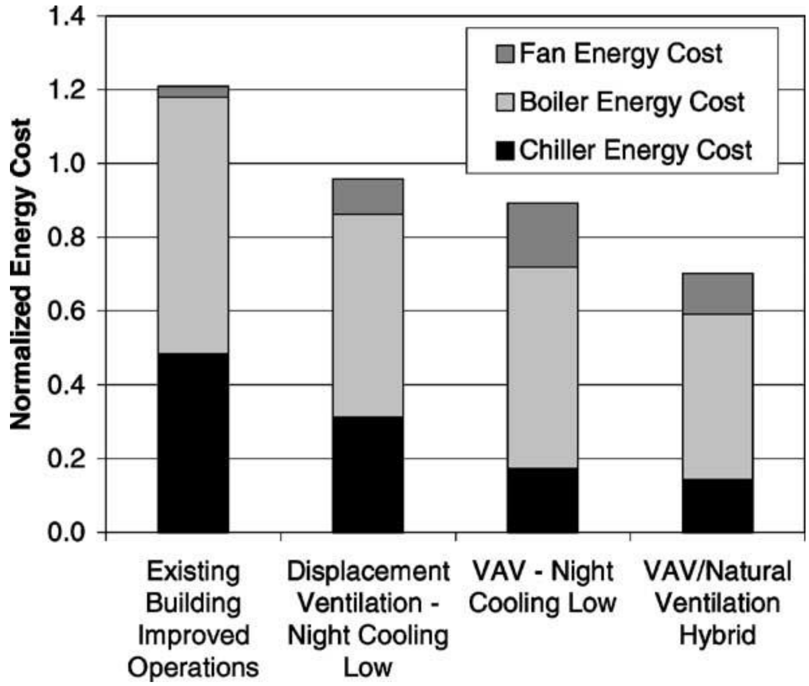


Figure 2-8. Normalized total annual energy cost of best-case system

Crawley [14] compares 20 current available building simulation program. As for the natural ventilation section, all programs have the function of single zone infiltration model. However, few of them deal with natural ventilation and fewer have a pressure network model. One important feature of this is that programmers are associated with mechanical system manufacturing and tend not to support non-mechanical system.

Pfafferott [15] shows monitoring and simulation results of a passive cooling building in Germany. Monitoring scope includes room and ceiling temperature of 12 office rooms, supply air rate and temperature, electricity consumption, climate and air temperature. Results show that the building consumes only about half primary energy or comparable and conventionally designed buildings. Indoor climate conditions also appear to be acceptable: operative temperature exceeds 25°C for only 280 h and 400 h for two major rooms during a 2600 working hours. Simulation of the building model is conducted in ESP-r and there is a good agreement between monitoring and simulation results for a long period of time. Figure 2-9 below shows image of the building and simulation model.

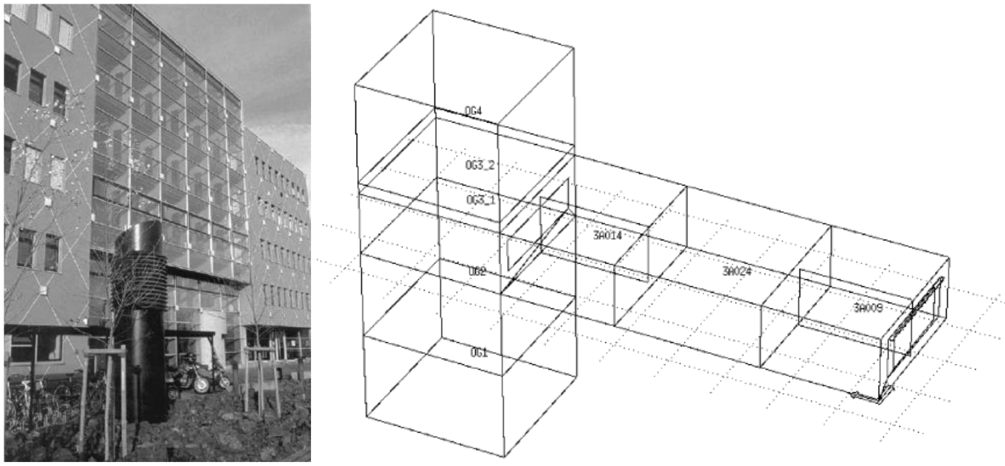


Figure 2-9. Image and model of DB Netz AG building (Germany)

Cook [16] presents a case study of a building using natural ventilation with night cooling. Simulation programs used are ESP-r for dynamic thermal modeling and CFX for CFD simulation. For the first building, Frederick Lanchester Library of Coventry University, the main body of this building is entirely naturally ventilated. Ventilation is driven by buoyancy forces created by internal heat gain. Due to the night cooling effect, simulated temperature could be maintained below outdoor peak at daytime. Figure 2-10 below shows temperature of the third floor of that building. Post-occupancy survey shows that occupants are satisfied with indoor environment of this building.

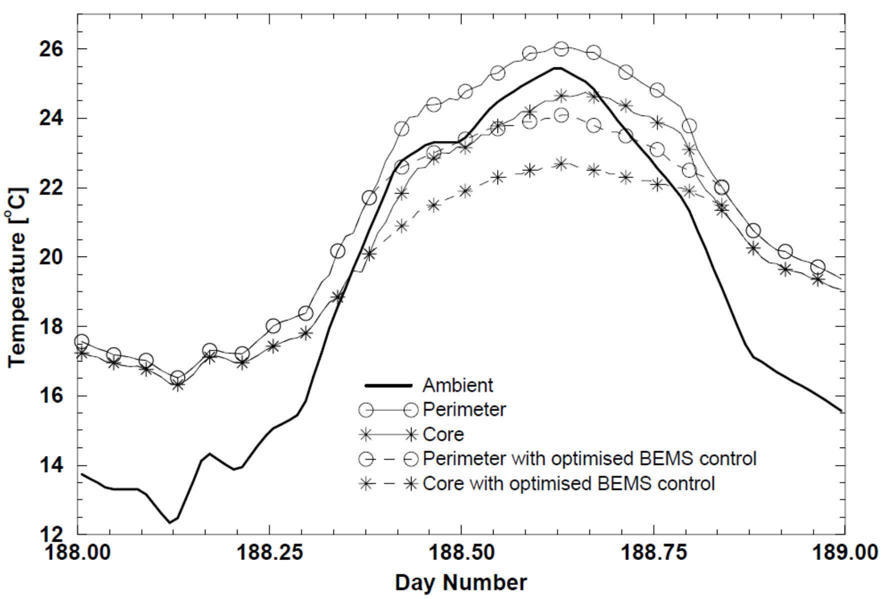


Figure 2-10. Simulated temperature of a typical day in summer

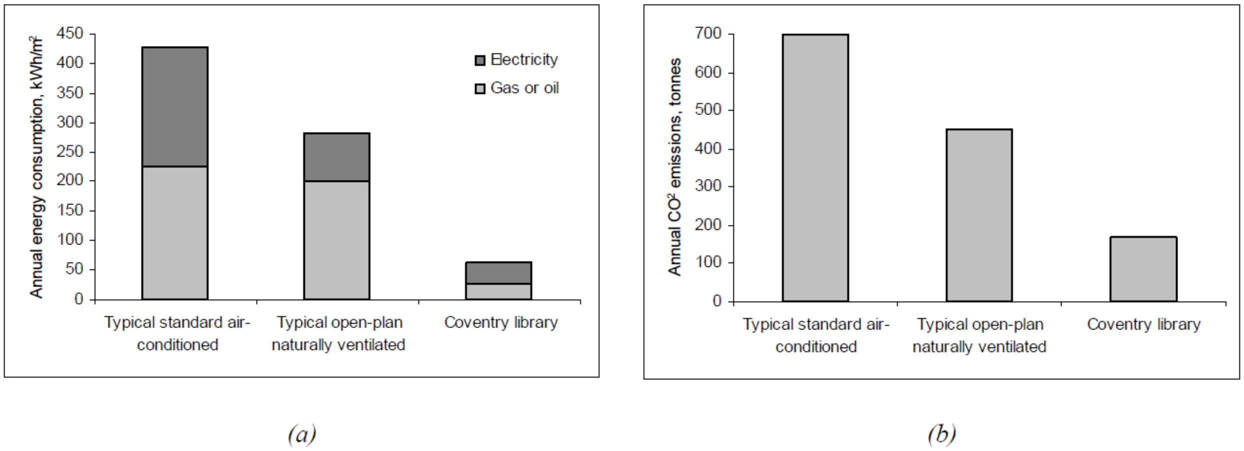


Figure 2-11. Simulated energy consumption and carbon dioxide emission (Respectively for typical standard air-conditioned, typical open-plan naturally ventilated, Coventry library)

Wang [17] conducted field survey on naturally ventilated building to investigate occupants' thermal comfort sensation in Harbin, a city located in cold climate in China. The results are compared with the adaptive comfort standard and previous research results. The neutral temperature is found to be 23.7°C, lower than that in warm climate and in accordance with the adaptive model. Upper limit of acceptable temperature is found to be 31°C, showing that occupants tend to accept high temperature in naturally ventilated buildings. Researchers also investigated occupants' feeling toward humidity and air velocity. Results show that it is possible to have a passively comfortable indoor environment in summer.

2.2 Summary and Discussion

For simulation of natural ventilation, the most popular field seems to be CFD simulation along with experiment verification. There are several popular CFD models and some improvement and modification are proposed based on them. Value of this work is that they provide some useful estimation of equation or coefficient which could be cited in the program design (For example, wind pressure coefficient in singe side ventilation).

As for the energy saving potential of natural ventilation, there are several common characteristics of existing research concentration.

The first aspect is that most simulations concerning energy saving potential are conducted for a specific building type under a specific weather type, typically the residence of the author. However, the energy saving potential of natural ventilation under a broader scope is rarely mentioned. As existing results show, the suitability of natural ventilation ranges from completely reducing cooling load (for example, cool weather conditions in Europe) to very limited use.

As for simulation methodology, most researchers choose a combination of airflow simulation and thermal simulation, which are conducted by two different programs. This might be because current programs lack the ability to conduct a coupled airflow and thermal simulation. Accurate CFD requires a deep understanding of building physics which might not be widely recognized by every architectural designer. This simulation is also very time-consuming: a typical day simulation might cost several real days, not to say a yearly detailed simulation. Considering that architectural design has a significant impact on its potential for natural ventilation, it is necessary to develop an effective, efficient and user-friendly simulation program to assist natural ventilation design in the early stage.

Another limitation of ongoing research is that there is a lack of study on the control algorithm of the hybrid system. In some cases the control algorithm is too simple to fully take advantage of the natural cooling source. For example, in some night cooling studies the building is only open at night and entirely closed at daytime; in some other cases natural ventilation is only used in shoulder season like May or September, while at other times the building is operated under pure mechanical cooling mode. A proper control should be a comprehensive consideration of night cooling, indoor and outdoor environment, comfort condition and feasibility in real practice.

3 INTRODUCTION TO COOLVENT

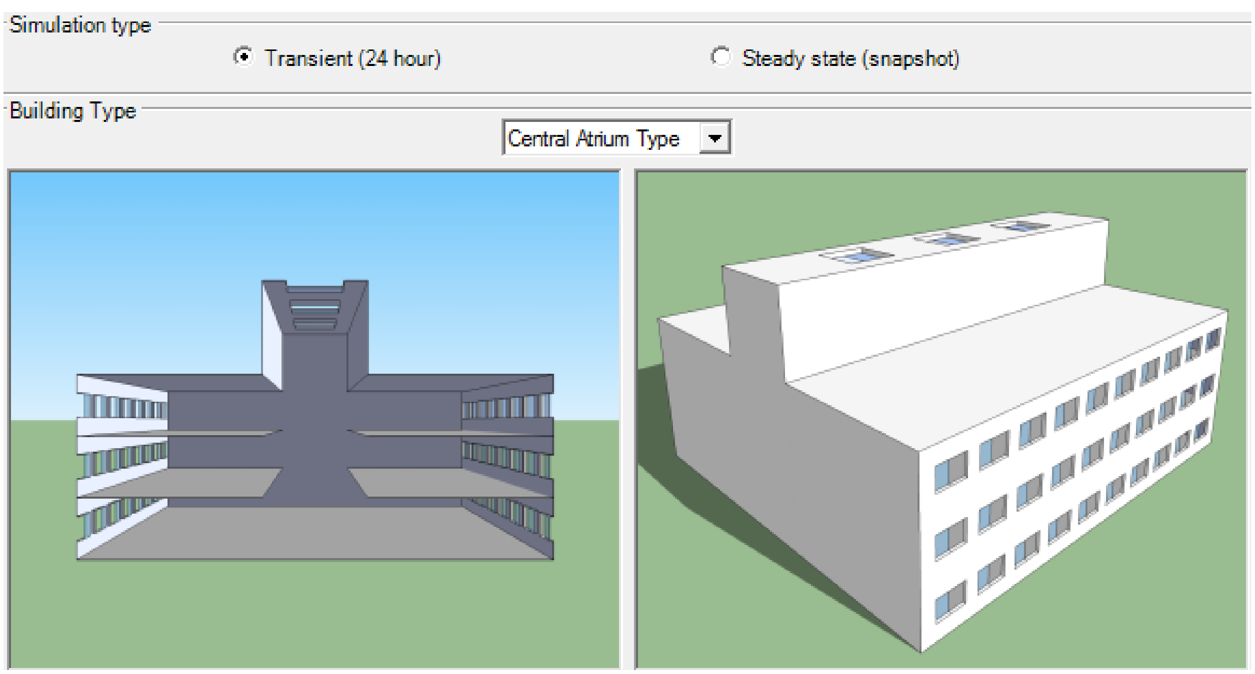
Coolvent is building simulation program concentrating on natural ventilation. It is a fast, accurate, user-friendly program simulating wind and buoyancy combined ventilation. It has been developed by Tan [18], Yuan [19], Menchaca [20] and being improved with new features currently. The aim of this program is to help designers who might not have deep understanding toward natural ventilation learn about the effect of building design on natural ventilation and corresponding energy performance. Another aim is to run the simulation quickly, in contrast with the current CFD simulation which might spend several days.

3.1 CoolVent Interface

Inputs of CoolVent are collected in the interface. There are six tabs of inputs which are main inputs, transient inputs, building dimensions, internal opening and fan, ventilation strategies and thermal comfort models. These six aspects will be briefly covered below.

3.1.1 Main Inputs

Main inputs deal with basic information of the building including geometry, occupancy schedule and surroundings. Figure and input list of this tab are listed below.



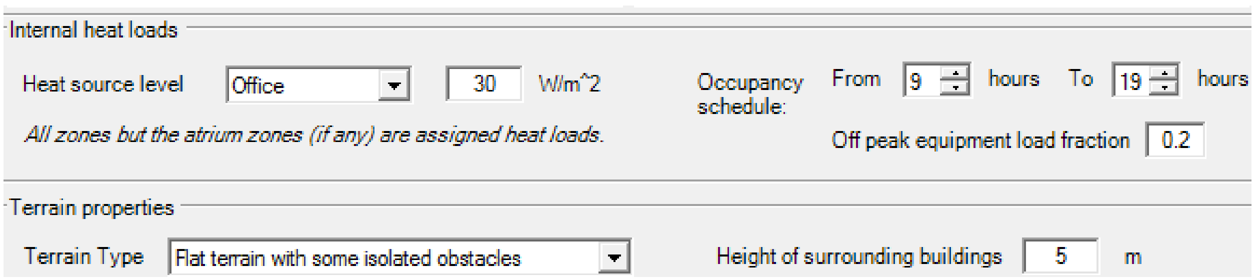


Figure 3-1. CoolVent Input tab – Main inputs

Input name	Available choice	Note
Simulation type	Transient (24 hour)	Choose between 'Transient' (24h simulation, average data of a month) and 'Steady' (constant weather data).
	Steady state (snapshot)	
Building type	Central Atrium Type	Four most common building type available for natural ventilation simulation
	Chimney Type	
	Cross Ventilation	
	Single Sided Ventilation	
Heat source level	Office 30 W/m^2	Design value (maximum value) for internal sensible heat gain (occupant and utility), could also be defined by user
	Residential 20 W/m^2	
	Educational 40 W/m^2	
Occupancy schedule	Defined by user (default value 9am to 7pm)	Internal heat gain at maximum value when the building is occupied
Off peak equipment load fraction	Defined by user (default value 0.2)	Fraction of internal heat gain when the building is not occupied
Terrain type	Flat terrain with some isolated obstacles	Terrain type has an impact on wind velocity. This makes velocity outside the building different from that of the monitoring site, which is
	Rural area with low buildings	
	Urban, industrial or forest area	
	Center of a large city	

		typically an airport
Height of surrounding buildings	Defined by user (default value 5 m)	Same as 'Terrain type'

Table 3-1. CoolVent input tab – Main inputs

3.1.2 Transient Inputs

Two factors specified here are weather data and building orientation. Figure and input list of this tab are listed below.

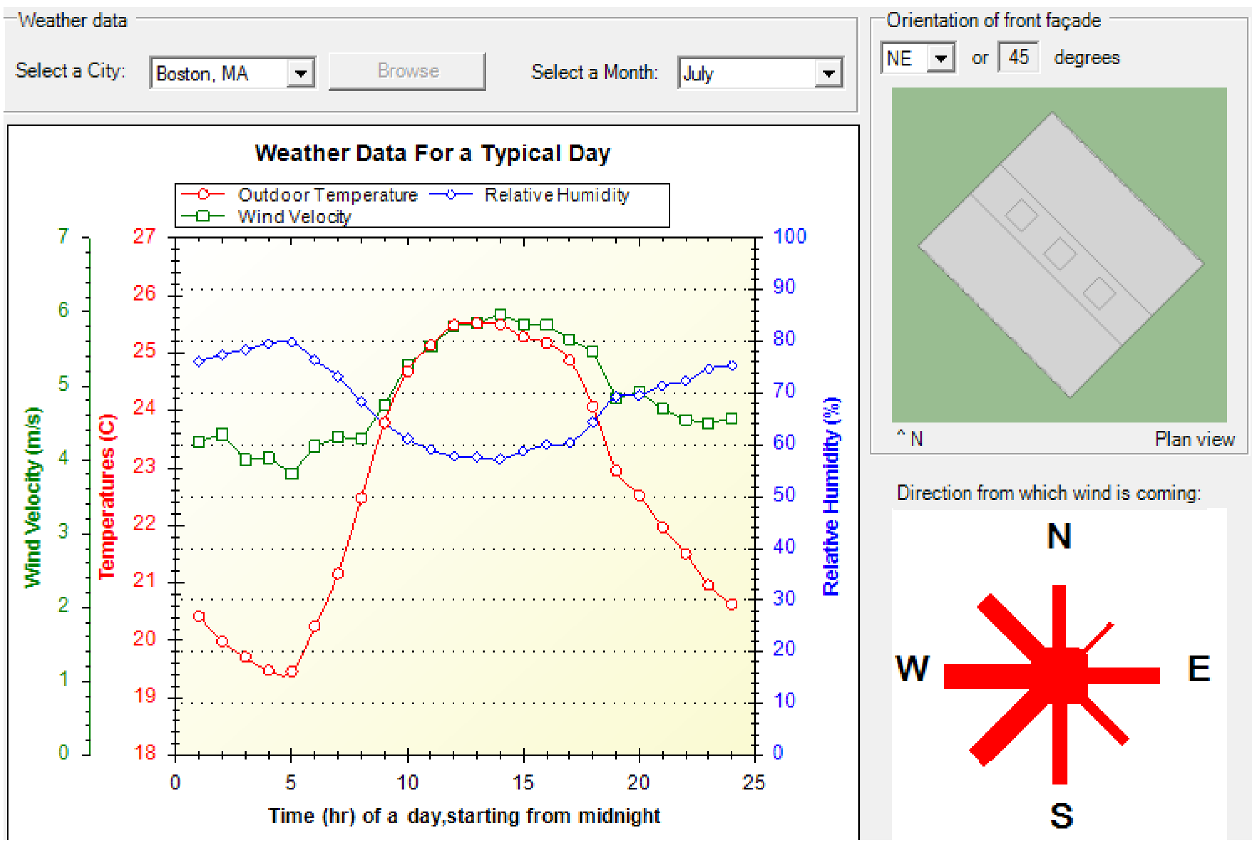


Figure 3-2. CoolVent input tab – Transient inputs

Input name	Available choice	Note
Select a city	13 cities (default Boston, MA)	Weather data from DOE database, could also be imported by user
Select a month	January to December (default July)	Can simulate one or several months on an hourly basis

Orientation of front façade	NE or 45 degrees	Has an impact on incidence angle of wind

Table 3-2. CoolVent input tab – Transient inputs

3.1.3 Building Dimensions

Building dimensions deal with the specific dimension of the building (building type is defined in the Main inputs section). Figure and input list of this tab are listed below. A diagram is also shown.

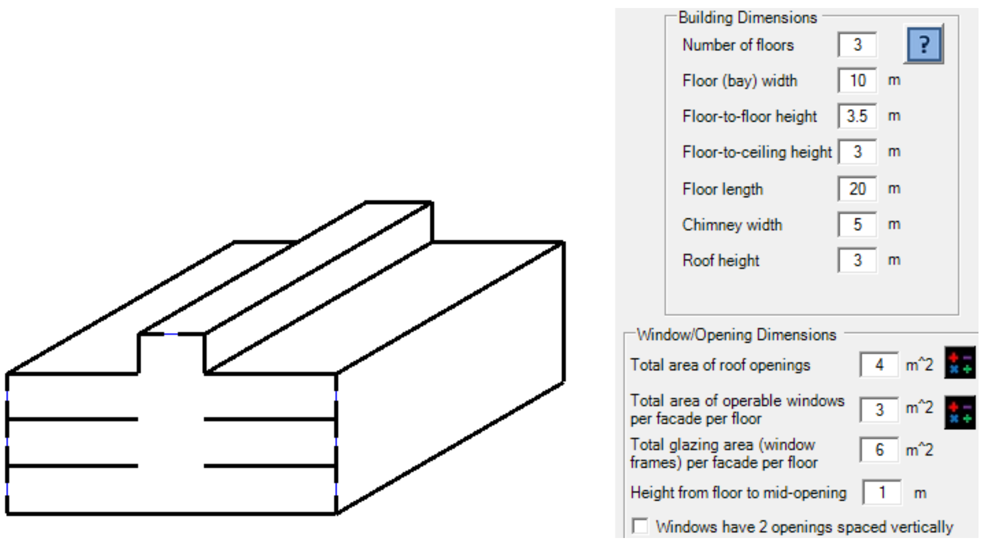


Figure 3-3. CoolVent input tab – Building dimensions

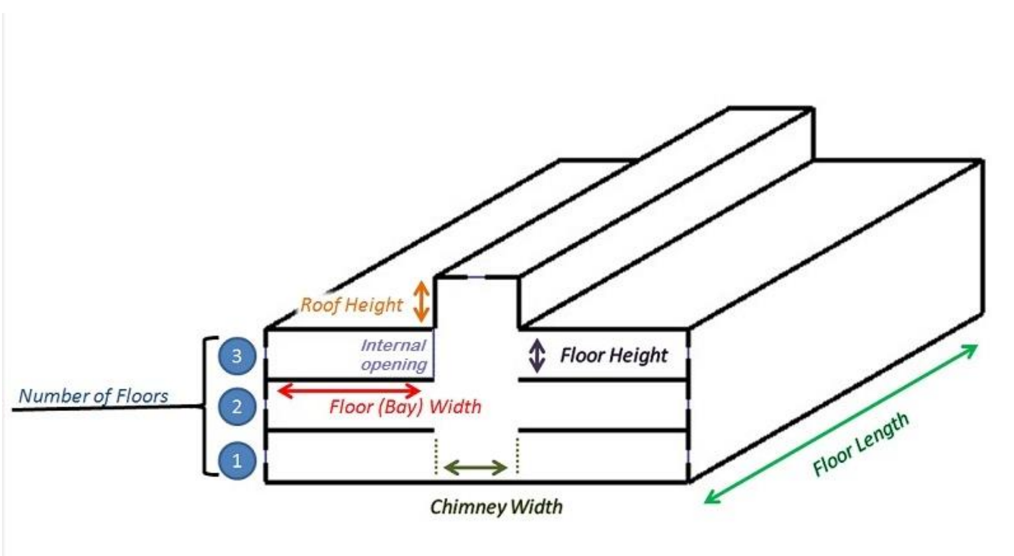


Figure 3-4. Explanation of dimension factors

Input name	Available choice	Note
Number of floors	Defined by user (default value 3 m)	Building dimension will impact calculation of airflow and energy
Floor (bay) width	Defined by user (default value 10 m)	
Floor-to-floor height	Defined by user (default value 3.5 m)	
Floor-to-ceiling height	Defined by user (default value 3 m)	
Floor length	Defined by user (default value 20 m)	
Chimney width	Defined by user (default value 5 m)	
Roof height	Defined by user (default value 3 m)	
Total area of roof openings	Defined by user (default value 4 m)	
Total area of operable windows per façade per floor	Defined by user (default value 3 m)	Has an impact on airflow through window
Total glazing area (window frames) per façade per floor	Defined by user (default value 6 m)	Has an impact on solar radiation (not all windows could be opened for ventilation)
Height from floor to mid-opening	Defined by user (default value 1 m)	Same as other dimension factors
Window have 2 openings spaces vertically	Yes / No	

Table 3-3. CoolVent input tab – Building dimensions

3.1.4 Internal Opening Specification

Internal opening considers the effect of windows, doors and other opening inside a building between different spaces. This will affect the calculation of air flow by adding a resistance between spaces.

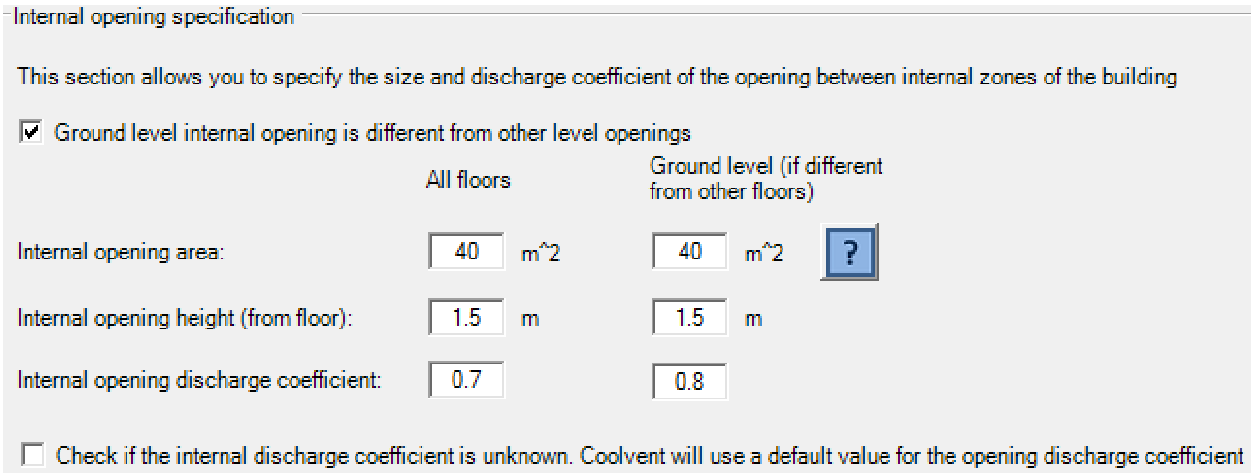
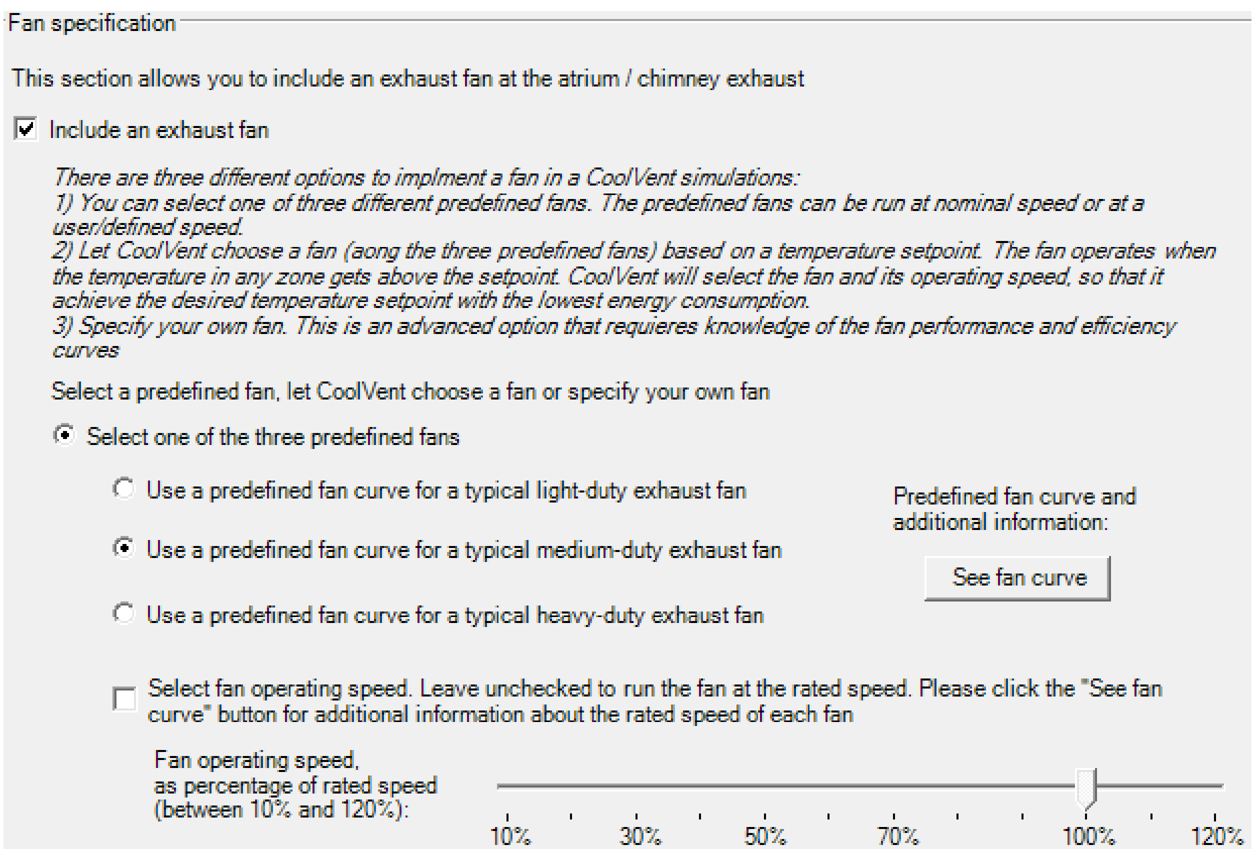


Figure 3-5. CoolVent input tab – Internal opening specification

3.1.5 Fan Specification

User can also choose a fan to increase air flow rate. Fan-assisted natural ventilation is a widely employed method to improve the effectiveness of natural ventilation.



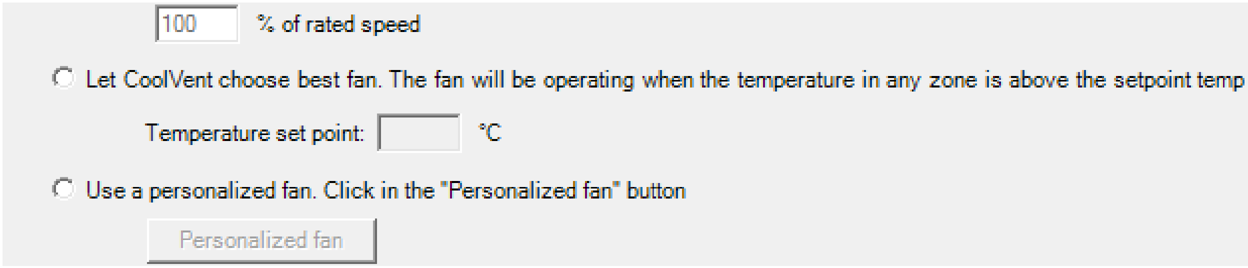


Figure 3-6. CoolVent input tab – Fan specification

The program provides curves of three roof exhaust fans. Fan curve could also be defined by the user.

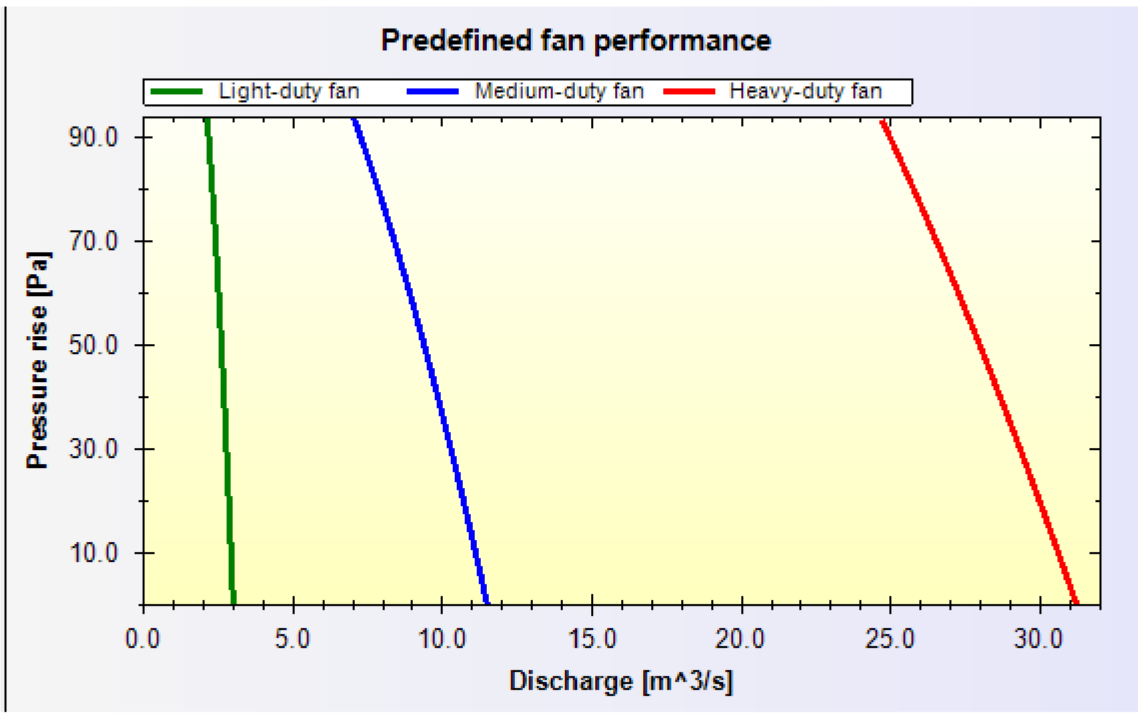


Figure 3-7. Fan curves provided by CoolVent

3.1.6 Ventilation Strategies

Ventilation strategies section specifies parameters of thermal mass, night cooling and window operation. Figure and input list of this tab are listed below.

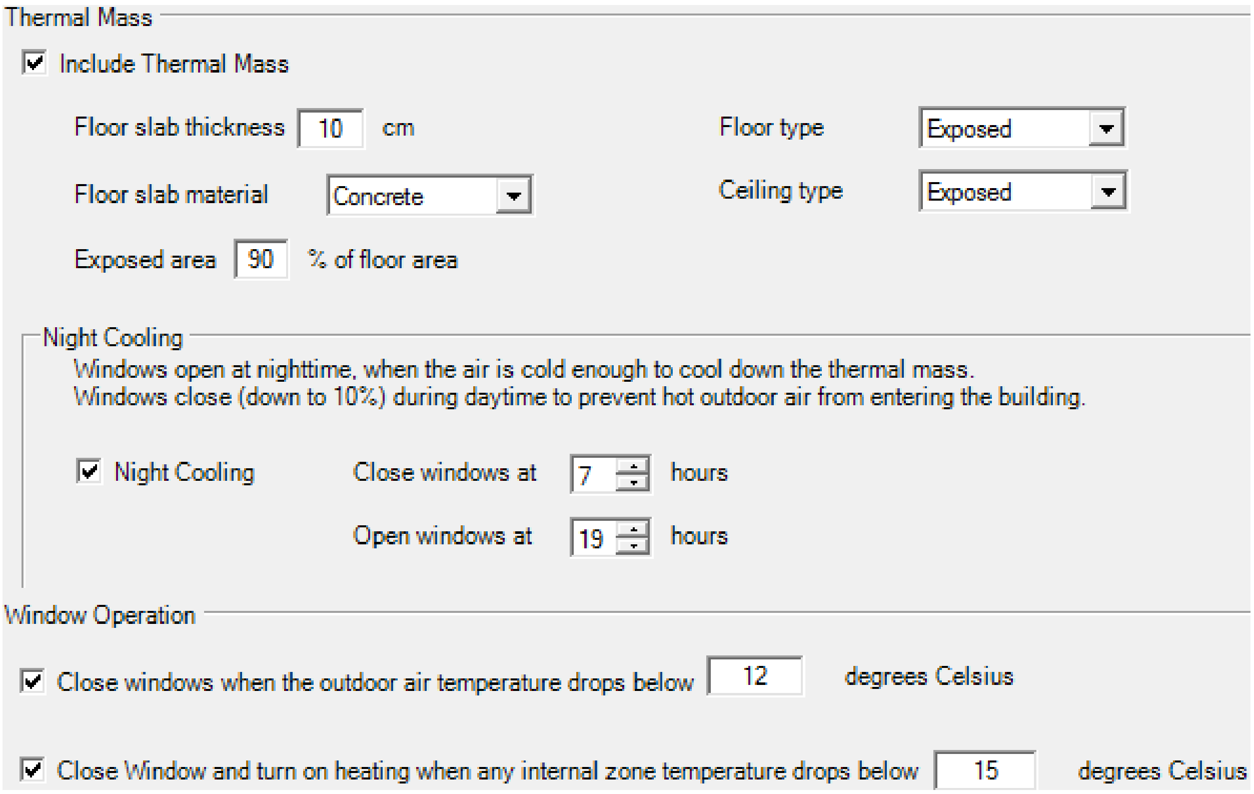


Figure 3-8. CoolVent input tab – Ventilation strategies

Input Name	Example value	Note
Include thermal mass	Yes / No	Include thermal mass in simulation
Floor slab thickness	Defined by user (default value 10 cm)	These parameters affect heat transfer area, capacity and coefficient.
Floor type	Exposed	
	Carpeted	
	Raised	
Floor slab material	Concrete	
	Brick	
	Steel	
Ceiling type	Exposed	
	Suspended	
Exposed area	Defined by user (default value 90%)	

Night Cooling	Yes / No	Turn on night cooling in simulation
Close windows at	Defined by user (default value 7:00)	Time period of night cooling
Open windows at	Defined by user (default value 19:00)	
Close windows when the outdoor air temperature drops below	Defined by user (default value 12℃)	To prevent cold indoor temperature, less important in natural ventilation simulation
Close window and turn on heating when any internal zone temperature drops below	Defined by user (default value 15℃)	

Table 3-4. CoolVent input tab – Ventilation strategies

3.1.7 Thermal Comfort Models

Figure and input list of this tab are listed below.

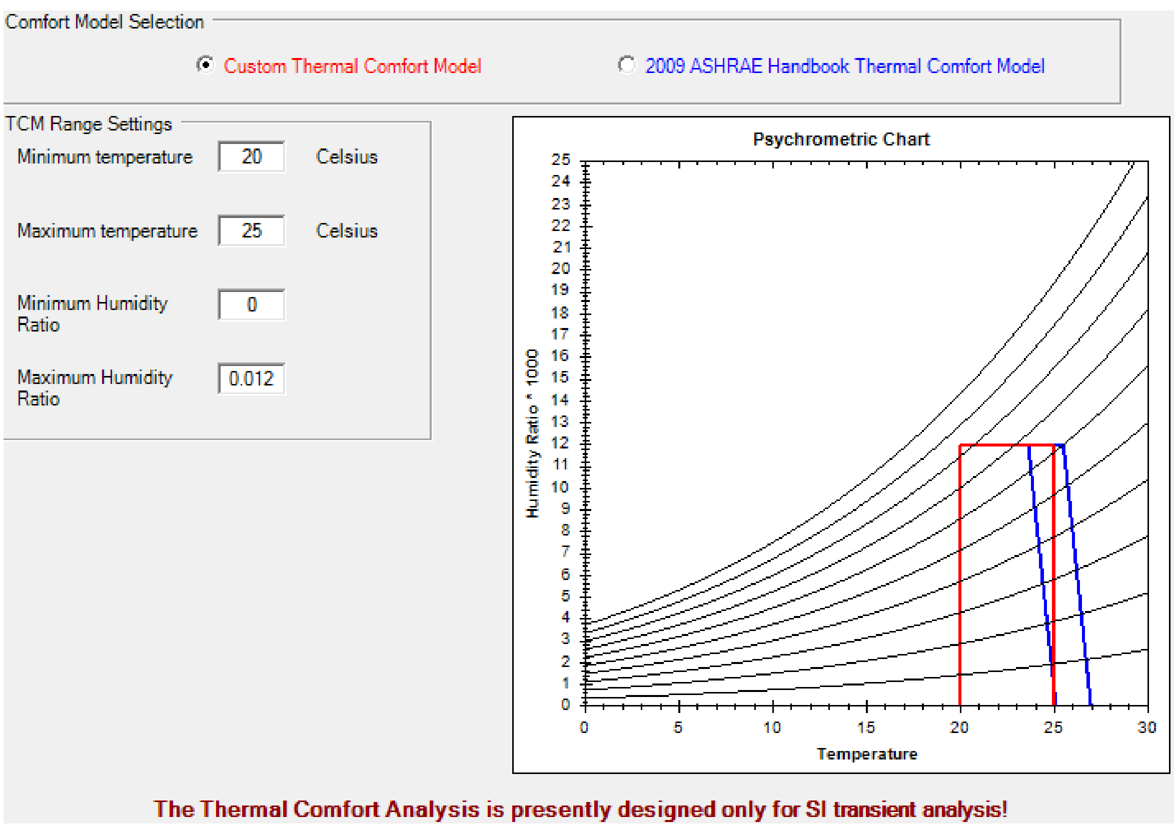


Figure 3-9. CoolVent input tab – Thermal comfort models

Input Name	Example value	Note
Comfort model selection	Custom Thermal Comfort Model	Custom or standard value
Minimum temperature	20°C	Defined by user in custom standard; no need to define if chosen ASHRAE standard
Maximum temperature	25°C	
Minimum Humidity Ratio	0 kg/kgda	
Maximum Humidity Ratio	0.012 kg/kgda	

Table 3-5. CoolVent input tab – Thermal comfort models

3.2 Calculation Method

CoolVent's calculation is based on a multi-zone model [21]. Different zones are represented by different nodes and there is a resistance (internal or exterior) connecting them. Although the newest version of CoolVent is able to include thermal stratification of a room, the previous well-mixed assumption is enough for the content covered in this thesis. Hence, we do not consider thermal stratification in this section. Several key calculation principles are summarized below.

3.2.1 Airflow Model

$$F_{ij} = C_d A \left(\frac{2\Delta P_{ij}}{\rho}\right)^n; \quad (1)$$

F_{ij} is volume flow rate of air

C_d is discharge coefficient of the opening

A is area of the cross section

ΔP_{ij} is the calculated pressure difference between zone i and zone j

ρ is density of air

n is a constant, which is assumed to be 0.5 for large openings

Pressure difference is calculated using the following equation:

$$\Delta P_{ij} = \left[P_{s,i} - \rho_i g(h_o - h_i)\right] - \left[P_{s,j} - \rho_j g\left(h_o - h_j\right)\right] + C_p \frac{\rho h^2}{2} sign(v)\,; \quad (2)$$

P_s is the static pressure

h is elevation relative to ground

v is the free wind velocity

$sign(v)$ is positive if air flows from zone i to zone j

3.2.2 Energy Conservation Model

The energy conservation equation applies to air inside a room. Change of its temperature depends on two factors: temperature and flow rate of airflow with other zones and internal heat gain. Only sensible heat gain is considered. The equation that describes this process is:

$$\rho_i c_{v,air} V_i \frac{dT_i}{dt} = \sum F_{ji} c_{pair} T_j - \sum F_{ij} c_{pair} T_i + Q_{solar} + Q_{TMi} + Q_i + Q_{ac}\ ; \quad (3)$$

V_i volume of room i

F_{ji} is mass flow rate from zone j to zone i

Q_{solar} is solar energy transfered through window

Q_{TMi} is heat gain through thermal mass (heat conduction)

Q_i is internal heat gain (utility, occupants)

Q_{ac} is air conditioning energy (positive if heating)

In order to simulate this process discretely in computer, a numerical method is used to turn the continuous equation to a discrete one. An average value of the explicit and implicit solutions is as follows:

$$\rho_i c_{vair} V_i \frac{T_{i,t+1} - T_{i,t}}{\Delta t} = \frac{1}{2}\Big(\sum F_{ji} c_{pair} T_{j,t} - \sum F_{ij} c_{pair} T_{i,t} + \sum_x h_{x,i} A_{x,i}\big(T_{TMx,i} - T_{i,t}\big) + Q_i + Q_{ac} + Q_{solar}\Big) + \frac{1}{2}\Big(\sum F_{ji} c_{pair} T_{j,t+1} - \sum F_{ij} c_{pair} T_{i,t+1} + \sum_x h_{x,i} A_{x,i}\big(T_{TMx,i} - T_{i,t+1}\big) + Q_i + Q_{ac} + Q_{solar}\Big); \quad (4)$$

3.2.3 Thermal Mass

Each piece of thermal mass is divided into several layers (shown in the figure below) with a specific resistance to get a more accurate solution. The exterior layer is involved in an exchange with room or outdoor air.

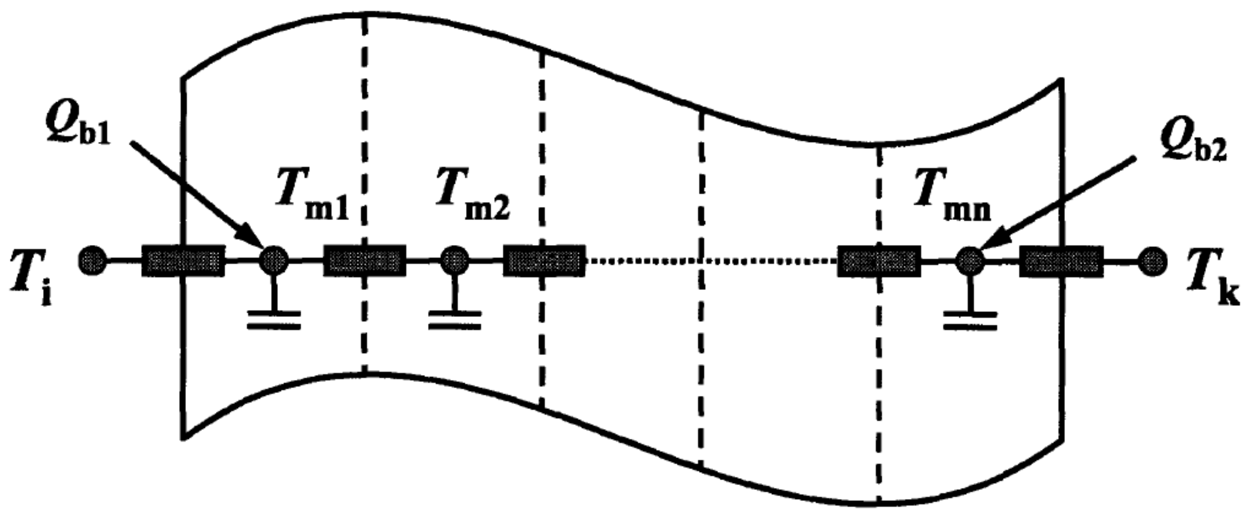


Figure 3-10. Multi-layer thermal mass modeling network

The process could be described by the following equation:

$$Q_{TMi} = \sum_x h_x A_x (T_{TMx} - T_i) \ ; \tag{5}$$

Q_{TMi} is heat gain through thermal mass, same parameter in 1.2.3

h_x is convective heat transfer coefficient

In order to simulate the heat exchange process discretely in computer, a numerical method is used to turn the continuous equation to a discrete one. An average value of the explicit and implicit solutions is as follows:

$$\rho_{TM} c_{\ pTM} \Delta x_i \frac{T_{i,t+1} - T_{i,t}}{dt} = \frac{1}{2}\left(\frac{T_{i-1,t} - T_{i,t}}{R_{i-1}/_2 + R_i/_2} - \frac{T_{i,t} - T_{i+1,t}}{R_i/_2 + R_{i+1}/_2}\right) + \frac{1}{2}\left(\frac{T_{i-1,t+1} - T_{i,t+1}}{R_{i-1}/_2 + R_i/_2} - \frac{T_{i,t+1} - T_{i+1,t+1}}{R_i/_2 + R_{i+1}/_2}\right); \tag{6}$$

3.2.4 Air Conditioning Energy

Mechanical air conditioning is an important supplement to a naturally ventilated space. It can maintain indoor environment at comfortable level when outdoor condition becomes unsuitable for natural ventilation. Air conditioning energy (Q_{ac}) is a part of the energy conservation calculation.

Air conditioning is turned on under pure air conditioning mode or hybrid mode (a combined use of natural ventilation and mechanical cooling). If indoor air temperature exceeds comfort limit (which is typically 26°C for a conditioned room in summer), then this module is turned on and cooling load is calculated and added up to the energy balance equation. In summer, no air conditioning will be used if indoor temperatures of all zones fall within comfort limit. When air conditioning is being used, this calculation is applied to zones with air temperature outside the comfort limit (>26°C); the air conditioning is zero if zone temperature is within the comfort limit (<26°C).

It is needed to calculate cooling load for a room at a specified time. One important approximation we made here is expressed in the equation below. Previous temperature is used to calculate cooling energy required for current period.

$$Q_{ac,i,t} = \rho_{air} V_i c_{v,air} \left(T_{limit} - T_{i,t-1}\right); \quad (7)$$

$Q_{ac,i,t}$ cooling load of room i at time t

T_{limit} upper comfort limit

$T_{i,t-1}$ air temperature of room i at time $t-1$

3.3 Verification of CoolVent

3.3.1 Natural Ventilation Case

Natural ventilation module of CoolVent has been tested against three types of data: CONTAMW, steady state CFD and transient field monitoring results. A detailed description could be found in previous work [19]. The building model used is a naturally ventilated building in Luton, UK. Figure and detailed information of the building is attached below.

Parameter	Value
Building type	Central atrium type

Number of floor	3F
Dimension	25 m × 17.2 m × 14.5 m
Glazing ratio	45% (S and N)
Thermal mass	Concrete slab 10 m thick

Table 3-6. Basic building information

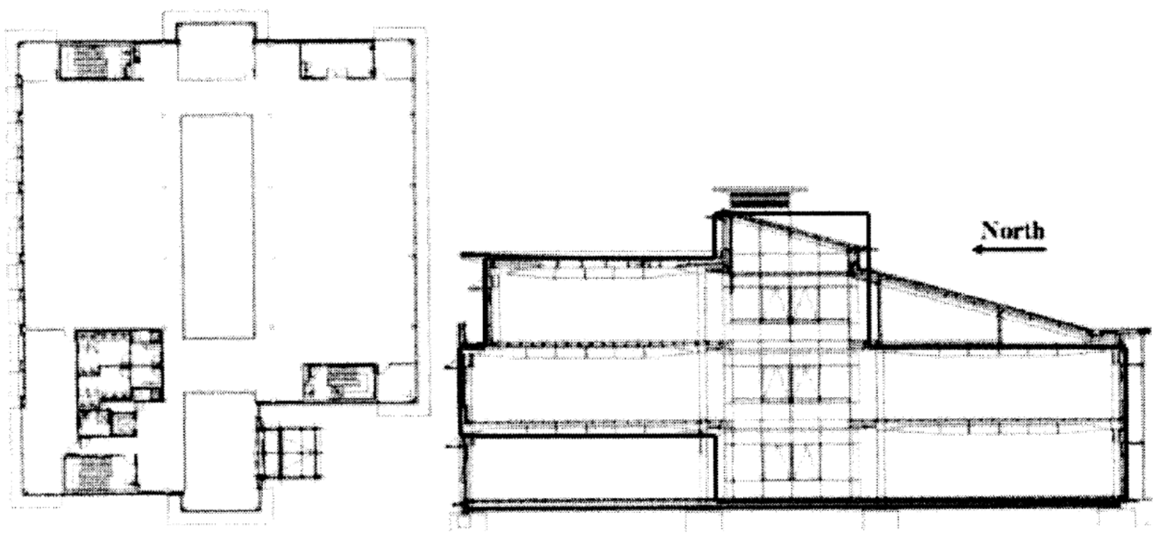
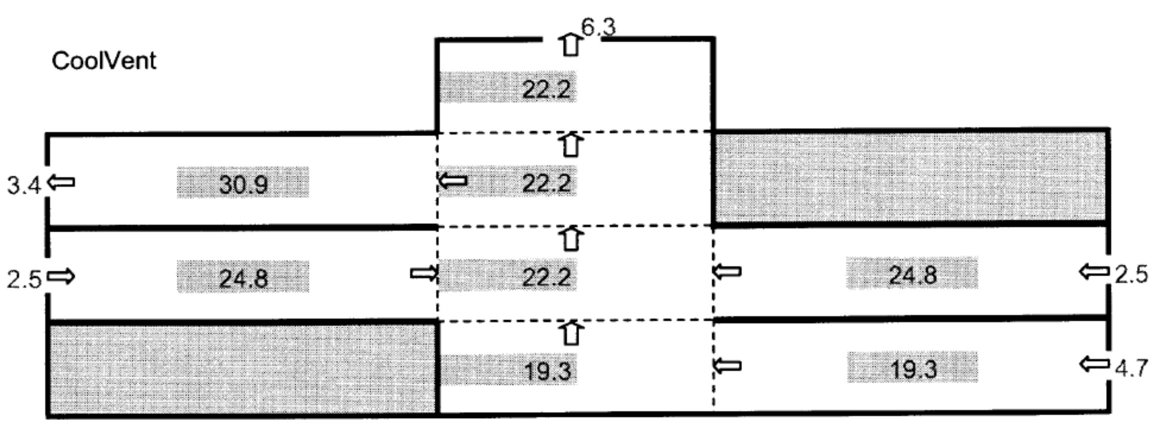
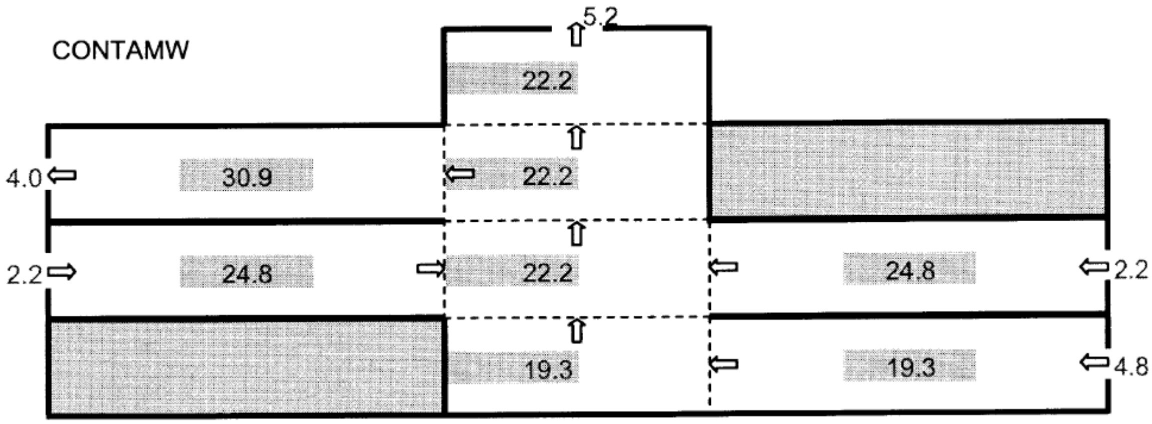


Figure 3-11. Image of tested building (defined as Luton type building)

The figure below shows comparison between CoolVent and CONTAMW. It should be noted that considering CONTAMW could not conduct a coupled simulation, simulated temperature from CoolVent is used as input for CONTAMW and thus inputs into these two programs are identical. Results show that results from these two programs are very close to each other while the maximum disagreement is only 13%.



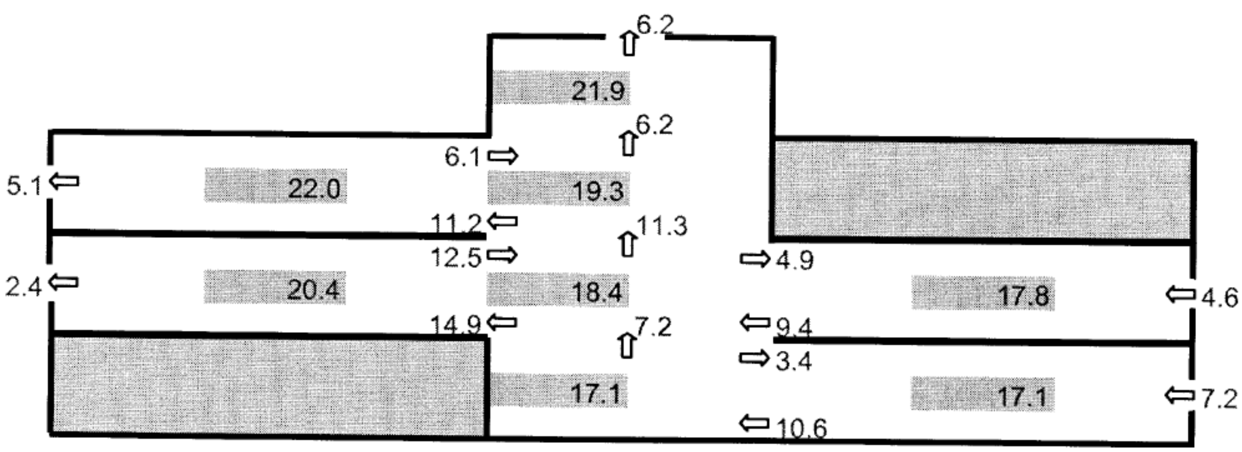
(a) CoolVent



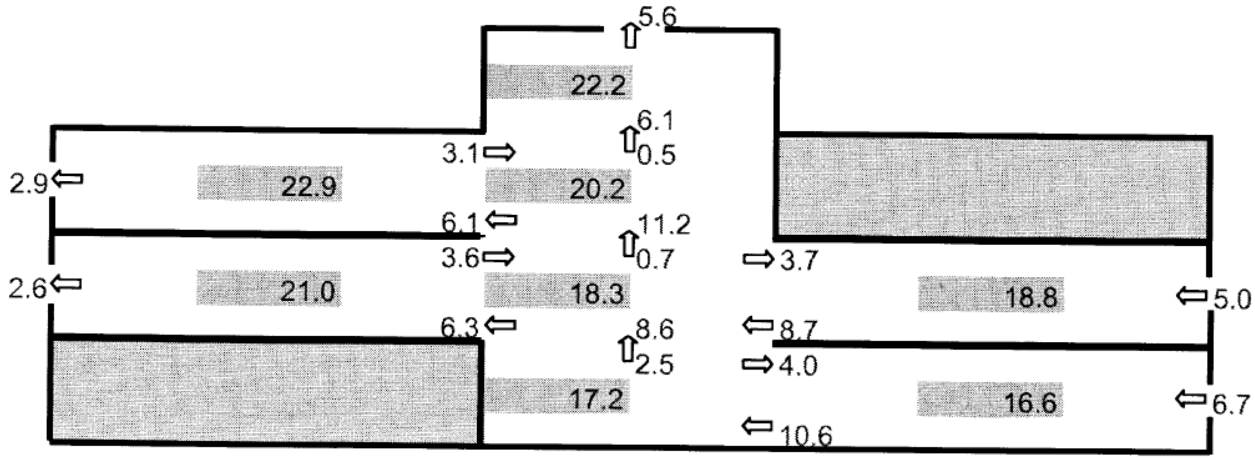
(b) CONTAMW

Figure 3-12. CoolVent vs. CONTAMW simulation results for a Luton building

The figure below shows comparison between CoolVent and CFD. It shows that there is good agreement for temperature predictions between these two programs: difference is mostly within ±1°C. Air flow rate predicted by CoolVent is also within 20% away from the CFD result in most cases. One exception is result on the upper-left part of the model, which is caused by the well-mixed and unidirectional flow assumption of CoolVent.



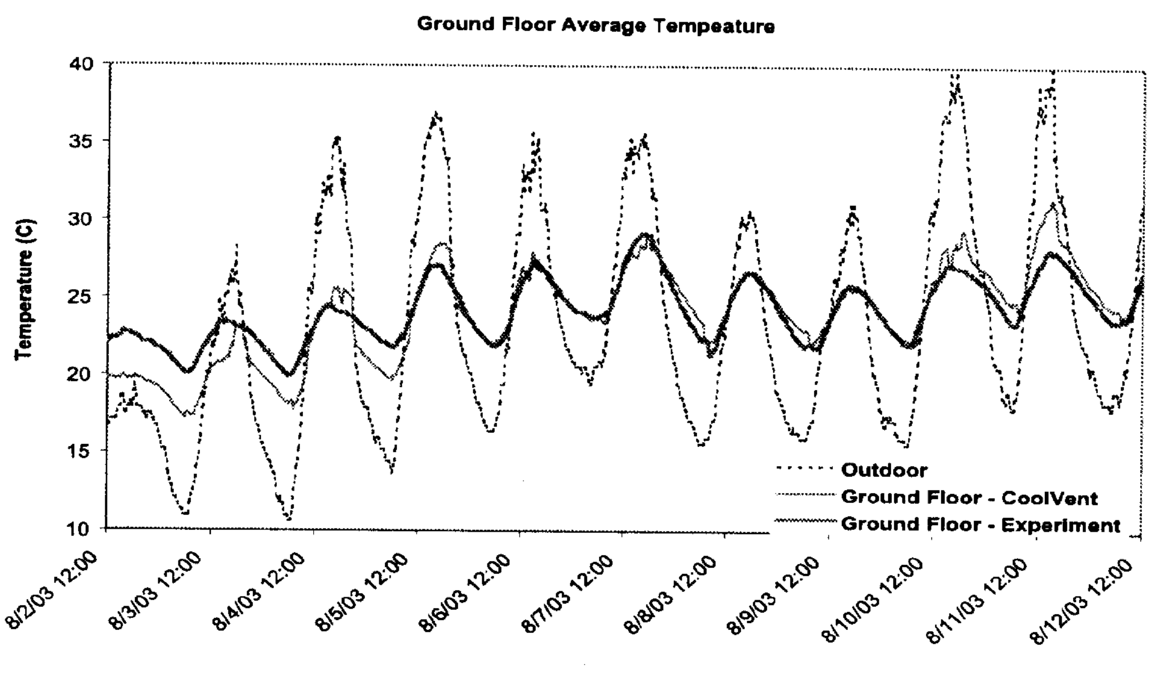
(a) CoolVent



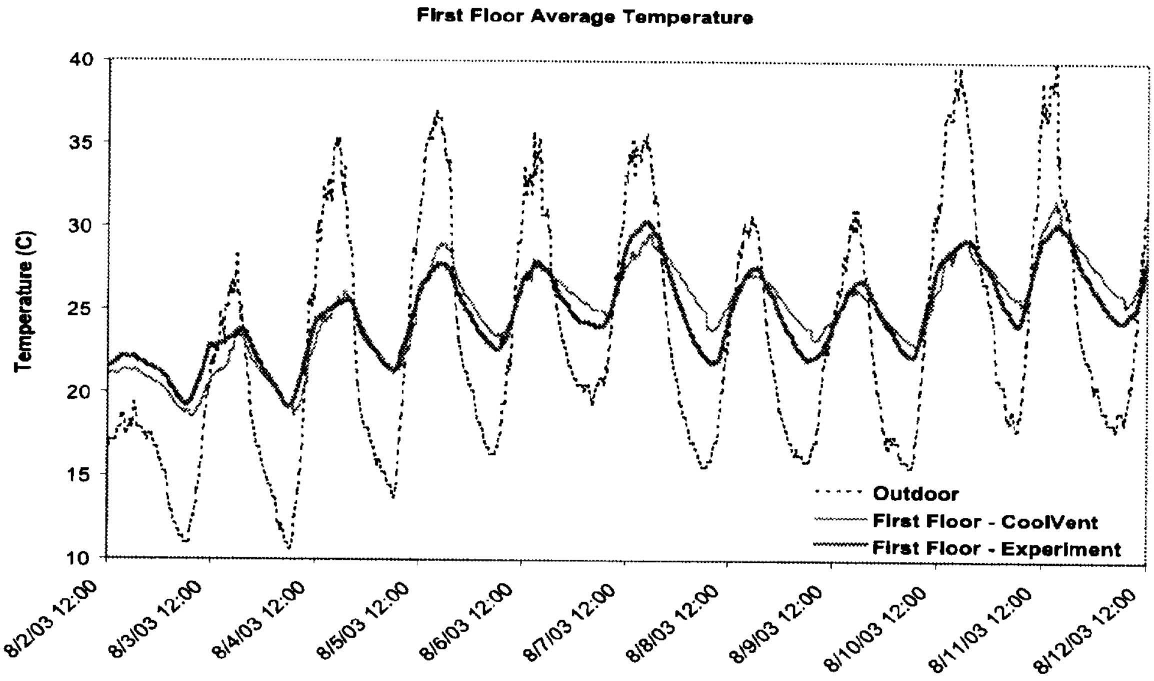
(b) CFD

Figure 3-13. CoolVent and CFD simulation results summaries for Luton building

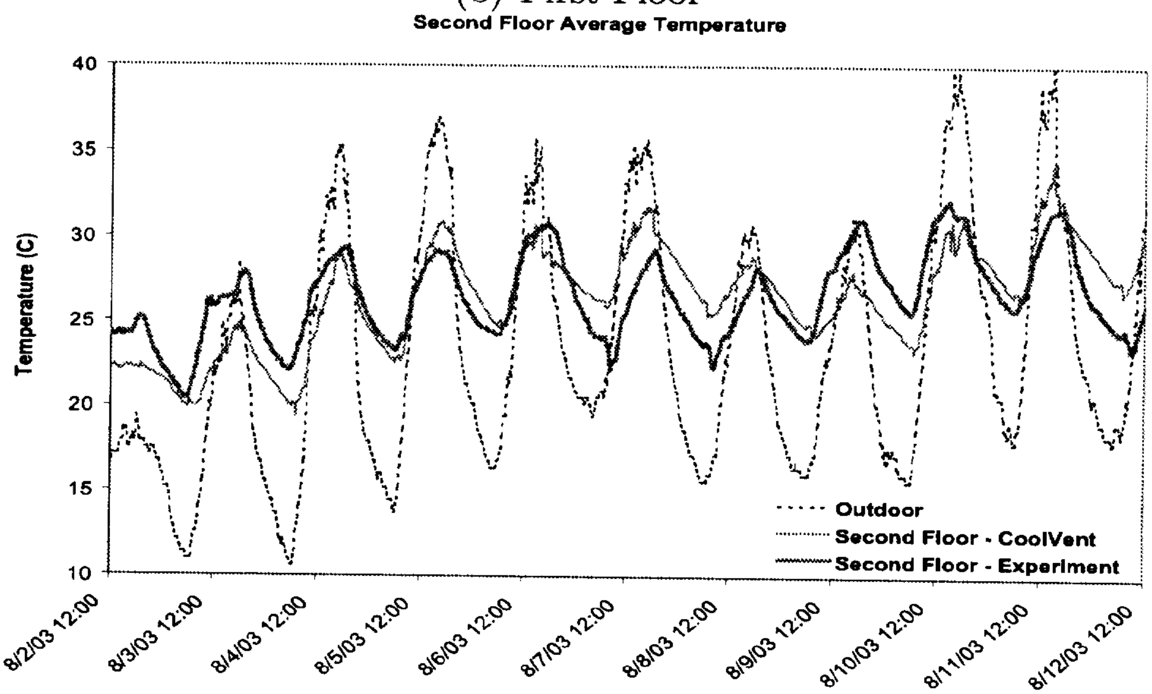
It has also been verified with monitoring results from a natural ventilation building. Inputs of CoolVent are generated based on actual operation of the building and simulated and monitored air temperatures of each floor are compared and in good agreement. The figure below shows the comparison.



(a) Ground Floor



(b) First Floor



(c) Second Floor

Figure 3-14. The floor-average temperatures in Luton building

3.3.2 Air Conditioning Case

This study integrates CoolVent with another program developed by MIT building technology program – Design Advisor. A detailed description of this program could be found in previous work [24], in which the energy calculation results of the MIT Design Advisor have been verified (this verification is briefly listed below). By comparing energy calculation results from CoolVent and Design Advisor, the effectiveness of CoolVent's can be verified.

First verification of the Design Advisor against EnergyPlus is presented below. The chart below shows basic information of the building model being tested.

Parameter	Value
Weather type	Boston, MA
Façade orientation	East
Dimension	5 m × 5 m × 3 m
Thermal mass	Concrete
R-value of wall	4.6 m^2K/W
Convection coefficients	Indoor surface to air: 4 $W/m^2.K$ Outdoor surface to air: 14 $W/m^2.K$
Schedule	7am to 8pm
Internal heat gain	6 W/m^2
Ventilation rate when occupied	3.6 ACH

Table 3-7. Chart Basic information of tested building

Simulated results of monthly thermal load from MIT Design Advisor and Energy Plus are compared in the figure below. It shows that simulation results from Design Advisor are very close to that from Energy Plus. From January to May to thermal load calculated by design advisor is slightly higher than that from the Energy Plus. However during October and December the thermal load from Energy Plus is slightly higher. The relative difference is mostly within 10%, as it is shown in Figure 3-15.

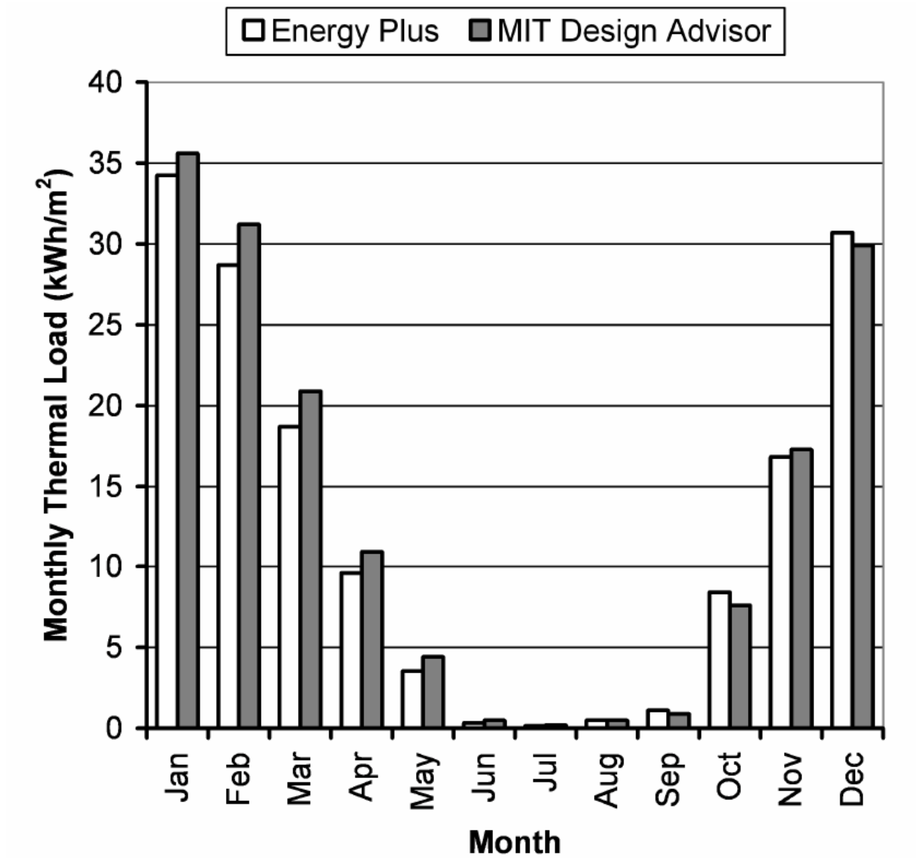


Figure 3-15. Monthly comparison between Energy Plus and MIT Design Advisor

The model building we use here is a five floor central atrium type building (basic information and figure of this model in CoolVent are shown below). This building model in CoolVent is simulated under the pure air conditioning mode. All windows are closed all the time; meanwhile there is no fan in operation. We also constructed an identical building model in Design Advisor to run simulation in both programs.

Parameter	**Value**
Building type	Central atrium type
Heat source level	30 W/m^2
Number of floors	5
Operable windows per façade per floor	3 m^2
Floor area	2100 m^2

Table 3-8. Basic information of test building

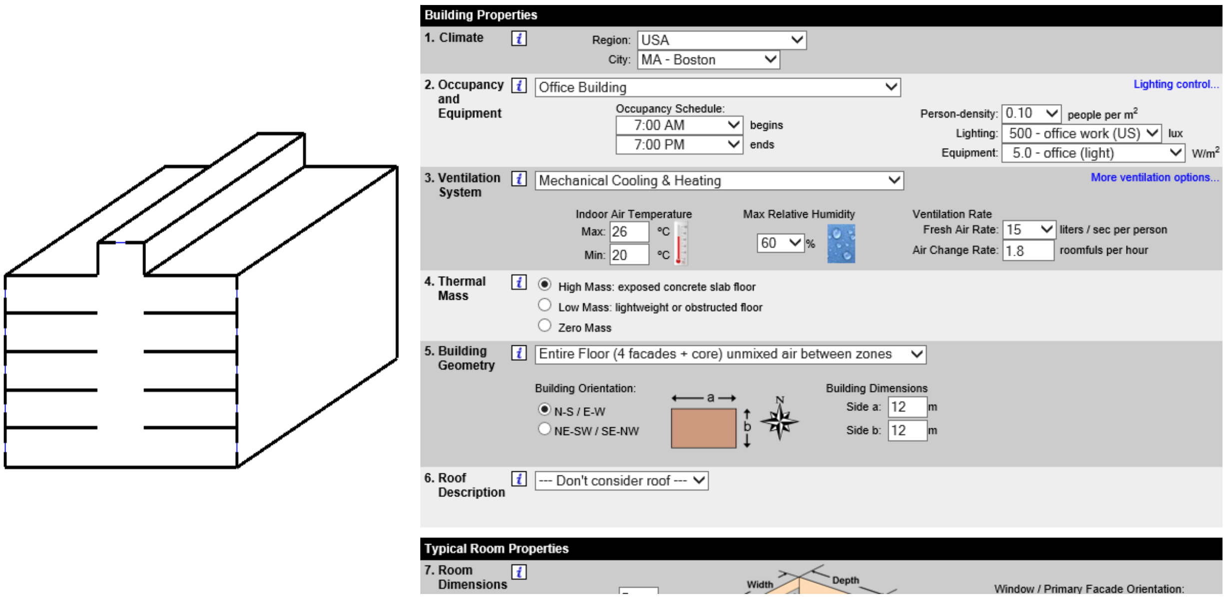


Figure 3-16. Building model in CoolVent and Design Advisor

Several important parameters that need to be in accordance are: building geometry, insulation level and internal heat gain. Having ensured that these values are same in both program, we derived simulation results of annual cooling energy (thermal energy, in kWh/m^2) for four typical cities where natural ventilation is potentially a good option. The figure below shows the comparison.

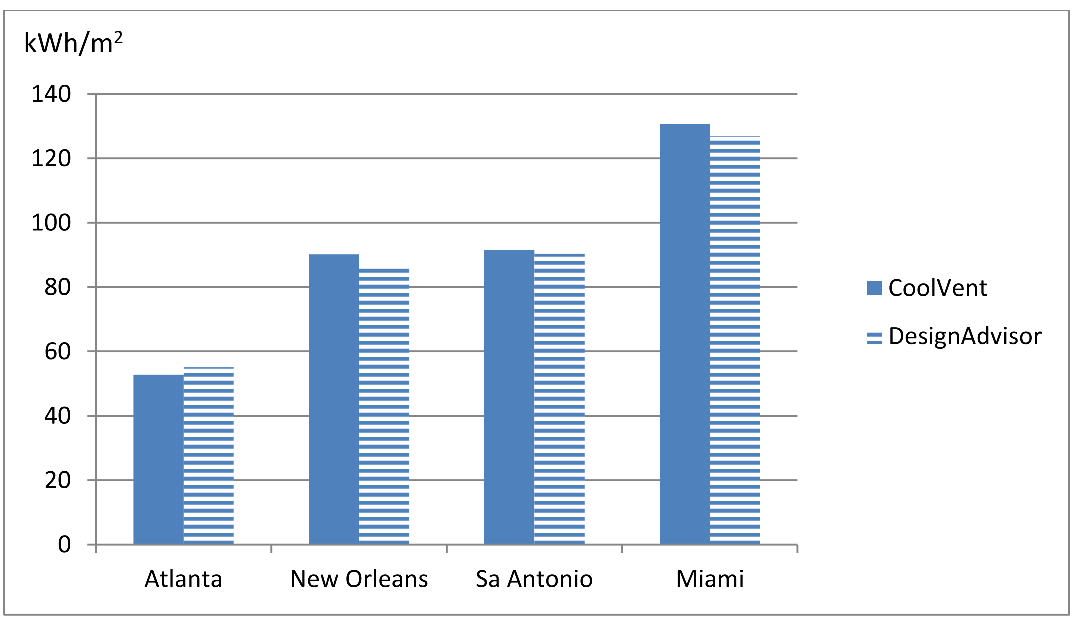


Figure 3-17. Annual cooling energy comparison between CoolVent and Design Advisor

It could be seen from the figure that results from these two programs are very close to each other. Hence, the effectiveness CoolVent's energy calculation function is verified against Design Advisor. It should also be noted that deviation between two series becomes larger if cooling energy is trivial (for example, in some cool area such as Fairbank, Los Angeles). However, this deviation will not affect our simulation of hybrid mode ventilation because weather climates are mainly between moderate to warm for natural ventilation simulation (such as the four cities given above).

3.4 Efficiency of CoolVent

It is important to improve the efficiency of CoolVent while keep its result accurate. In this section, we will test simulation of CoolVent under two different time steps (30 s and 90 s) and compare simulation speed and relative difference between them for a purely natural ventilation building. Again we use a five floor central chimney building (the same one in 3.3) as the model building and conduct the test in a purely natural ventilation case.

For a cooling season (April to September) simulation with a time step of 90 s, it takes 4 min and 10 s to complete the simulation. Weather data is on an hourly basis for the entire period. If we change the time step from 90 s to 30 s, this time will be increased to 12 min. It is almost three times of the 90s time step (because there are three times steps).

Time step / s	Time elapsed (Apr. – Sept.)
30	12 min
90	4.2 min

Table 3-9. Comparison between different simulation steps

Next step would be comparing the relative difference between these two steps. The 30 s case is chosen as base and absolute deviation of the 90 s case will be calculated by the following equation:

$$\emptyset = \frac{\sum_1^n \left| t_{i,90s} - t_{i,30s} \right|}{n};$$

$\emptyset$ is mean absolute deviation between 90 s and 30 s time step

$t_{i,90s}$ is temperature of 90 s case at time spot i

$t_{i,30s}$ is temperature of 30 s case at time spot i

n is the number of total time spots

Mean deviation is calculated to be 0.3°C. This suggests that we use 90 s as the time step instead of 30 s because simulation speed could be greatly enhanced without causing a large deviation.

4 SIMULATION IN COOLVENT

4.1 DOE Benchmark Database

To estimate the potential of energy saving, we need statistical data of building types of the country. This includes building category, geometry, floor area, internal load, occupants and other parameters related to simulation.

The DOE Commercial Building Benchmark Models [22] is a set of standard benchmark building models for new and existing buildings. This model is representative of shapes, thermal zoning and operation of real buildings and serves as a good series of model to conduct simulation.

There are 15 benchmark buildings representing commercial types; these buildings are set to be located at 16 different locations (covering all climate zones in the US); standards in three time periods are considered (pre-1980, 1980-now and new). A Detailed introduction to this database could be found at a paper.

In order to reduce computational burden, we select some representative building types and climate zones from the entire database. To select representative building type, the average area weight factor (average value for ten representative cities across the country) is calculated and listed below.

Building type	Weight factor	Cumulative weight factor
Standalone retail	29.5	29.5
Warehouse	12.0	41.5
Midrise apartment	10.3	51.8
Quick service restaurant	10.3	62.1
Strip mall	9.8	71.9
Medium office	5.9	77.8

Table 4-1. First six building types that have largest averaged weight factor of floor area

Although the warehouse which might be suitable for natural ventilation, we will limit our scope to the remaining five building types. It is necessary to create corresponding building models in CoolVent. Several important building parameters that are important:

building geometry, insulation level, occupants and internal load. Ten cities are selected from the sixteen cities. They are Atlanta, Chicago, Duluth, Fairbanks, Houston, Los Angeles, Miami, Minneapolis, Phoenix and Seattle. Weather data is derived from DOE's website.

It should be noted that the DOE benchmark data base is not appropriate for analysis at the state level because "the datasets used to generate the models and the weighting factors are too small to form a valid statistical model at this level". Instead, "the reference building models will be used for DOE commercial buildings research to assess new technologies including ventilation".

4.2 Pure Natural Ventilation

4.2.1 Principles

In this part, performance of pure natural ventilation will be evaluated. The building will be constantly operated under natural ventilation mode. To estimate suitability of natural for different building type under different climate types, the percentage of hours during which comfort standard is met will be calculated.

Several major assumptions concerning this mode are:

(1) All windows will be kept opening all the time;

Openness of all windows is set to be 1 all the time.

(2) Natural ventilation will be assisted with fan;

Natural ventilation with and without fan will both be simulated and compared.

Rated flow rate of the fan is decided on an approximation method. Take medium office type as an example, total floor area of this building is about 5000 m^2 and room height is 3 m. If we assume an air change rate per hour (ACH) of 5, required air flow rate and rated flow rate of the fan could be decided by using the following equation:

$$5000m^2 \times 3m \times \frac{5}{h} = \frac{75000m^3}{h} = 20.8\, m^3/s$$

Available fan curves in CoolVent are shown in the figure below. The fan will be placed in the top opening and number of the fan could be adjusted by changing area of the top opening (figure showing building geometry attached below). For the previous calculated value (20.8 m^3/s), we choose to put two medium-duty fans on the top opening. Airflow rate will be affected by the fan as well as buoyancy and wind effect. The fan is expected to provide steadier and more sufficient air flow through the building.

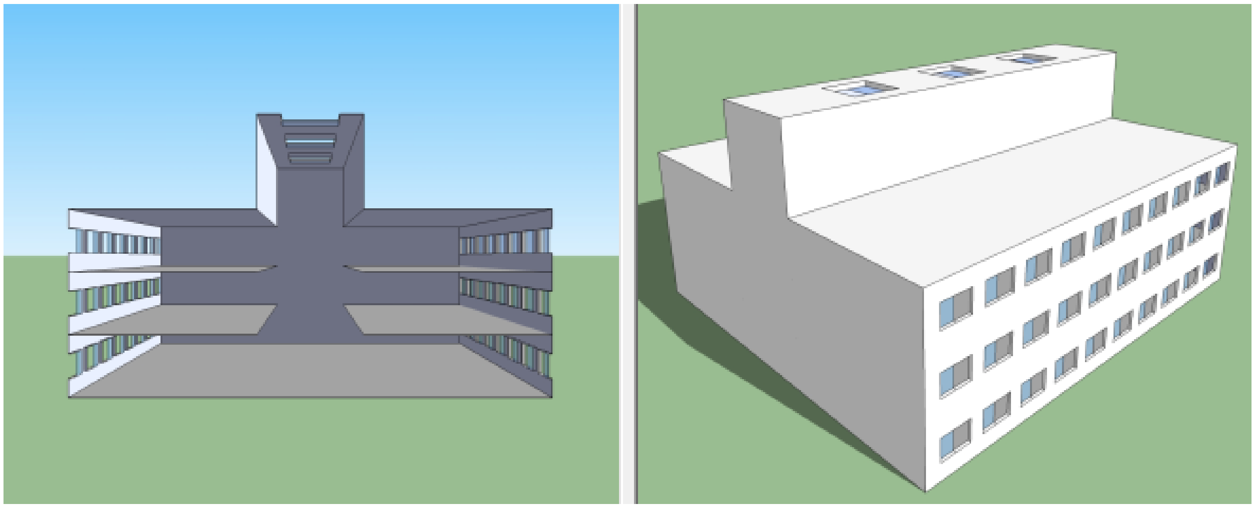


Figure 4-1. Building geometry for top opening and side window

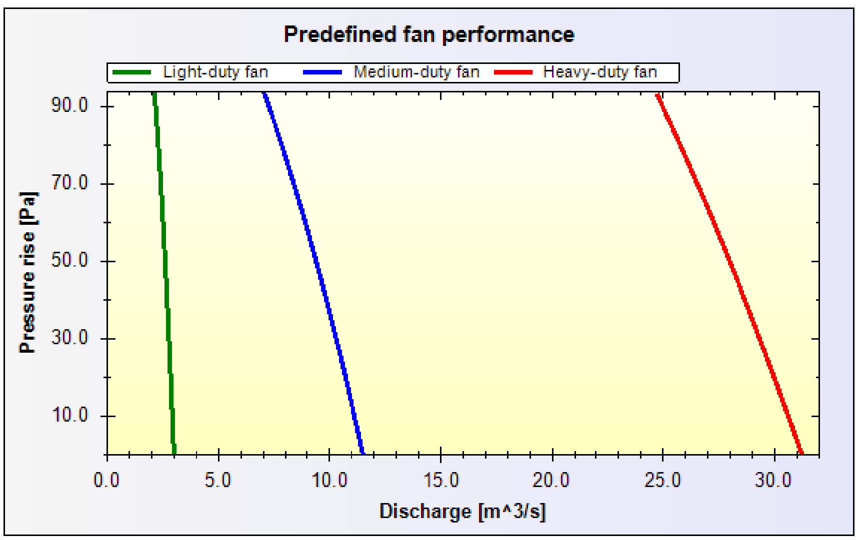


Figure 4-2. Available fan curves in CoolVent

(3) Only occupied hours will be considered;

Again take the medium office as an example, during a typical cooling season (May to October), there are 107 working days and office hour of each day is 8 hours. The total hours is calculated to be 856 hours. The comfort percentage is derived by dividing comfortable hours by this total hour. The definition of comfortable hours will be covered in the following part. The chart below shows occupied hour of each building type.

Building type	Occupied from	Occupied till	Cooling season total (May to October)
Medium office	9am	5pm	1224
Middle rise apartment	6pm	6am (next day)	1836
Quick service restaurant	9am	10pm	1989
Standalone retail	9am	9pm	1836
Strip mall	9am	9pm	1836

Table 4-2. Statistics of occupied hours for each type

(4) Definition of comfortable environment;

As it has been summarized in the literature review section, previous research concerning thermal comfort in natural ventilation building has shown that occupants tends to accept higher temperature as neutral or comfortable if the building is naturally ventilated. Hence, comfort standard we use here is set to be 28°C.

Another important aspect affecting indoor comfort is humidity. Current version of CoolVent does not calculate mass transfer and gain. Considering the high air exchange rate in natural ventilation case, it might be reasonable to use the outside humidity ratio to represent the indoor one. However, the impact of internal gain has to be estimated. The simple calculation below shows this process.

For instance, assume outdoor humidity ratio to be 10 g/kgda, with an ACH of 5, floor area 5000 m^2, 270 occupants, humidity load 80 g/h per person. Change of humidity ratio of air ventilated is:

$$\frac{(80g/h) \times 270}{(5/h) \times 5000m^2 \times 3m \times (1.2kg/m^3)} = 0.24g/kgda$$

Compared with the base value (10 g/kdga), this addition of 0.24 is negligible. Results are similar for other building types. Hence, this substitution is reasonable and easy to implement. Upper comfort limit for humidity ratio is 14 g/kgda.

(5) No mechanical system except fan;

Mechanical system (such as air-conditioning system, outdoor fresh air system, dehumidification system) will not be included in this simulation. Their affection will be analyzed in following sections.

4.2.2 Verification of Fan

To ensure the effectiveness of this simulation, it is necessary to test flow rate of the fan. This is because there might some inconsistency between flow resistance and fan curve which causes deviation of actual flow rate from the design value.

Take the figure below as an example, it shows simulation in CoolVent of the medium office model. Typical design of natural ventilation in buildings has an air change rate (air flow volume per hour divided by building volume) of around 5 [23]. Hence, the design value is set to be 5 ACH, which corresponds to an airflow rate of 20.76 m^3/s, while the simulated flow rate at a certain moment is 22.92 m^3/s.

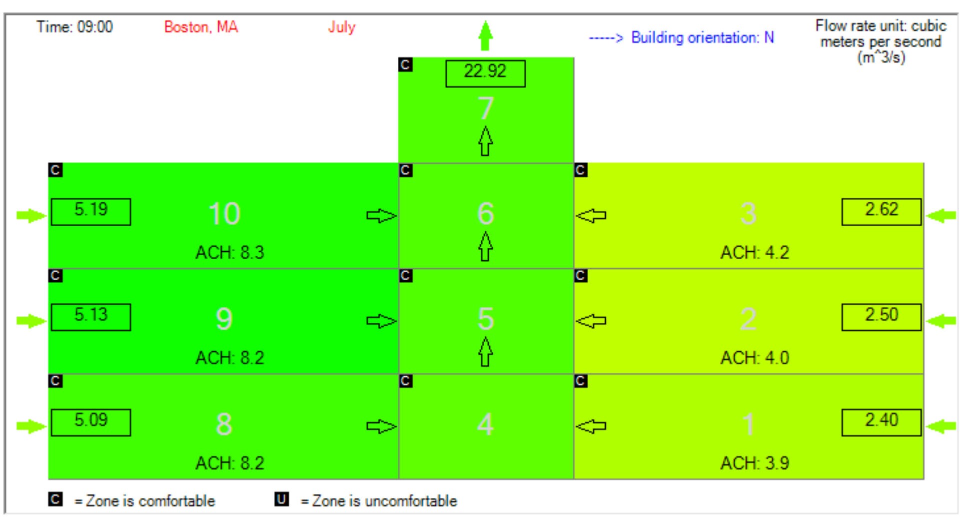


Figure 4-3. Simulation in interface of medium office

The table below summarizes this verification process. The average, maximum and minimum values of simulated airflow are shown in comparison with the design value. It shows that the fan is properly sized and placed.

	Design (m^3/s)	Avg. (m^3/s)	Max (m^3/s)	Min (m^3/s)
Medium office	20.8	23.2	23.5	22.8
Middle rise apartment	13.1	11.6	11.8	11.4
Quick service restaurant	5.1	5.7	5.8	5.7
Standalone retail	10.0	11.5	11.7	11.4
Strip mall	9.7	11.7	12.0	11.5

Table 4-3. Verification of fan operation

4.2.3 Simulation Results

Two building types are selected for preliminary analysis: medium office and middle rise apartment. They are simulated under different climate zones for the entire cooling season (May to October) and comfort conditions during occupied hour are analyzed.

For building types which has a number of sub-areas, the results are presented in two ways: by total area and by sub area. Definition of "by total area" is that if any sub area cannot meet comfort standard, the entire building at that moment will be regarded as not suitable for natural ventilation. Definition of "by sub area" is a percentage derived from average each sub area, the building might be "partly" suitable for natural ventilation. Tables below show comfort distribution for medium office.

Medium office

Parameter	Value
Ventilation type	Central atrium
Floor area	4982 m^2
Number of floor	3
Internal gain	20 W/m^2
Occupancy schedule	9am to 5pm

Table 4-4. Basic information of medium office

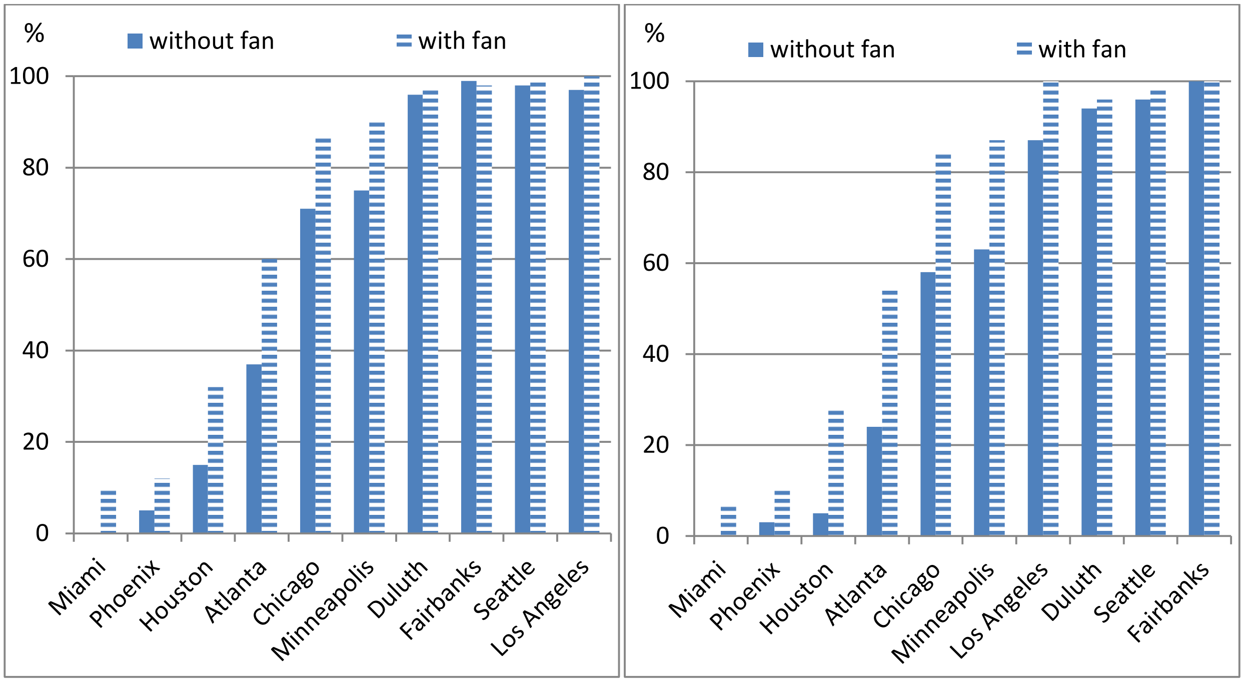


Figure 4-4. Comfort percentage of medium office by sub area and total area

Mid-rise apartment

Parameter	Value
Ventilation type	Central atrium*
Floor area	3135 m^2
Number of floor	4
Internal gain	12 W/m^2
Occupancy schedule	6pm to 6am

(*Although this building type is not commonly seen in the design of apartment buildings, it is the most suitable building model currently available in CoolVent's database that could be used to estimate the potential.)

Table 4-5. Basic information of mid-rise apartment

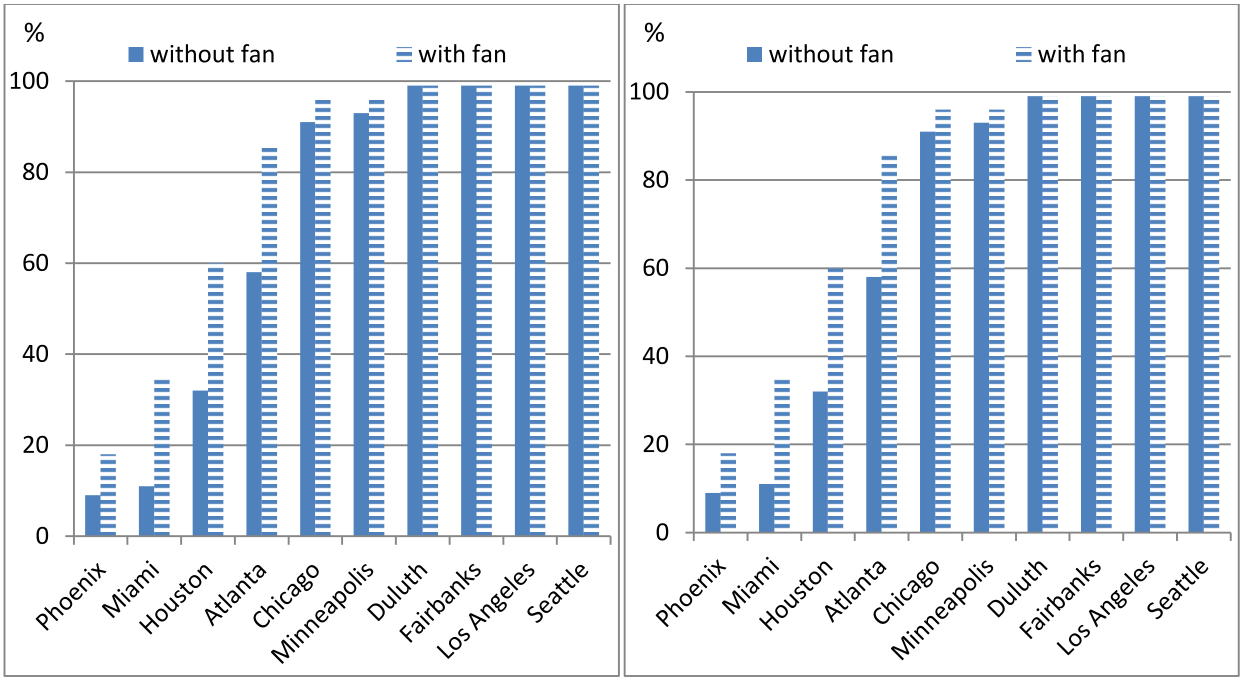


Figure 4-5. Comfort percentage of mid-rise apartment by sub area and total area

Quick service restaurant

Parameter	Value
Ventilation type	Chimney
Floor area	232 m^2
Number of floor	1
Internal gain	300 W/m^2
Occupancy schedule	9am to 10pm

Table 4-6. Basic information of quick service restaurant

Strip mall

Parameter	Value
Ventilation type	Cross Ventilation
Floor area	2090 m^2
Number of floor	1
Internal gain	45 W/m^2
Occupancy schedule	9am to 9pm

Table 4-7. Basic information of strip mall

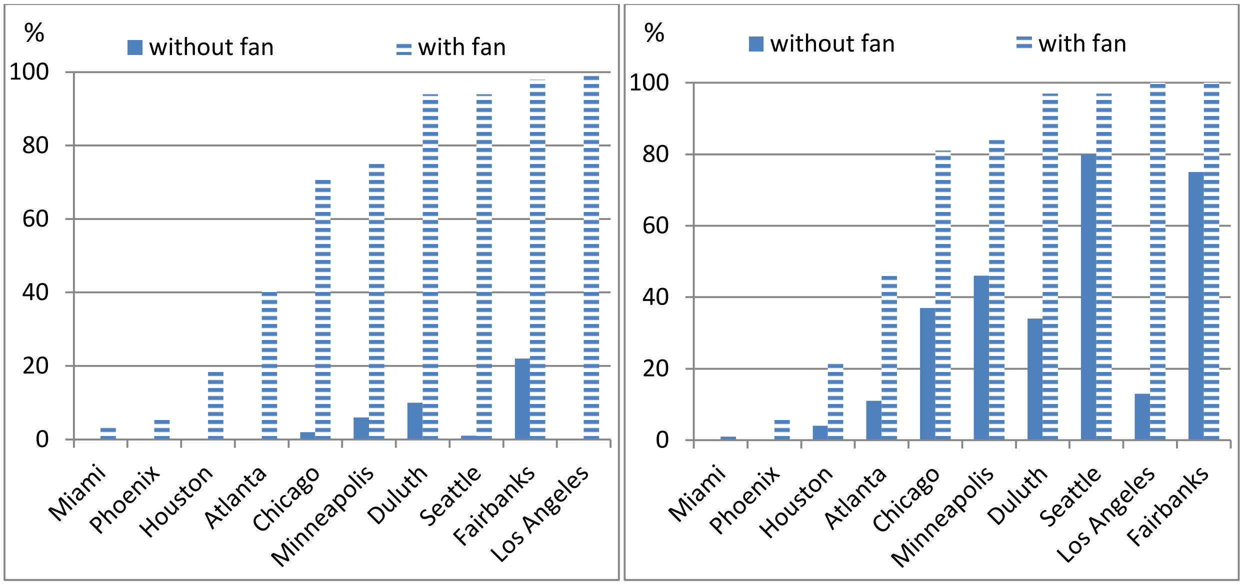


Figure 4-6. Comfort percentage of quick service restaurant and strip mall by total area

Standalone retail

Parameter	Value
Ventilation type	Central atrium
Floor area	2294 m^2
Number of floor	1
Internal gain	45 W/m^2
Occupancy schedule	9am to 9pm

Table 4-8. Basic information of standalone retail

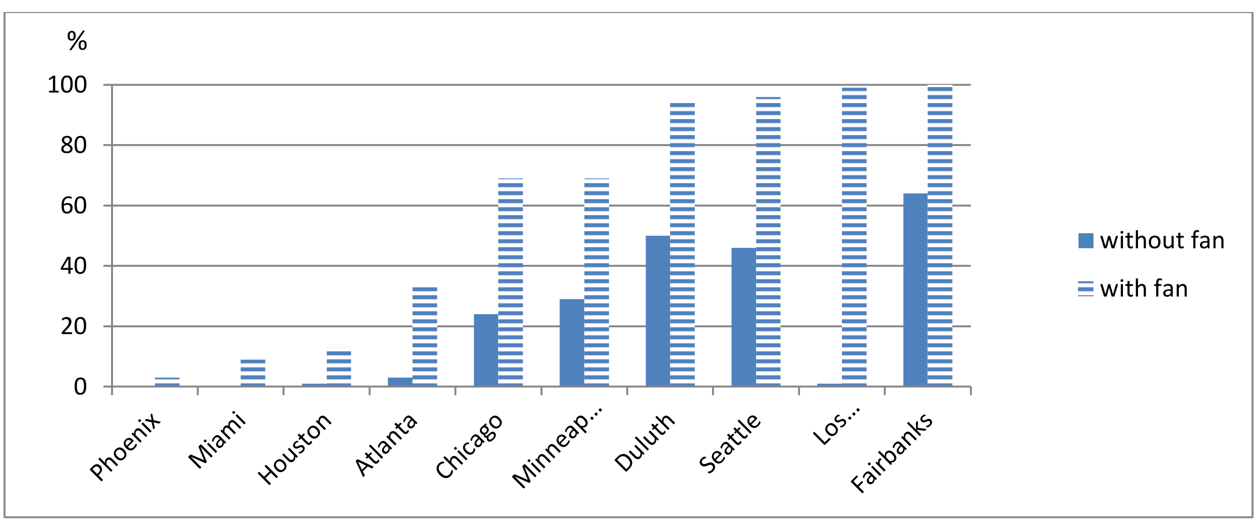


Figure 4-7. Comfort percentage of standalone retail by total area

4.2.4 Discussion

The five building types are simulated under the pure natural ventilation mode (with and without fan, using different standard). Results show that natural ventilation has different performance under different weather types.

It is obvious that comfort percentage under “by sub area” method is greater than that under “by total area”. The building will be considered uncomfortable under “by total area” method if only one sub area is uncomfortable. However, it will be considered partly comfortable under by sub area method and adds a value to the comfort percentage.

Mid-rise apartment has the highest comfort percentage among the five building type. This is because it has a low level of internal heat gain and its occupied schedule is during night time when outside is cool. Middle office has the lowest internal heat gain among the remaining three building types and thus a better comfort condition both with and without the fan. For Duluth, Fairbanks, Los Angeles and Seattle it is able to maintain a comfortable indoor environment for at 80% of the occupied hours. When it comes to Atlanta, Chicago and Minneapolis, the comfort percentage varies from 20% to 100%, which suggests a combination use of natural ventilation and air conditioning might be a better option. The potential for Phoenix, Miami and Houston is limited (maximum 60%, mostly between 0 to 30%).

The other three commercial building types have high internal gain and do not perform well under pure natural ventilation without fan. When a roof exhaust fan is added, its performance is significantly improved.

For the fan is in operation, it might be possible to predict comfort percentage in a much quicker way. Take the medium office as an example (model shown in the figure below). Internal heat gain is 25 W/m^2, floor area per area is 640 m^2, the ACH is maintained by fan at a rate of ACH 5. Upper limit for outside air temperature could be estimated by the following equation:

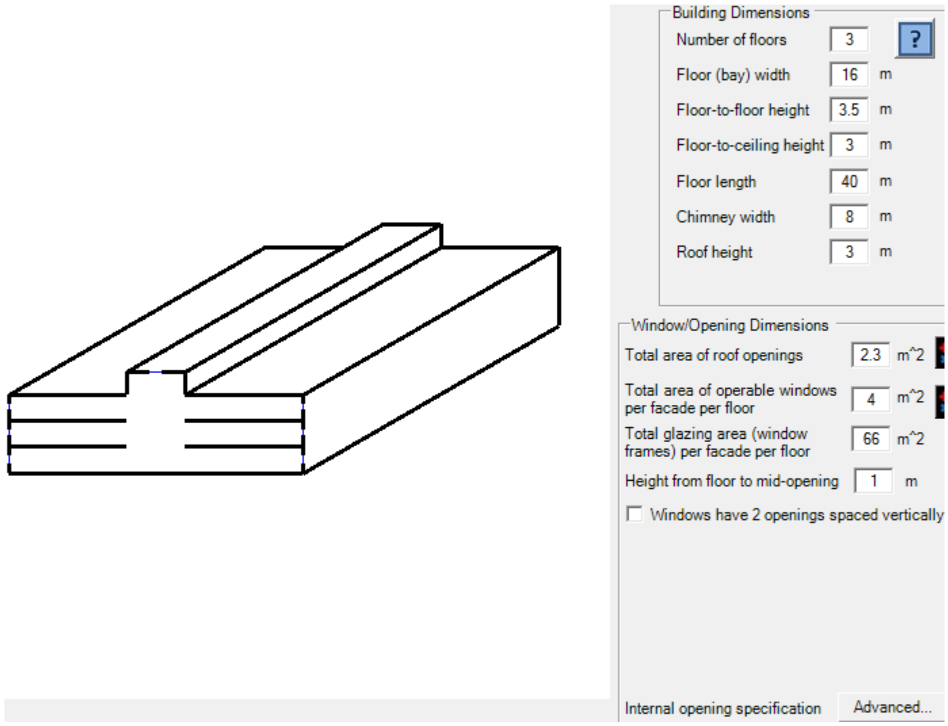


Figure 4-8. Building model for medium office

$$T_w = T_{limit} - \frac{Q_{i,gain}}{ACH \cdot V_{bldg} \cdot c_p \cdot \rho}$$

$$T_w = 28℃ - \frac{\left(25\ \frac{\mathrm{W}}{m^2}\right) \times 640\ m^2}{\left(\frac{3.5\ m^3}{\mathrm{s}}\right) \times \left(\frac{1000\ \mathrm{J}}{\mathrm{kgK}}\right) \times \left(\frac{1.2\ \mathrm{kg}}{m^3}\right)} \approx 24℃$$

Since some zones have zero heat gain but are counted in the calculation of airflow, airflow is averaged to be 3.5m^3/s through each opening on each floor. The corresponding temperature limit is calculated to be 24°C. A rough estimation of the percentage of time that the building is under comfort conditions could be derived by simply analyzing weather data. Result from this method is compared with the simulation results for four cities (Houston, Minneapolis, Atlanta and Chicago). It should be noted that only temperature is considered in the evaluation of the simulated results.

City	Weather data ($T_{outdoor}$<24°C)	Weather data ($T_{outdoor}$<28°C)	Max. night cooling simulation (upper indoor limit 28°C)	Zero night cooling simulation (upper indoor limit 28°C)
Atlanta	26	52	54	30
Chicago	56	92	84	62
Houston	9	30	28	16
Minneapolis	62	86	87	68

Table 4-9. Comparison between quick method and simulation results (medium office)

Simulated comfort percentage (column 4) is higher than value predicted (column 2) from the weather data. This is primarily because of the thermal mass. Current simulation assumes maximum night cooling, which might reduce more than half of the sensible cooling load at daytime (as shown in “results from night cooling case”). By closing windows and turning off roof fan when the building is not occupied, the case is simulated again to see the effect of thermal mass. It could be seen that (column 5) comfort percentage is reduced and close to the value of the quick estimation (column 2). This is because the window is not entirely closed and there is still an airflow rate calculated. The program cannot proceed with zero window openness or a zero airflow value. Hence, there is still some level of night cooling helpful in maintaining comfort at daytime. Quick weather data estimation with an upper limit of 28°C (Column 3) is a good approximation when night cooling is taken in account. Results in column 3 and 4 are close to each other.

4.3 Pure Air Conditioning

4.3.1 Principles

In this section, air conditioning load for different building types under different climate zones will be evaluated. The aim is to provide a baseline for further simulation on hybrid mode and control algorithm. The building will be operated under pure air conditioning mode. Several major assumptions of this mode are listed below.

(1) All windows will be kept closed all the time (Hence, no night cooling);

(2) Comfort standard is different from natural ventilation case;

Compared with comfort standard used in natural ventilation simulation (28°C for temperature and 14 g/kgda for humidity ratio), occupants in air conditioned environment tend to have stricter requirement on comfort standard. It is set to be 26°C for temperature and 12 g/kgda for humidity ratio.

(3) Latent load is considered;

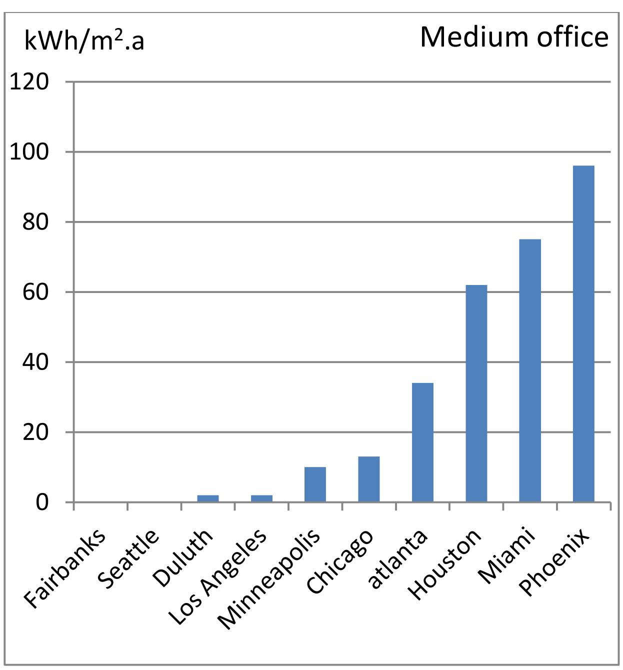
There are two sources of latent load: occupants and fresh air. The rate of latent load from occupants is a function of number of occupants. Rate of latent load from fresh air is calculated by multiplying air flow rate and humidity ratio difference.

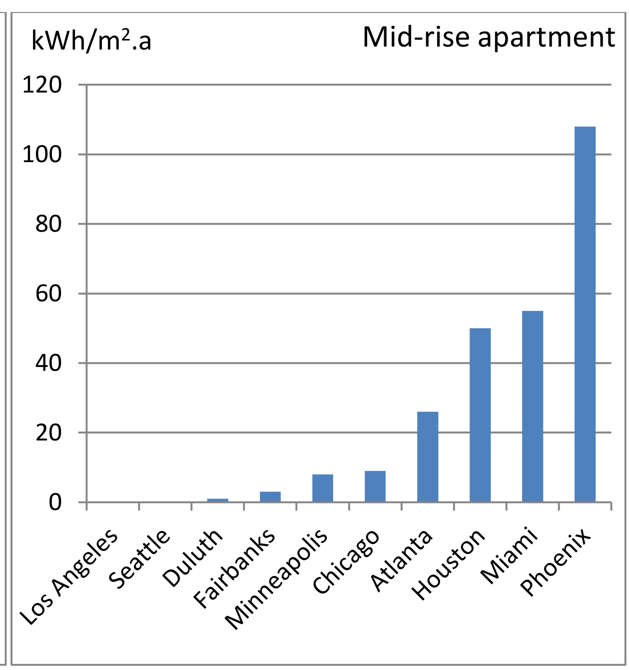
(4) All hours will be considered;

The building is assumed to be conditioned 24h per day. However, load will be affected by schedule hours of different building types.

4.3.2 Simulation Results

Total cooling energy (thermal energy instead of electricity) required for an entire cooling season is shown in figure below.





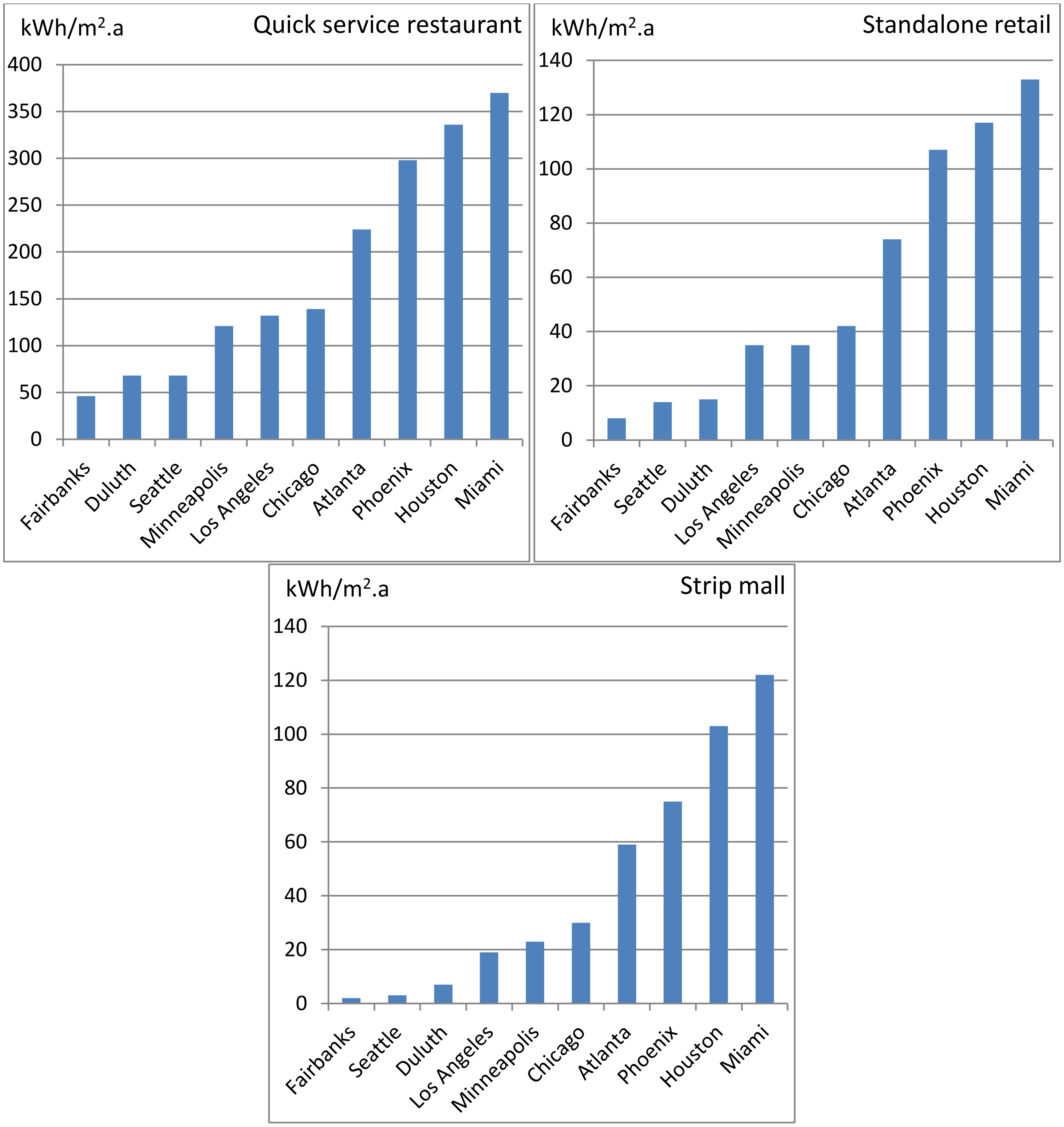


Figure 4-9. Cooling energy for five building types with only AC

4.3.3 Discussion

Results show that both building type and climate zone have an impact on calculated air conditioning load. Under all climate zones, quick service restaurant requires significant cooling energy because of its high internal load. For other building types, air conditioning load strongly depends on weather data. These calculation results provide a base value for further analysis.

4.4 Night Cooling

4.4.1 Principles

As it has been discussed in the literature review section, night cooling is a good option for reducing air conditioning load at daytime. By opening the building at night and ventilating the space with a fan, temperature of the thermal mass is reduced. Hence, thermal mass is capable of absorbing heat gain at daytime and make indoor space cooler without using air conditioning. Indoor space temperature might be lower than outdoor temperature even at daytime.

In this section night cooling will be simulated in detail. Building model used is standalone retail. Previous simulation shows that its air conditioning load is moderate, which is suitable for night cooling simulation. Its performance under three climate types, Atlanta, Phoenix and Minneapolis will be evaluated (representing moderate, hot and cool climate).

To account for temperature at night (it might be too high for ventilation), different combination of hours and months have been simulated. For example, by assuming midnight to be coolest, 23:00 to 1:00 might be most suitable for night cooling. By extending this period to 23:00 to 3:00, additional fan energy is incurred while the effect of reducing total energy consumption is uncertain (might be lower than the reduction from 23:00 to 1:00 or even negative).

4.4.2 Simulation Results

Simulation results for night cooling are shown in charts below. Total electricity consumption consists of two parts: roof fan energy and chiller energy (sensible cooling only). Energy of the roof fan is measured in units of electricity consumption. The sensible cooling load is measured in thermal energy units. By assuming a chiller COP (coefficient of performance) value of 3, thermal energy is converted to electric energy and added up to the roof fan consumption.

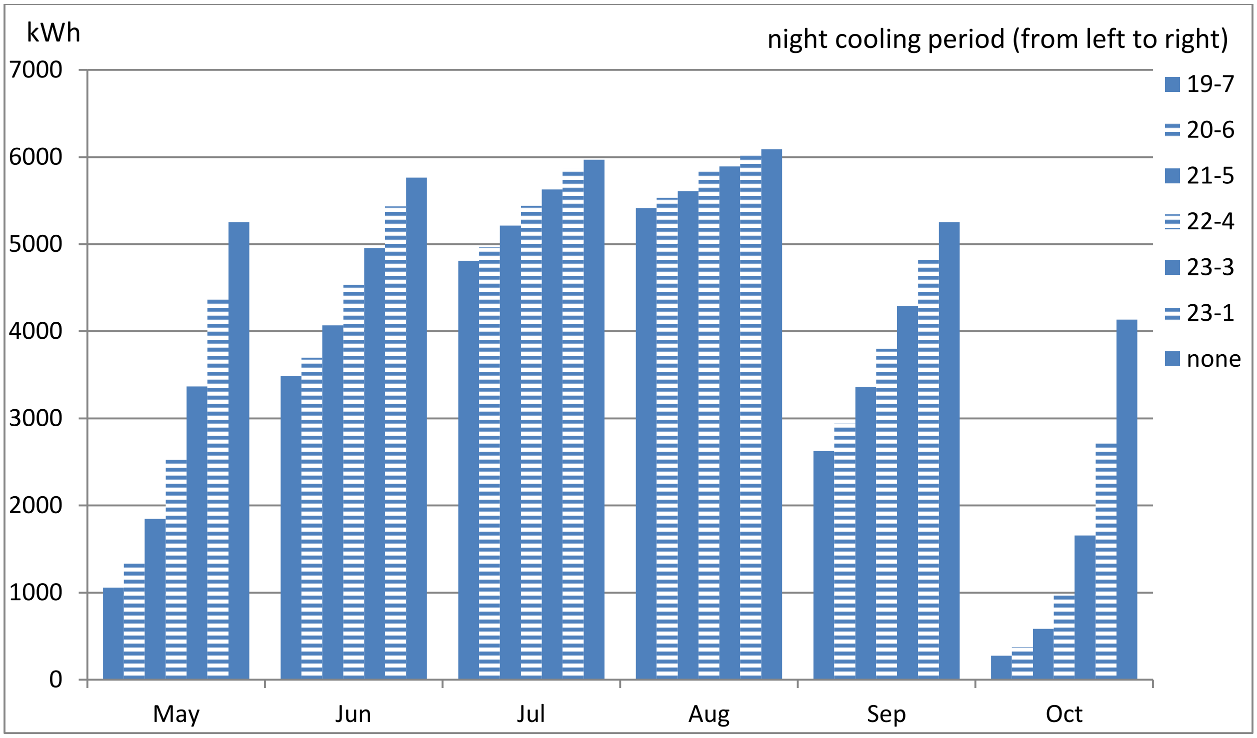


Figure 4-10. Total electricity for standalone retail in Atlanta

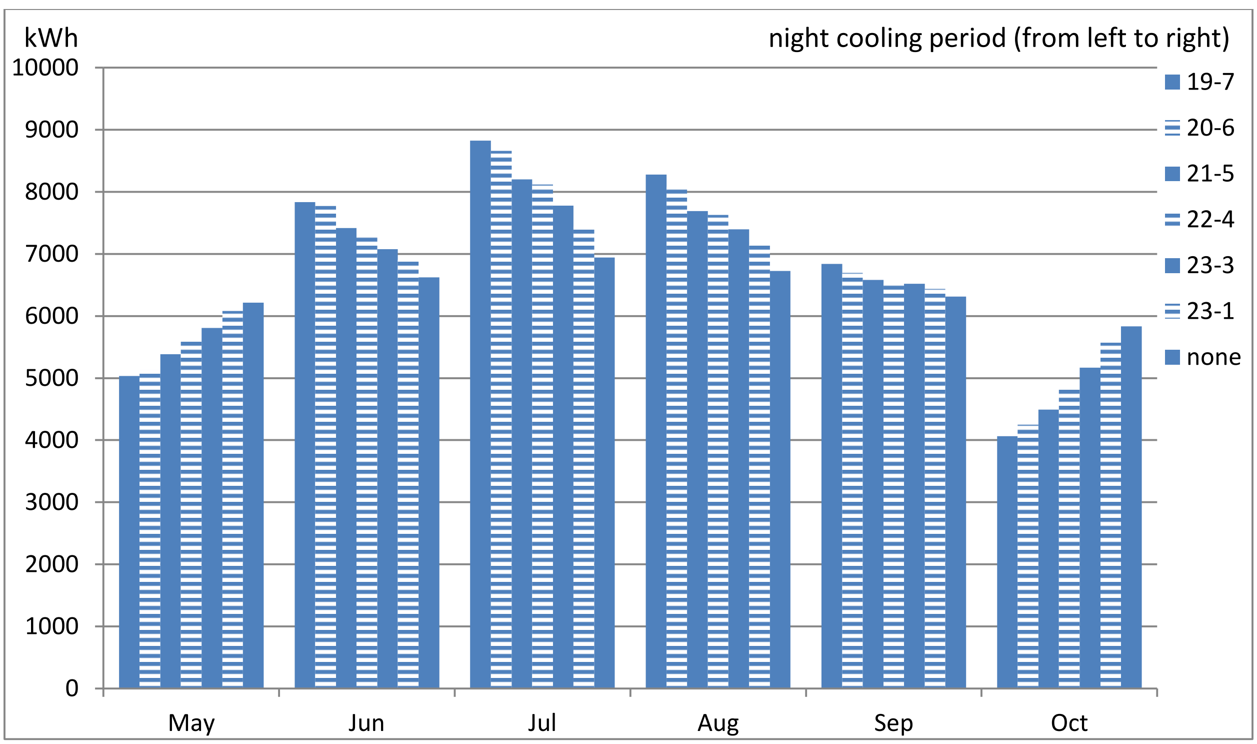


Figure 4-11. Total electricity for standalone retail in Phoenix

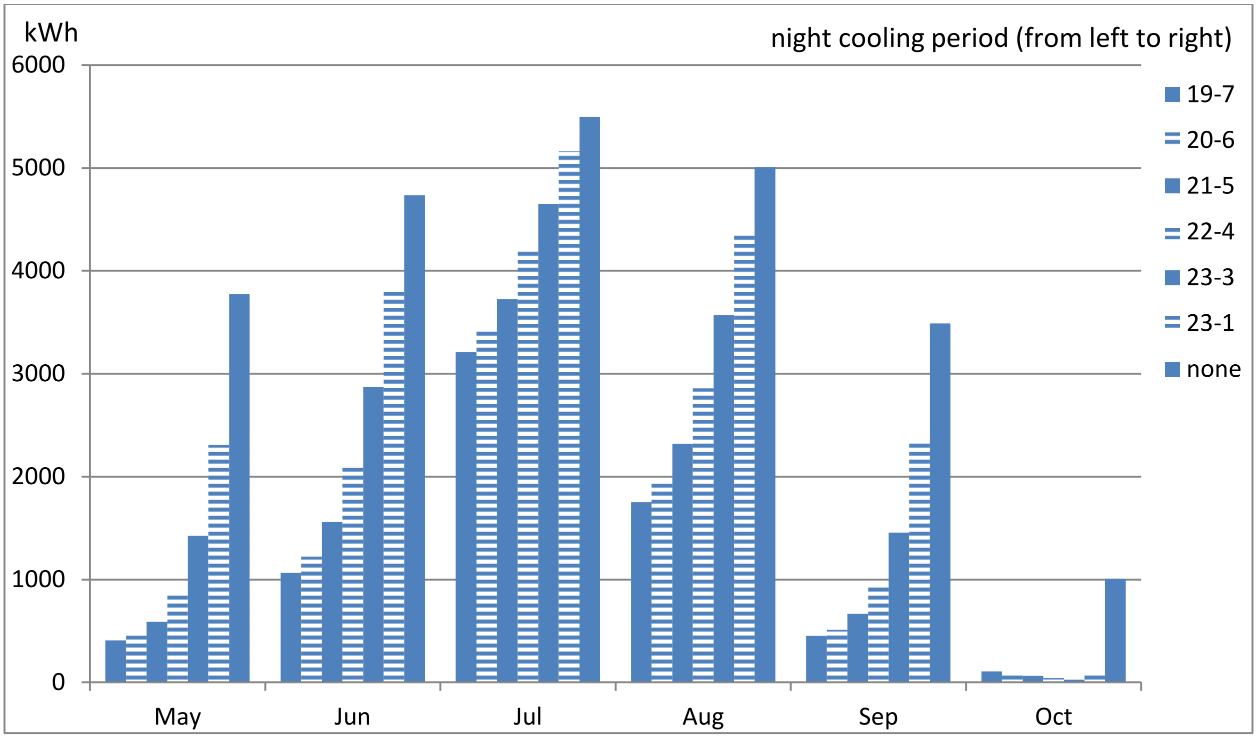


Figure 4-12. Total electricity for standalone retail in Minneapolis

4.4.3 Discussion

Results show that night cooling has energy saving potential in all climate types. However, the effect is different for different months. Energy consumption of the roof fan is relatively small compared with the chiller energy consumption (annual consumption is only 5% to 10% of chiller consumption). This suggests use this strategy as long as possible in a cool climate. However, if the weather is hot (like in June to September in Phoenix), the optimal strategy is turning off night cooling.

The energy saving potential for the standalone retail building (optimized for each month, corresponding to the minimum electricity consumption of chiller and fan) is shown in the table below. Energy considered is total electricity consumption for sensible cooling and fan energy.

City	Electricity for sensible cooling (kWh/a)	Optimized results (kWh/a)	Percentage Saving
Atlanta	44606	18116	59%

Phoenix	61083	39127	36%
Minneapolis	24338	6996	71%

Table 4-10. Sensible load saving potential of night cooling for different cities

It shows that night cooling might be most suitable for dealing with high internal load when outdoor climate is cool. The increased fan energy is less than the sensible load saving (for this building type which has high internal load and relatively low flow resistance), which suggests the optimal strategy to be maximum level. The relative magnitude of the roof fan could be estimated by the following calculation.

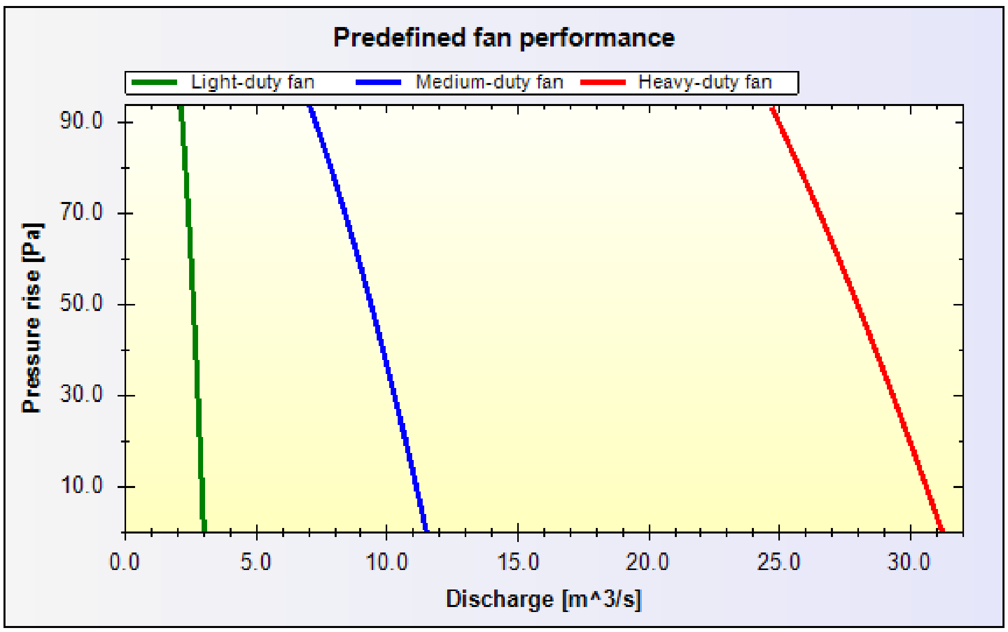


Figure 4-13. Fan curve in CoolVent

For this building model, a medium-duty fan is used. Monthly fan energy for a 12h (7pm to 7am next day) night cooling period is calculated below:

$$12h \times \frac{10m^3}{s} \times 50Pa \times 30day \Big/ \eta = 0.7 = 260kWh$$

Electricity consumption of chiller is about 6000kWh in August for Atlanta. Specific optimization for other building type and weather type could be derived using similar method.

4.5 Hybrid Mode

4.5.1 Principles

The hybrid mode is a combination of natural ventilation and air conditioning mode discussed before. Aim of this mode is to take full advantage of natural ventilation and reduce cooling energy as much as possible. Several important assumptions about this mode are listed below:

(1) Different comfort standards applied;

Comfort limit for air conditioning mode is set to be 26°C for temperature and 12g/kgda for humidity ratio. Comfort limit for natural ventilation mode is set to be 28°C for temperature and 14 g/kgda for humidity ratio.

(2) Energy of fan and chiller will be considered;

To convert thermal energy to electricity, a chiller COP of 3 is assumed. Total electricity consumption (fan and chiller) will be compared with chiller consumption in the pure air conditioning mode (chiller COP is assumed to be 3 and there is no fan energy consumption).

(3) Night cooling is optimized;

During night time, air conditioning will be turned off. The building will be open and ventilated with fan to get the thermal mass cooled. Duration of night cooling is based on conclusions from 4.4. For cool and moderate locations (Atlanta, Chicago, Duluth, Fairbanks, Los Angeles and Minneapolis), night cooling is at maximum level for all months. For hot zones (Houston, Miami and Phoenix), it is not used in July and August while at maximum level in other months. During other months, total energy consumption under maximum night cooling is less than or close to that under no night cooling. To simplify simulation, night cooling is assumed to be at maximum level.

(4) Control logic is necessary;

Control logic could be described in the following figure.

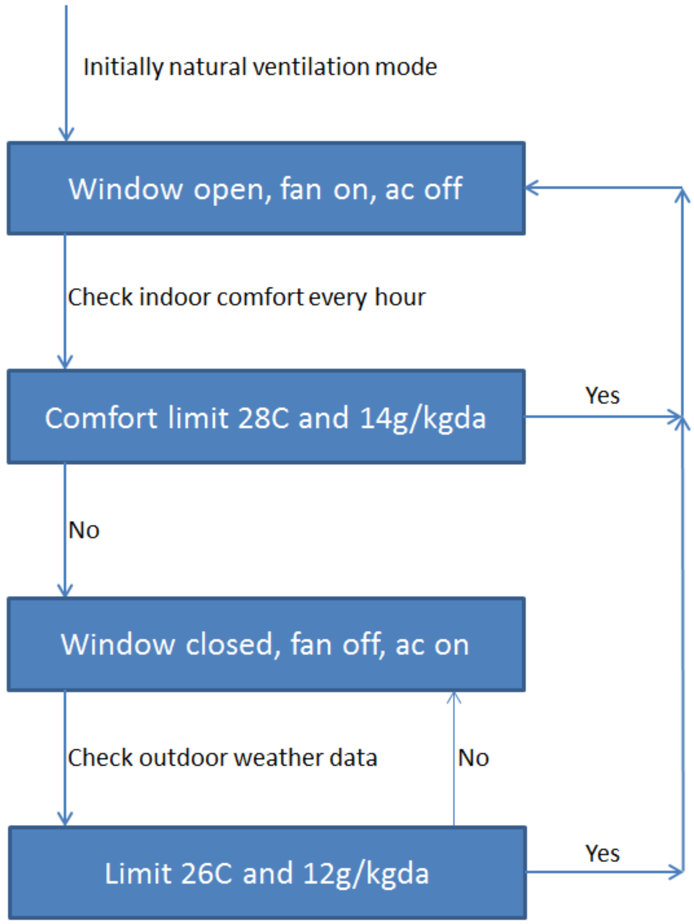


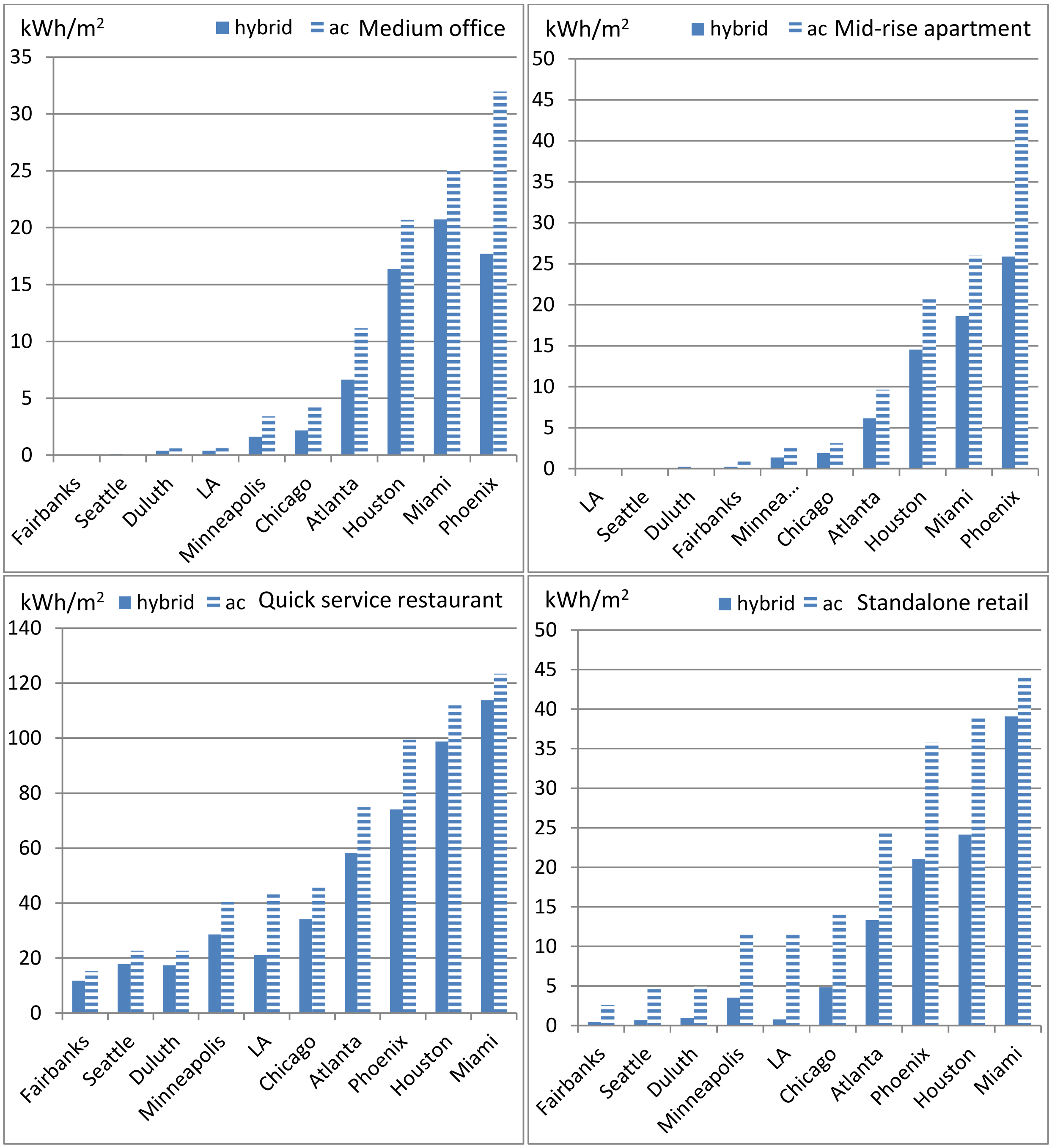
Figure 4-14. Control logic for hybrid mode

In natural ventilation mode (for both daytime and night ventilation, all windows are open, fan is in operation, air conditioning is off), program will check indoor environment every hour. If comfort standard for natural ventilation cannot be met in any zone, operation mode for the entire building will be switched to air conditioning. The building will be maintained at 26°C and 12 g/kgda. Program will check outdoor weather every hour. If outdoor environment is considered to be appropriate for natural ventilation, operation mode will be switched to natural ventilation. The fan will be set constantly in operation when the building is under natural ventilation mode.

4.5.2 Simulation Results

During some month, electricity consumption in hybrid is higher than that in ac mode. This is because the additional fan energy exceeds the saved chiller energy. Under this case, optimal operation will be pure air conditioning instead of hybrid mode.

These cases occur mostly when air conditioning load is trivial. Electricity consumption (roof fan and chiller total) for different types of building is shown in figures below.



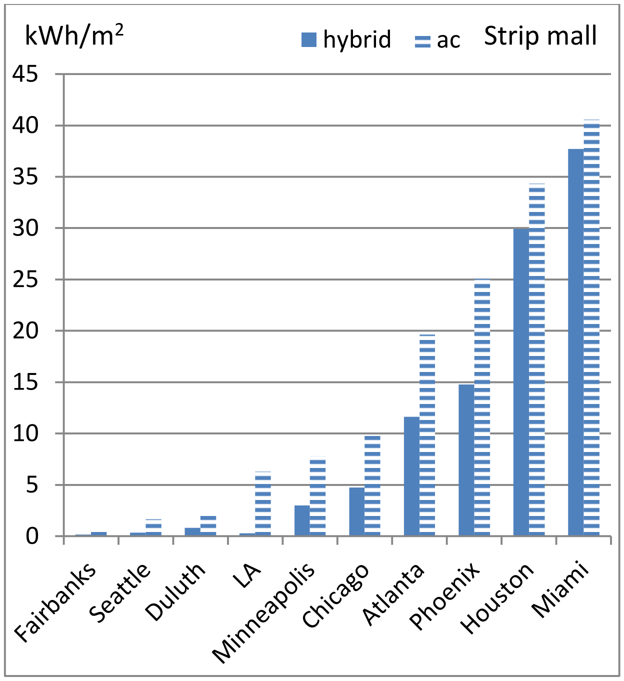


Figure 4-15. Comparison between hybrid and air conditioning mode for five building types

4.5.3 Discussion

Results show that hybrid mode is able to save cooling electricity while maintaining a comfortable environment. This mode strongly depends on the assumption that occupants under natural ventilation case tend to tolerate higher temperature and humidity ratio.

Weather type has a large impact on saving potential. Although air conditioning loads are all high, saving potential in Phoenix is larger than that in Miami and Houston. The reason for this difference is that Phoenix has a hot and dry climate while the other two cities are more humid.

From this result total energy saving potential for each climate zone could be estimated. Using a weighted average method, percent energy saving could be calculated by the following equation:

$$w = 1 - \frac{\sum_i A_i E_{hybrid,i}}{\sum_i A_i E_{ac,i}}$$

w is percentage electricity saving

A_i is the weight factor of building i′s floor area in certain climate zone

$E_{hybrid,i}$ is electricity consumption of building i under hybrid mode

$E_{ac,i}$ is electricity consumption of building i under air conditioning mode

See Appendix for detail, percentage saving of hybrid mode from air conditioning mode is shown in the figure below.

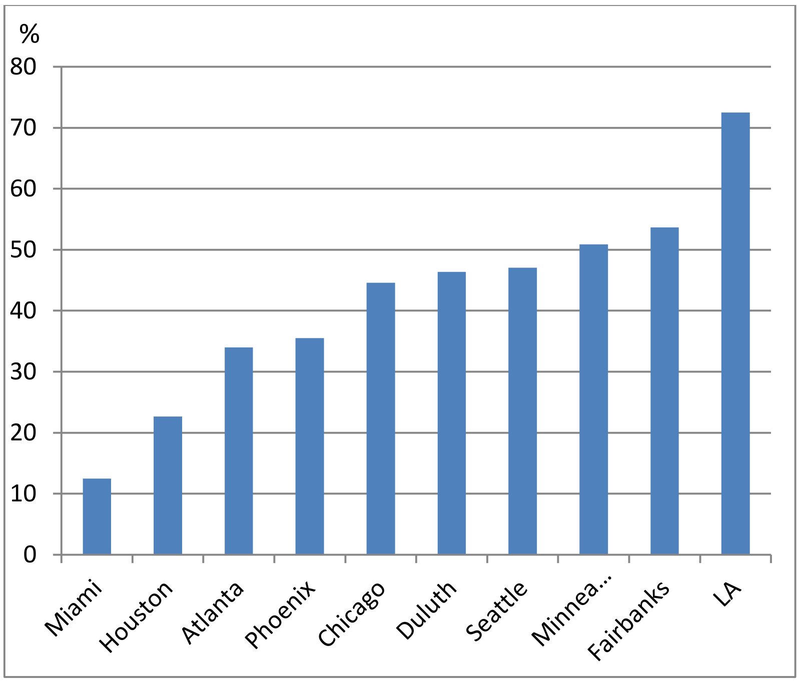


Figure 4-16. Percentage energy saving of hybrid mode from pure air conditioning

This figure shows that hybrid mode has a significant saving potential from 12% in Miami to 72% in Los Angeles.

4.6 Discussion

There are several limitations of this simulation that might need further improvement.

Air conditioning energy is represented solely by chiller energy, not including other part of the cooling system (such as supply fan energy). Energy consumption of all parts of the cooling system is represented by the overall COP of 3.

Another limitation is that due to the simplified calculation method, sometimes indoor air temperature is slightly higher than the upper comfort limit. For example, if the upper limit is 26°C, indoor air temperature might be 26.4°C but will be maintained below 27°C. This is because cooling amount is based on previous temperature and there is a lag.

Since CoolVent's simulation cannot proceed without calculating airflow rate. The "pure air conditioning" is not strictly defined air conditioning. The window opening fraction is kept at minimum value (opening fraction is 0.05) and the airflow rate is minimized as much as possible.

5 DEMONSTRATION MONITORING

5.1 General Description

Figure 5-1. South façade of the Artists for Humanity EpiCenter

The Artists for Humanity Epicenter is located in the south of Boston with three storeies and a floor area of 23,500 square feet. The three stories of the building serve respectively as gallery, office and studio. This LEED platinum building has a number of attractive features: photovoltaic system generating electricity, glass wall minimizing lighting needs and well insulated wall decreasing cooling/heating load. Among them the most outstanding one might be the fact that this building is the first naturally cooled commercial building without air conditioning in Boston. This building has proven to be cost and energy efficient.

In summer, the building is operated under pure natural ventilation mode. The building is open at night and the entire space is ventilated with fan located on the roof. This will take advantage of the cool outdoor air to lower the temperature of indoor air and building thermal mass. At daytime, windows of the building are closed to prevent hot outdoor air entering indoor space. The cooled thermal mass is capable of maintaining indoor temperature comfortable to some extent. There are also some ceiling fans to create internal air flow to improve occupants' comfort feeling when indoor temperature exceeds comfort limit inevitably.

The building has three floors with different schedules. The first floor is occupied the least. Since half the first floor is located underground, its heat gain from ambient is also less compared with the other two floors. The second floor is the main working area. There is a high level heat gain due to occupants and utilities. There are also two separate studios on the second floor which are used for manufacturing and painting. Because of the huge amount of heat gain, there are single air conditioning unit in each room. The third floor is a large open space and occupant schedule is uncertain.

Our measurement of the building includes temperature, humidity, air flow, thermal comfort and electricity consumption. Our goal is to evaluate the actual performance of natural ventilation in this well designed building.

5.2 Sensors

5.2.1 Type

The measurement is concentrated on the temperature and humidity distribution of the entire building during a summer period. Two types of sensor we chose are listed below.

(1) Onset U10 temperature data logger

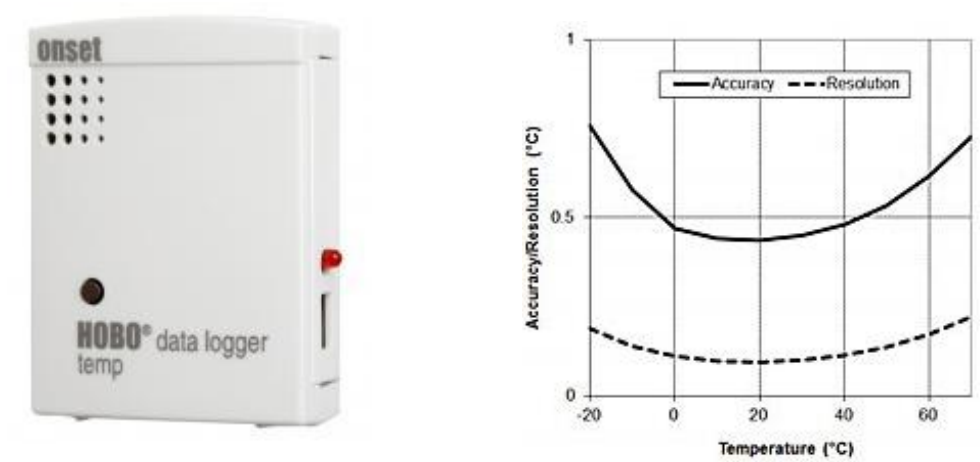


Figure 5-2. Onset U10 temperature data logger with accuracy and resolution

This sensor only measures temperature. There are 9 in total and are numbered No.16-24. Key parameters of this sensor are:

Specifications	Value
Measurement range	-20 to 70°C
Accuracy	± 0.53°C from 0 to 50°C
Resolution	0.14°C at 25°C
Sampling frequency	5 min

Table 5-1. Specifications of Onset U10 temperature data logger

(2) Onset U10 temperature relative humidity data logger

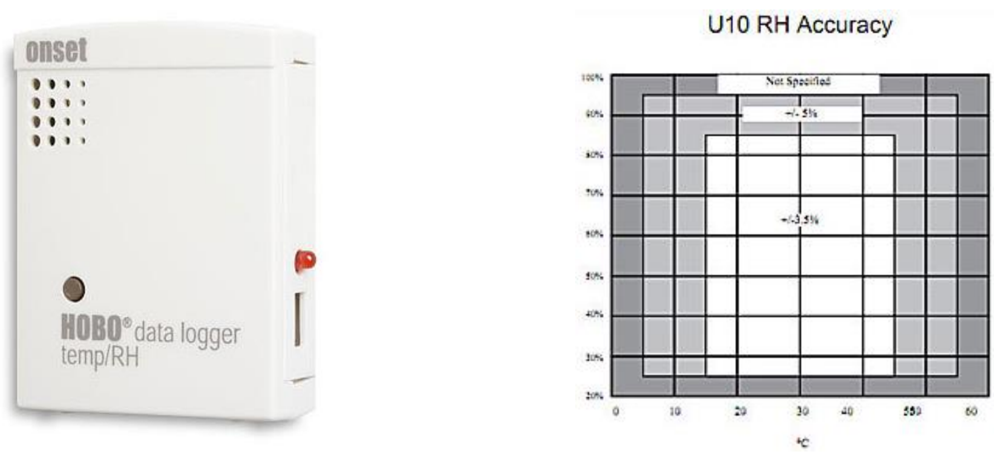


Figure 5-3. Onset U10 temperature RH data logger with accuracy and resolution of RH

This sensor can measure temperature and relative humidity. There are 16 in total and are numbered No.1-15 and No.25. It has the same accuracy and resolution for temperature of the temperature only type. As for humidity measurement, key parameters are summarized in the chart below.

Specifications	Value (temp. / RH)
Measurement range	-20°C to 70°C / 25% to 95%
Accuracy	± 0.53°C from 0 to 50°C / See figure
Resolution	0.14°C at 25°C / 0.07% @ 25°C and 30% RH
Sampling frequency	5 min

Table 5-2. Specifications of Onset U10 temperature relative humidity data logger

5.2.2 Location

Monitoring period is from July 10th 2012 to August 21st 2012. Specific location and type of sensors are summarized in the chart below.

No.	Location	Note
1	2F, front aisle between wood shop and screen printing	Air temp.
2	3F, fan vent	Ventilation temp. and RH
3	2F, in office on desk	Air temp. and RH
4	1F, 2nd pillar	Air temp. and RH
5	2F, fan vent	Ventilation temp. and RH, missing on Jul 25th , no data
6	1F, fan vent	Ventilation temp. and RH
7	2F, 3D design room, under Parla's shelf	Air temp. and RH, missing on Aug 27th, partial data
8	Mezzanine on 1F, on a column close to TV	Air temp. and RH
9	3F, Slab temperature next to painting racks	Slab temp., missing on Aug 27th , partial data
10	2F, 3D design room, slab temperature	Slab temp., missing on Aug 27th, partial data
11	1F, under stairs, slab temperature	Slab temp., missing on Jul 25th, no data
12	3F, on a column close to partition	Air temp. and RH
13	3F, metallic mezzanine on a column	Air temp. and RH
14	-	Out of battery
15	2F, in the screen printing room in bin #6 rack	Air temp. and RH in separate studio
16	Mezzanine on 3F, on pillar far from exit, moved to 2F vent on Aug. 7	Air temp. / ventilation temp. and RH
17	2F, back aisle, office area	Air temp.
18	3F, like 12, close to exit	Air temp.
19	1F, on column next to emergency lights	Air temp.
20	2F, in wood shop, next to entrance	Air temp. in separate studio

21	2F, on the fan control/switch in the office area	Air temp.
22	1F, on column next to fire alarm	Air temp.
23	Mezzanine on 2F, in the stairwell	Missing
24	Mezzanine on 3F, on pillar far to exit	Air temp.
25	Outside weather data	Solar radiation avoided

Table 5-3. List of sensor locations

Below is the blueprint for the building. The first mezzanine is located on the second floor. The mechanical loft is placed on the first floor.

Figure 5-4. Floor plan of the Artists for Humanity Epicenter

First floor details

Figure 5-5. View of 1F

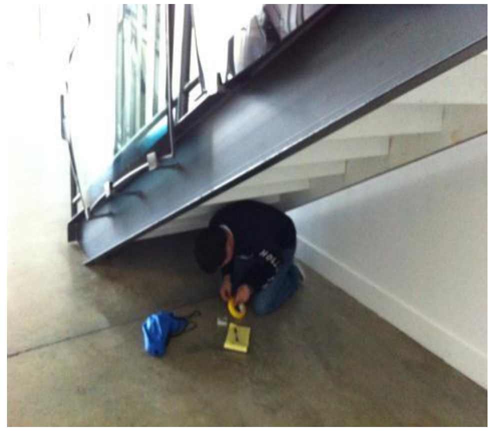


Figure 5-6. Sensor No.11 for slab temp.

Figure 5-7. View from window on 1F

Figure 5-8. Sensor No.4 on 1F

Second floor details

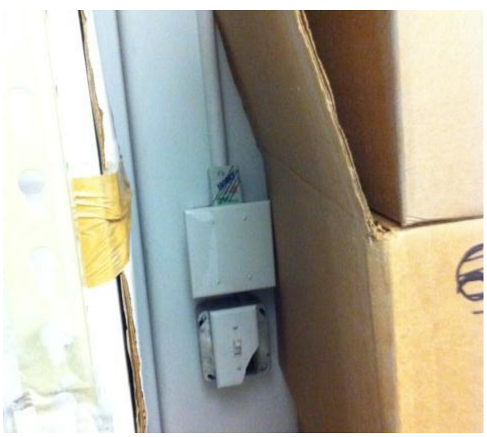


Figure 5-9. Sensor No.1 on 2F

Figure 5-10. Sensor No.21 on 2F (behind the desk)

Third floor details

Figure 5-11. Sensor No.9 on 3F

Figure 5-12. Sensor No.24 on 3F

5.3 Comfort Analysis

5.3.1 First Floor Comfort Analysis

These four sensors are located uniformly across the first floor. If we take 28°C and 14 g/kgda as the upper comfort limit for naturally ventilated environment, analysis shows that only during 11% of the measuring period inside temperature exceeds this value. It should be noted that this is a time and location weighted value.

Occupancy load on the first floor is very low. The space is rarely occupied and there is almost no utility load. Since half of this floor is located underground, heat gain through building envelop is also small. The most uncomfortable location is the location of No.8 sensor, which is located very close to a window and shows comfort condition close to outdoor environment. For other locations, comfort condition is good and temperature is more stable than that of No.8.

Table 5-4 summarizes uncomfortable percentage during occupied hours.

No.	Location	Uncomfortable temp.	Uncomfortable humid.
4	1F, 2nd pillar	8.2%	19.0%
8	Mezzanine on 1F, on a column close to TV	19.8%	35.7%
19	1F, on column next to emergency lights	7.4%	-

22	1F, on column next to fire alarm	10.1%	-
25	Outside condition	25.5%	32.9%

Table 5-4. Comfort analysis of 1F

Figures below show frequency distribution and time series of temperature from each sensor on the first floor. (Note: due to the location of outdoor temperature sensor, there is a daily peak caused by solar radiation)

No.4 and No.8

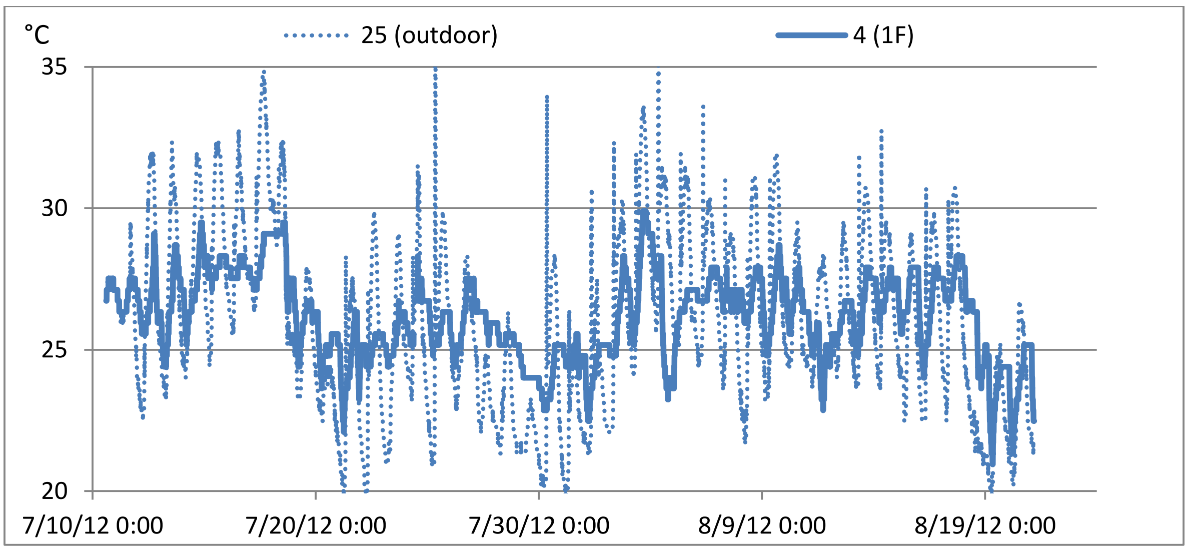


Figure 5-13. Time series of temperature from No.4

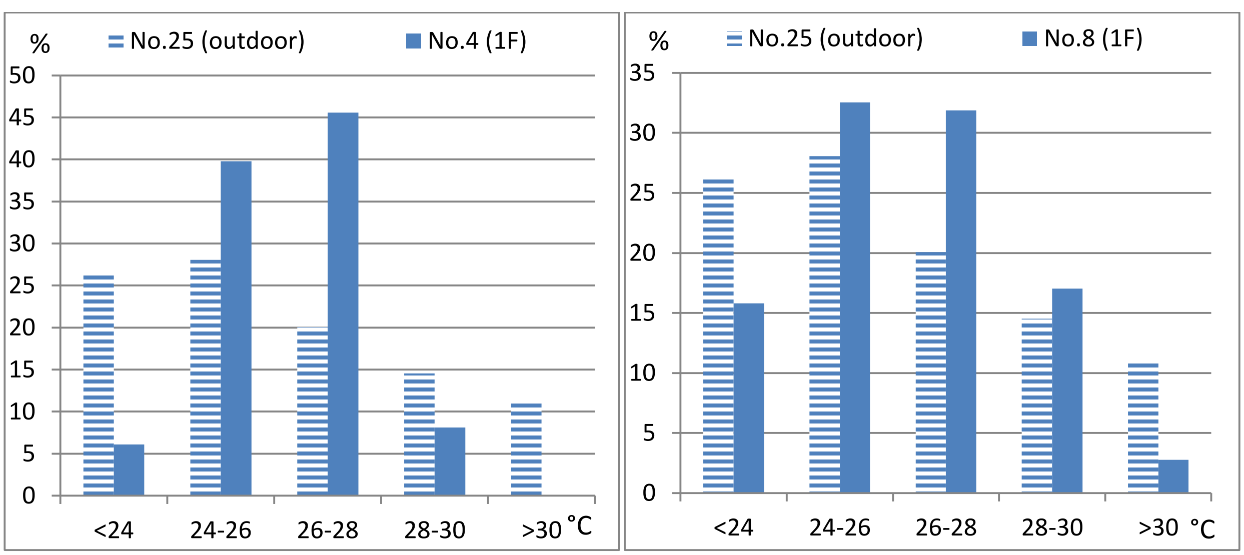


Figure 5-14. Frequency distribution of temperature from No.4 and No.8

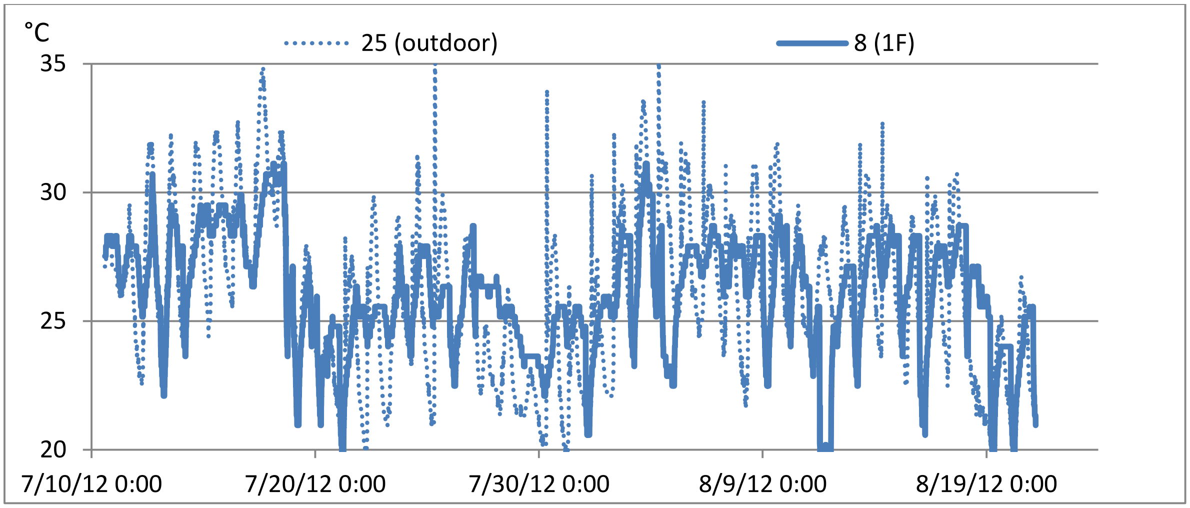


Figure 5-15. Time series of temperature from No.8

No.19 and No.22

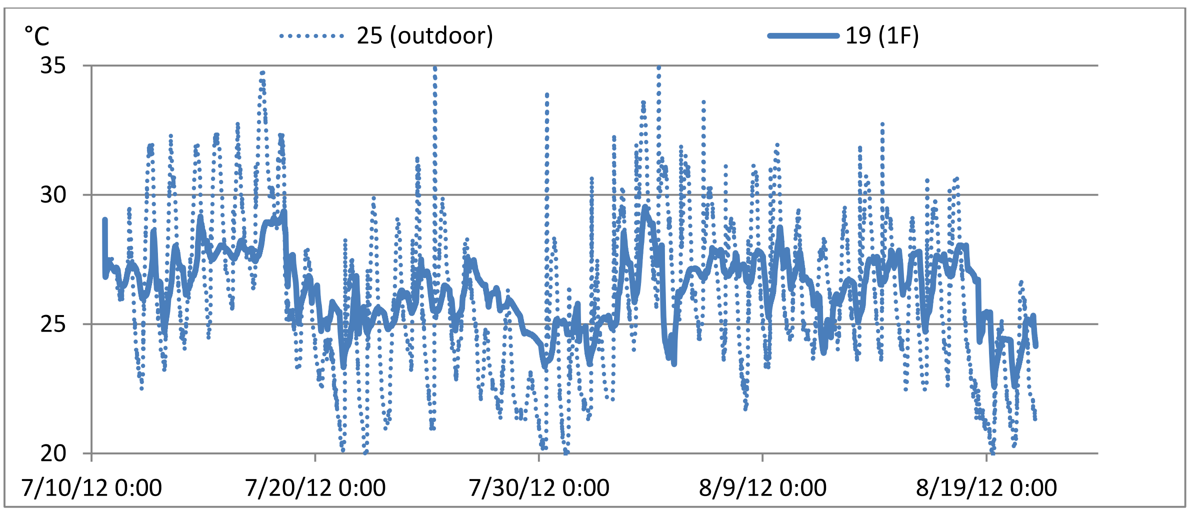


Figure 5-16. Time series of temperature from No.19

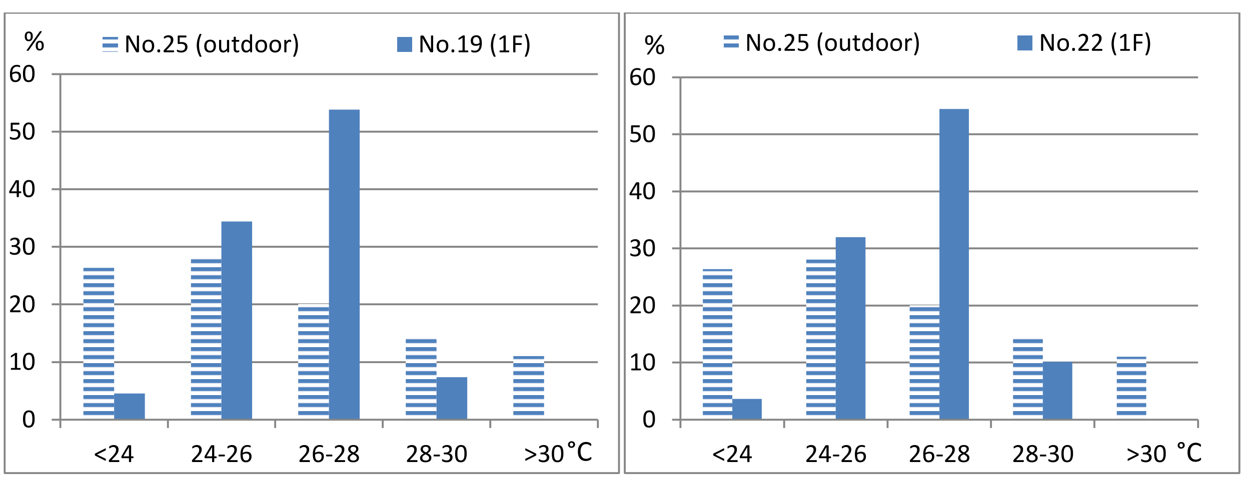


Figure 5-17. Frequency distribution of temperature from No.19 and No.22

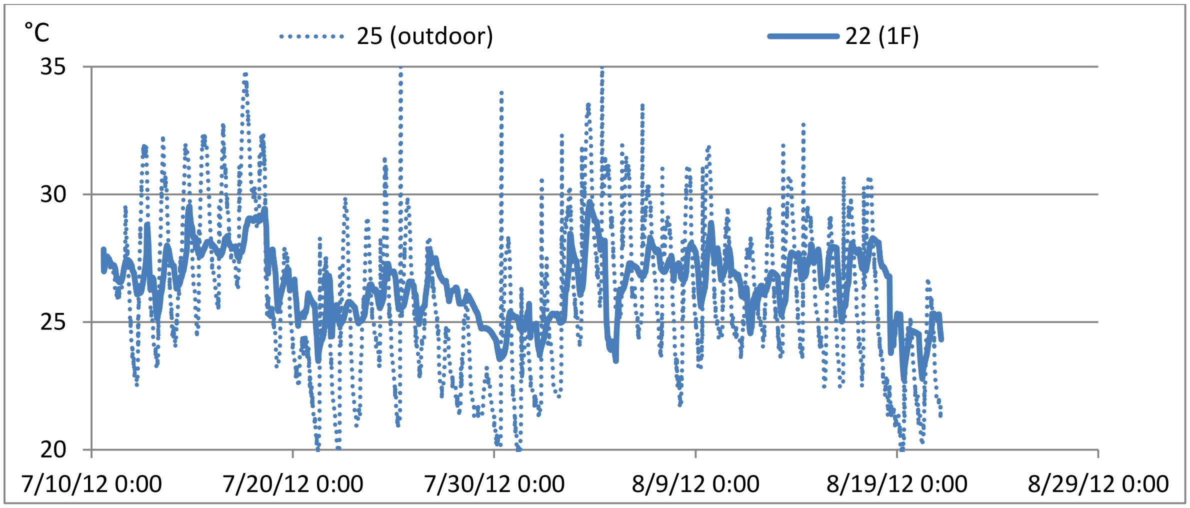


Figure 5-18. Time series of temperature from No.22

Discussion

It can be seen that comfort condition on the first floor is good and even better than that of the outdoor weather. Mostly temperature of the zone is below 28°C. When outdoor temperature exceeds 28°C, indoor one could still be maintained below 28°C. This is because of the low heat gain and good insulation. As for humidity ratio, results from the sensor near table are close to outdoor value; results from the sensor further from window have a lower uncomfortable percentage.

5.3.2 Second Floor Comfort Analysis

Same as the first floor, comfort condition on the second floor is summarized below. The second floor is the main office area and has high occupancy load. Utility load in the separate studios (for example, No.15) is higher than that of the office. Hence, the comfort condition is not so good.

No.	Location	Uncomfortable temp.	Uncomfortable humid.
1	2F, front aisle between wood shop and screen printing	36.8%	23.6%
3	2F, in office on desk	37.5%	20.7%
7	2F, 3D design room	26.9%	40.6%
15	2F, in the screen printing room	65.1%	19.8%

17	2F, back aisle, office area	35.0%	-
20	2F, in wood shop, next to entrance	53.7%	-
21	2F, on the fan control/switch in the office area	41.6%	-
25	Outside condition	25.5%	32.9%

Table 5-5. Comfort analysis of 2F

No.1 and No.3

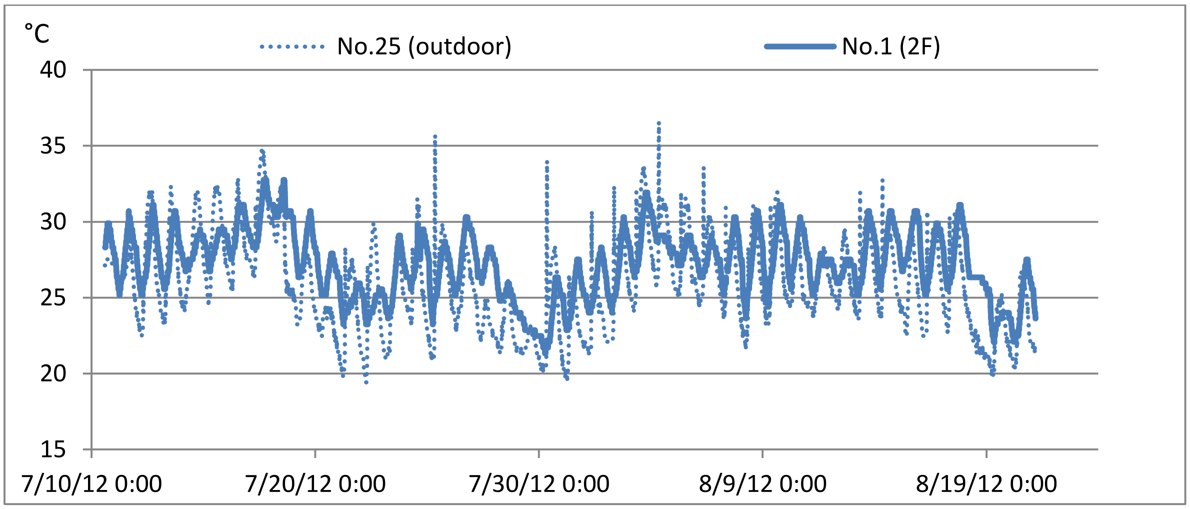


Figure 5-19. Time series of temperature from No.1

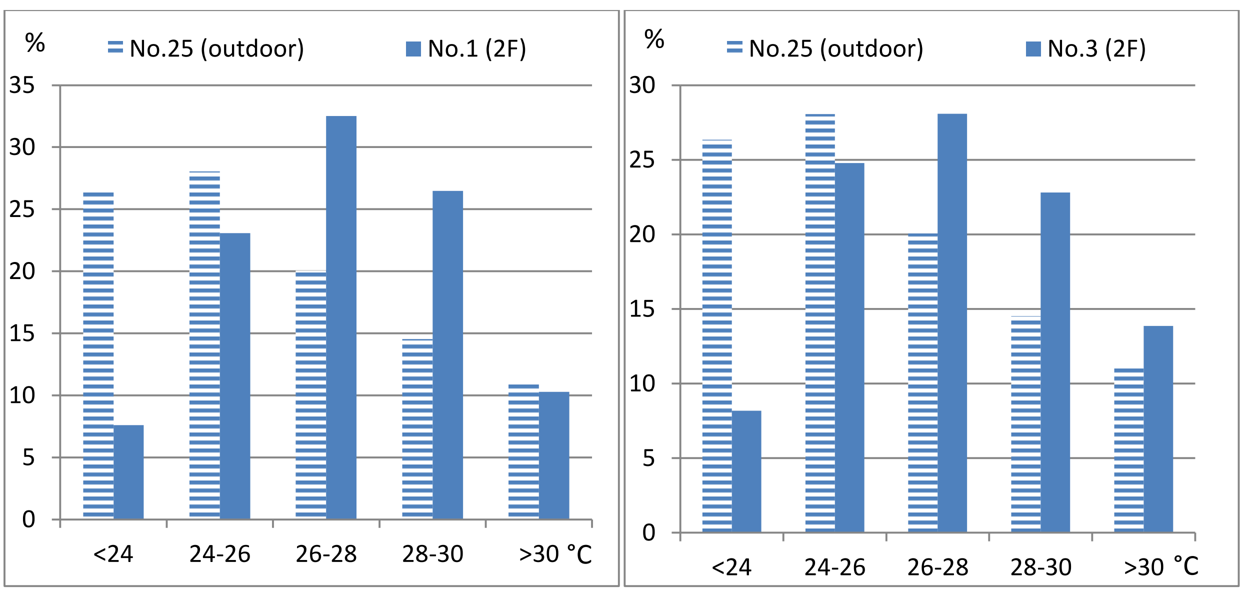


Figure 5-20. Frequency distribution of temperature from No.1 and No.3

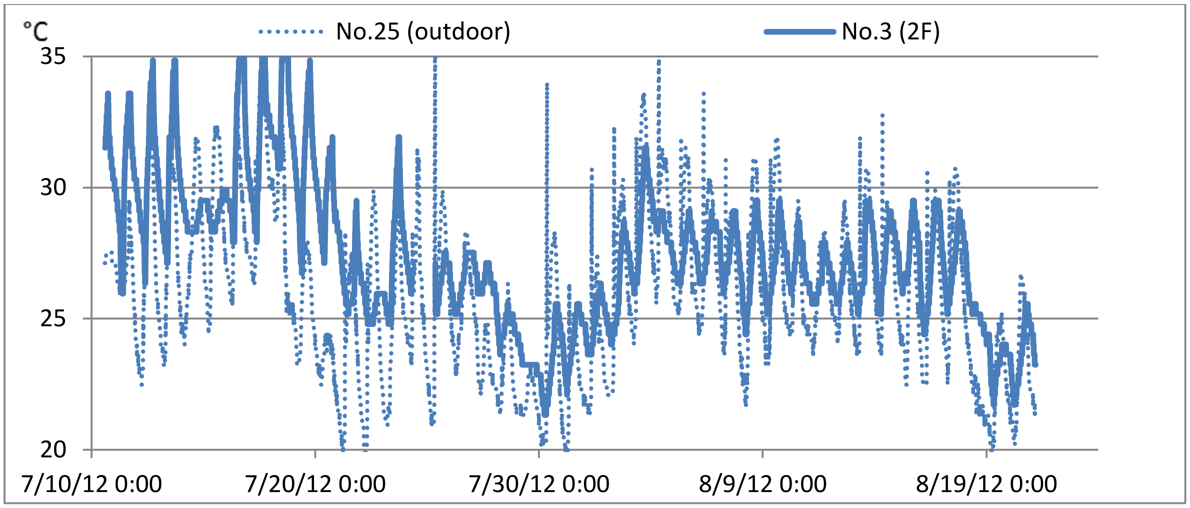


Figure 5-21. Time series of temperature from No.3

No.7 and No.15

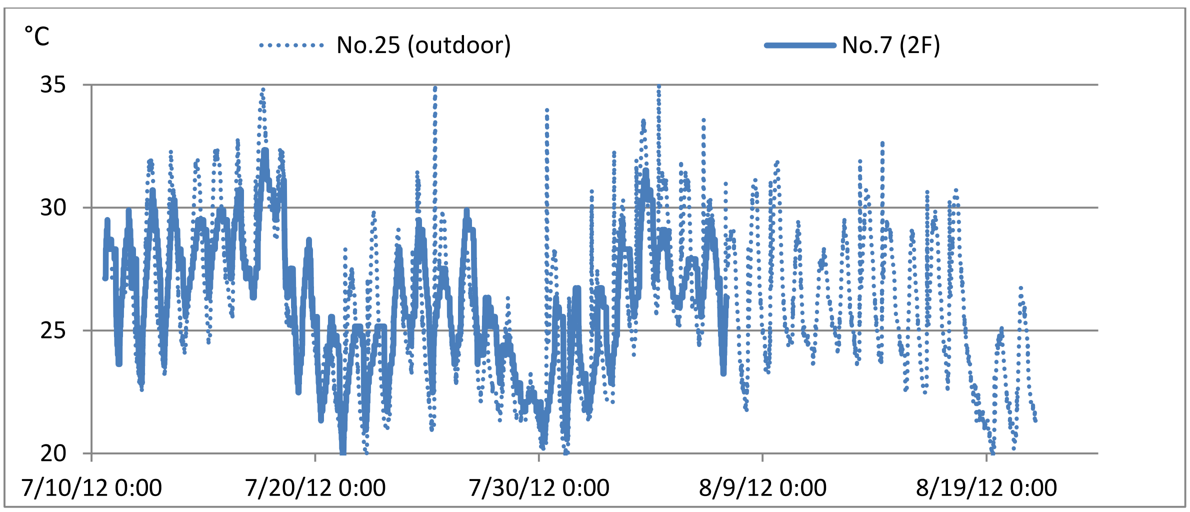


Figure 5-22. Time series of temperature from No.7

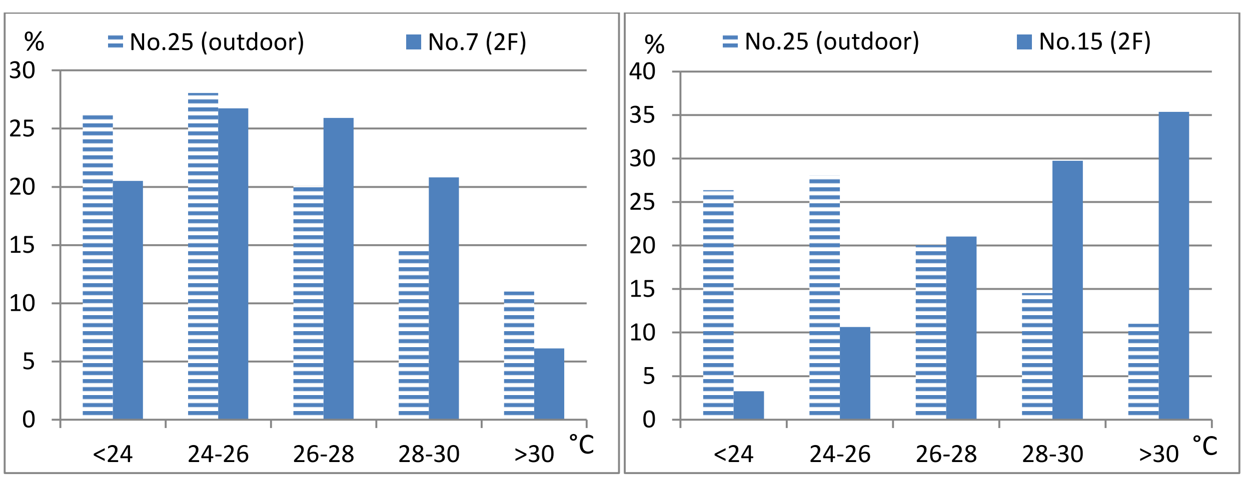


Figure 5-23. Frequency distribution of temperature from No.7 and No.15

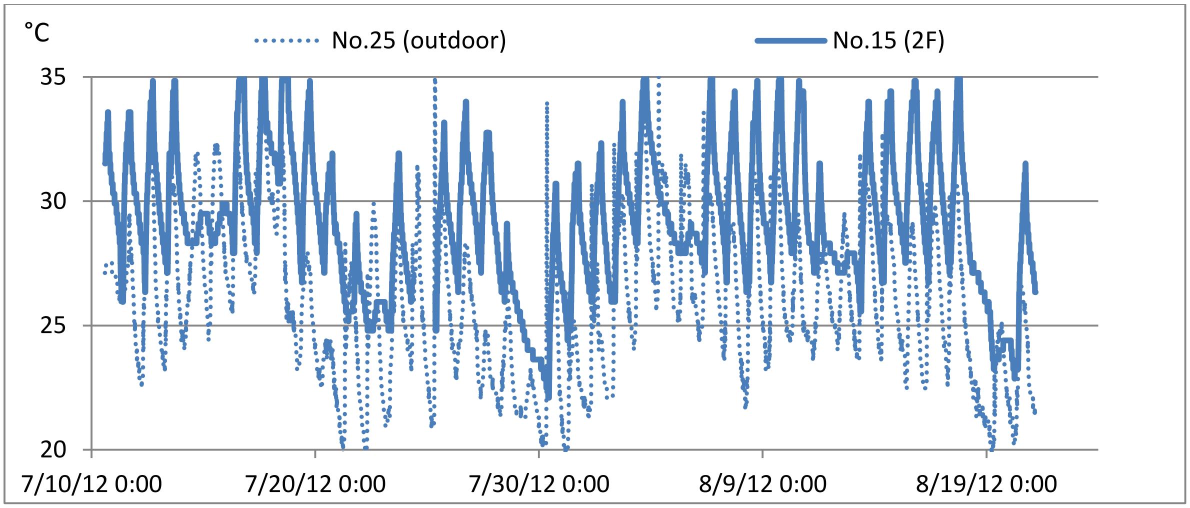


Figure 5-24. Time series of temperature from No.15

No.17 and No.20

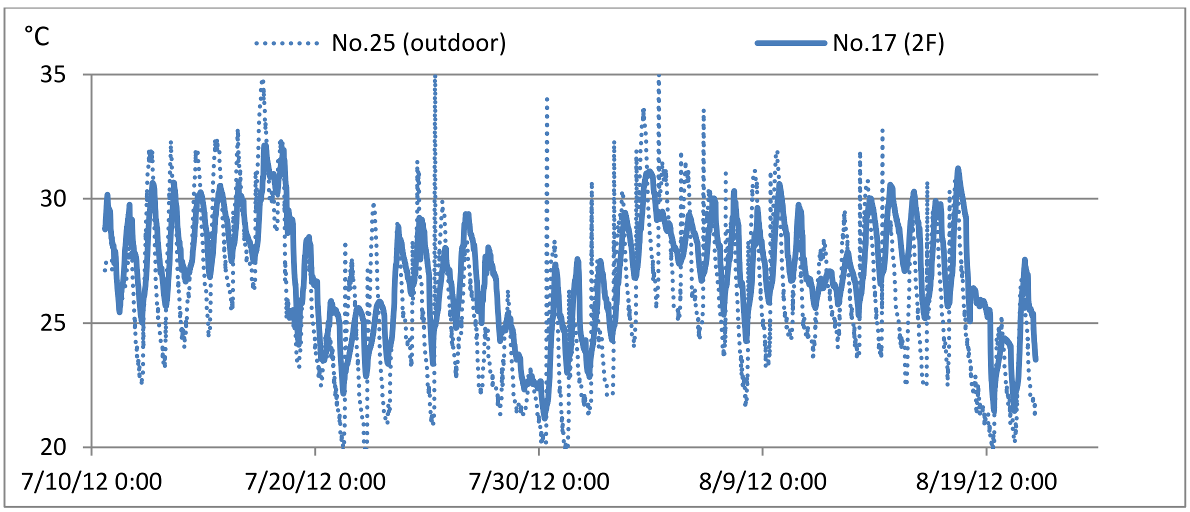


Figure 5-25. Time series of temperature from No.17

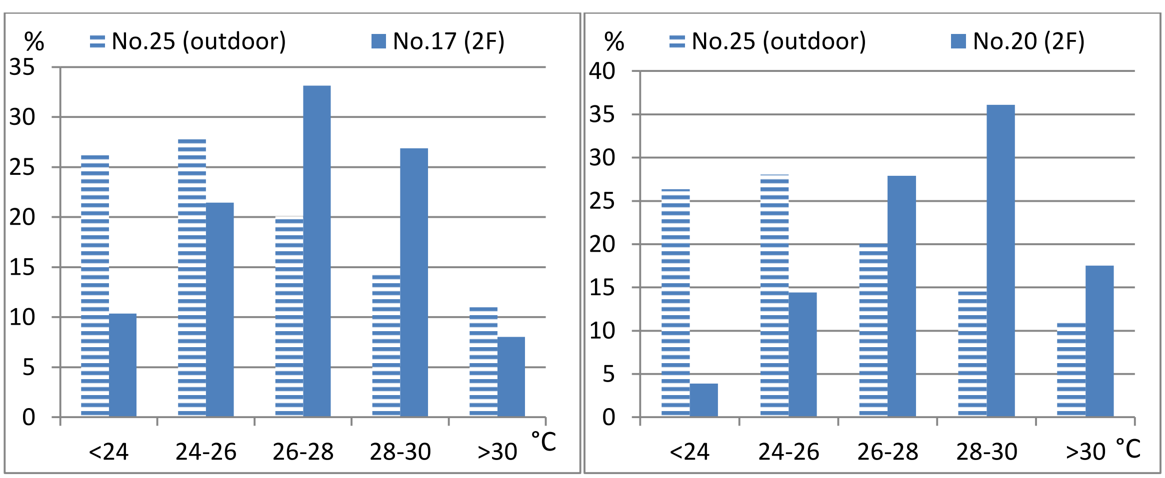


Figure 5-26. Frequency distribution of temperature from No.17 and No.20

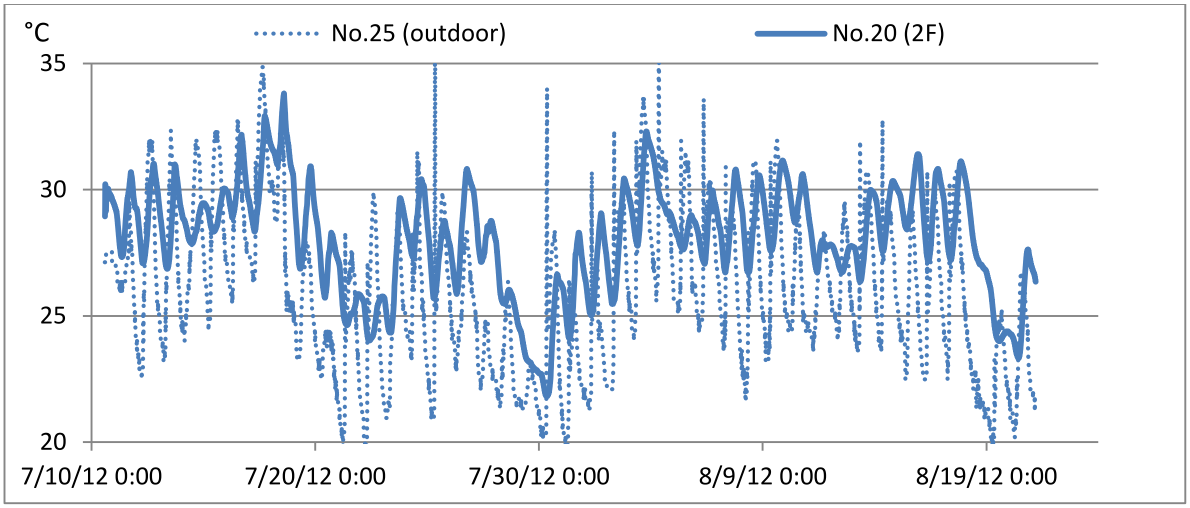


Figure 5-27. Time series of temperature from No.20

No.21

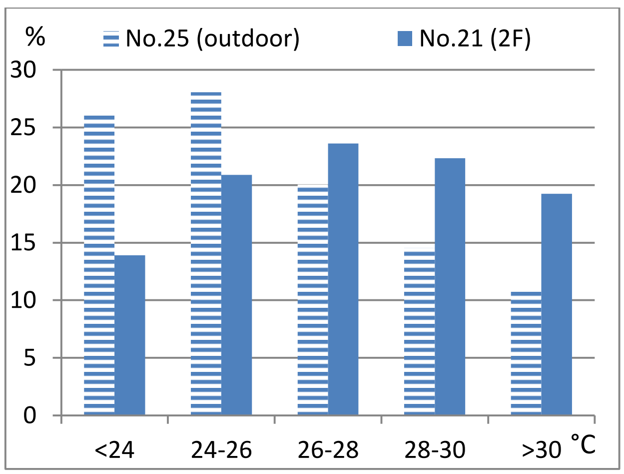


Figure 5-28. Frequency distribution of temperature from No.21

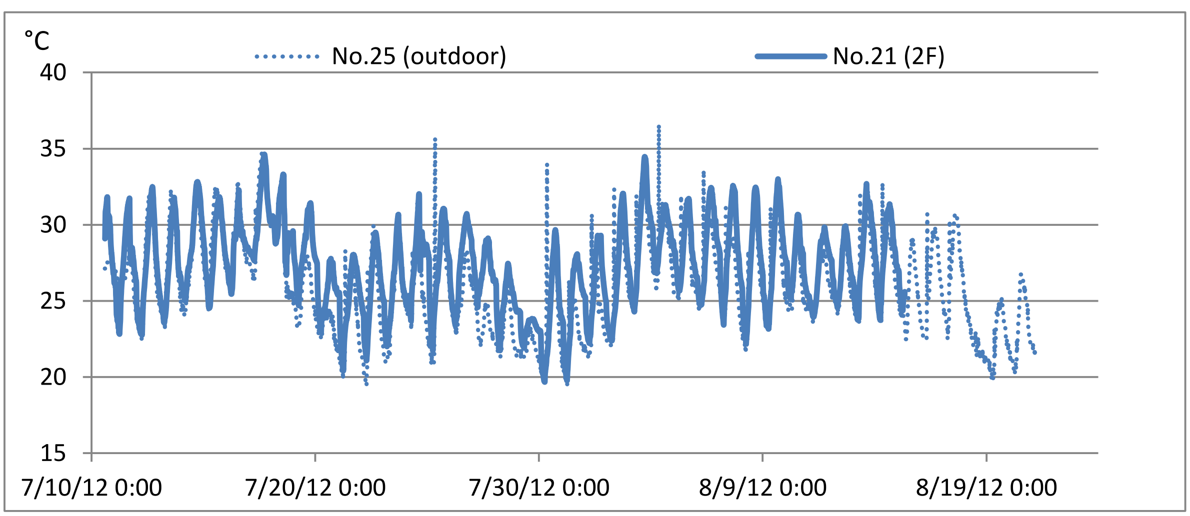


Figure 5-29. Frequency distribution of temperature from No.21

Discussion

Comfort condition is highly related to internal load. For No.7 a painting room with low internal load, the comfort condition is good. When it comes to office area, where the density of occupant and equipment gets higher, the uncomfortable percentage ranges between 35-45%. One reason for this is that windows in office area are closed at daytime while should be open when outdoor temperature is lower than indoor one at daytime. The office area takes advantage of the natural lighting; however, this incurs additional solar gain (shading is only used to prevent glare), which is another reason for the high temperature in daytime.

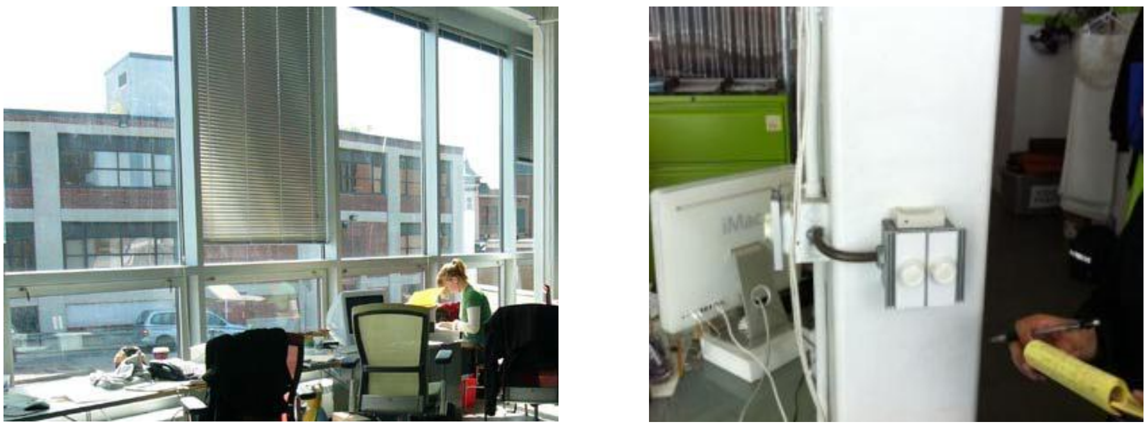


Figure 5-30. Office area (window facing south) and sensor

The most serious ones are two separate rooms with high power equipment for manufacturing. Ceiling fans are constantly operating to generate an air flow inside the closed space (not for natural ventilation with outdoor environment) but the air temperature is quite high. Uncomfortable percentages are 65% and 54% (based on temperature measurement from two sensors in these zones).

Figure 5-31. Sensors placed in two separate rooms

5.3.3 Third Floor Comfort Analysis

Comfort condition on the third floor is summarized below.

No.	Location	Uncomfortable temp.	Uncomfortable humid.
12	3F, on a column close to partition	33.2%	24.8%
13	3F, metallic mezzanine on a column	36.2%	23.6%
16	Mezzanine on 3F, on pillar far from exit (Before Aug 7th)	40.0%	-
24	Mezzanine on 3F, on pillar far to exit	46.1%	-
25	Outside condition	25.5%	32.9%

Table 5-6. Comfort analysis of 3F

No.12 and No.13

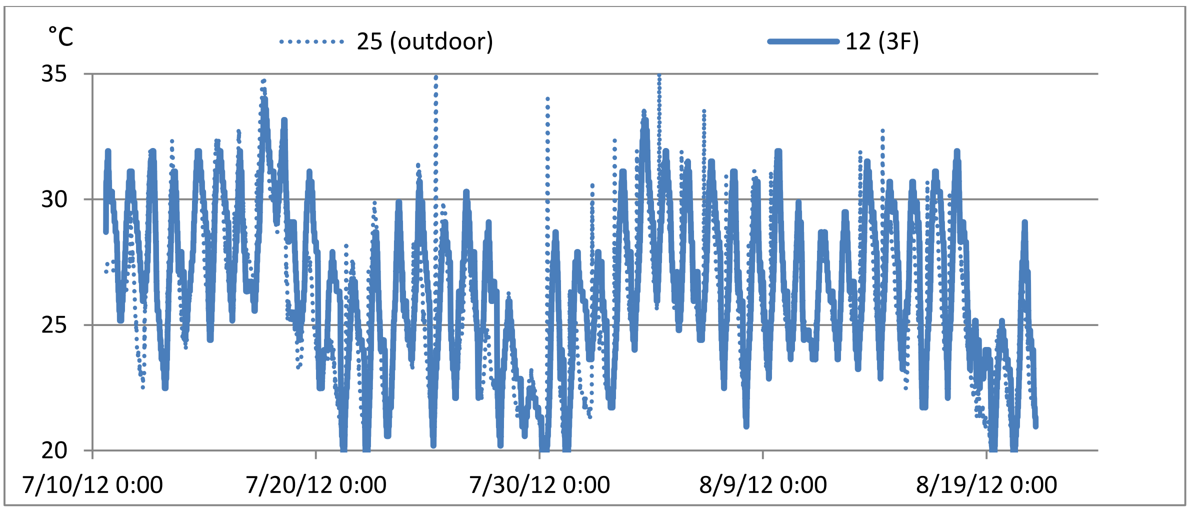


Figure 5-32. Frequency distribution of temperature from No.12

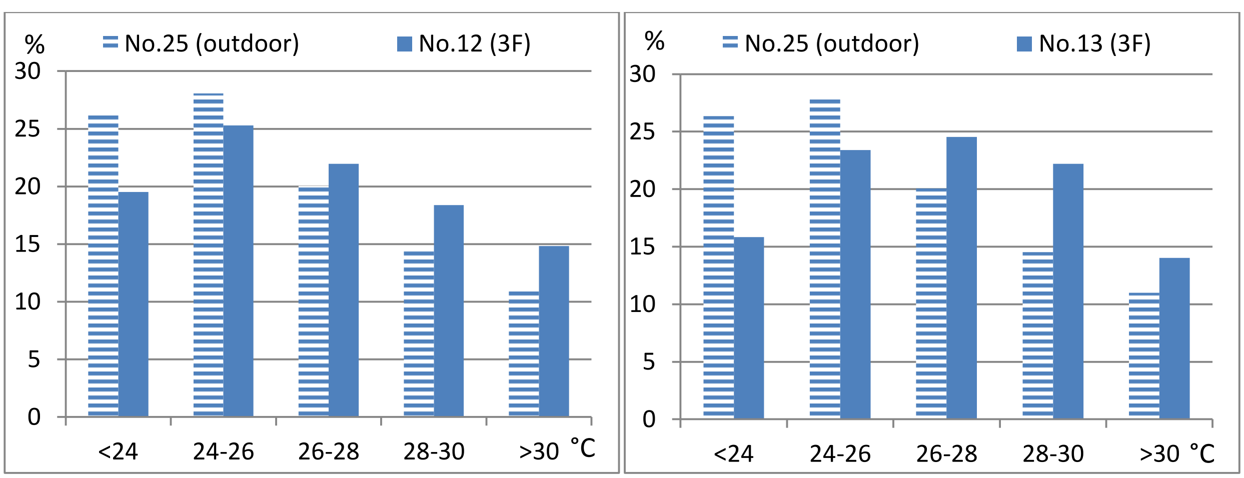


Figure 5-33. Frequency distribution of temperature from No.12 and No.13

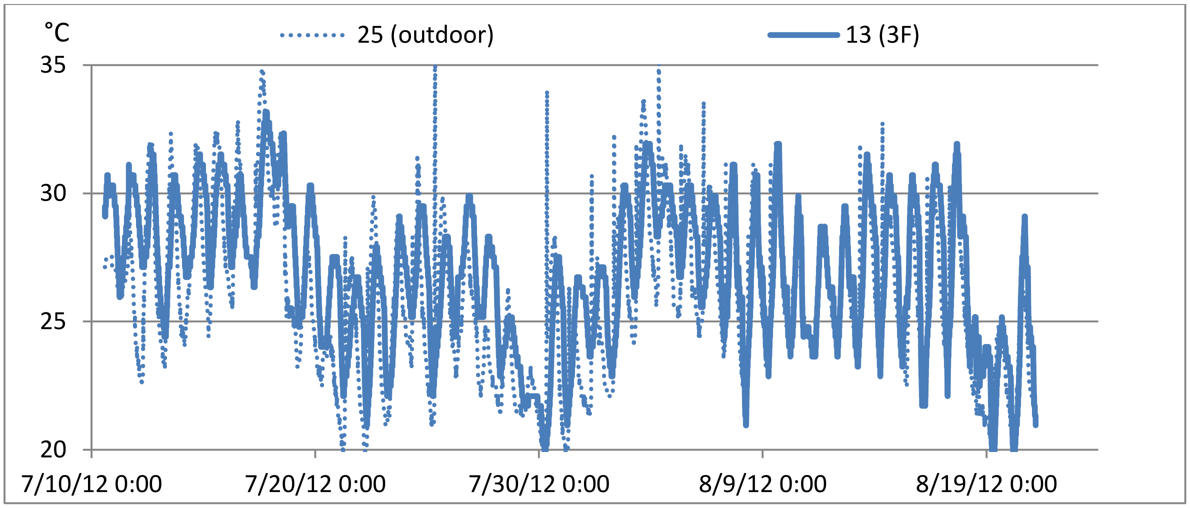


Figure 5-34. Frequency distribution of temperature from No.13

No.16 and No.24

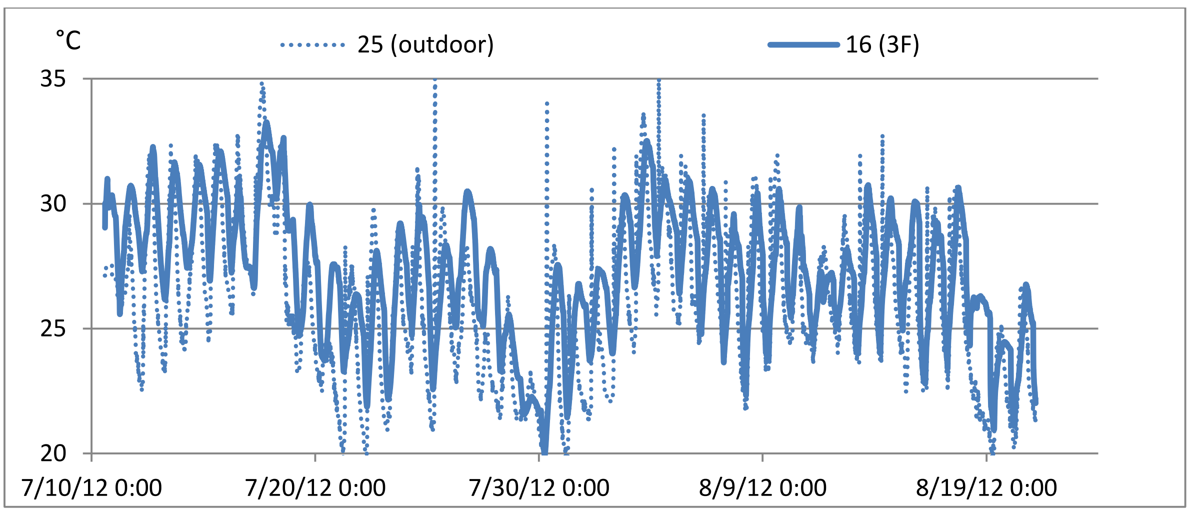


Figure 5-35. Frequency distribution of temperature from No.16

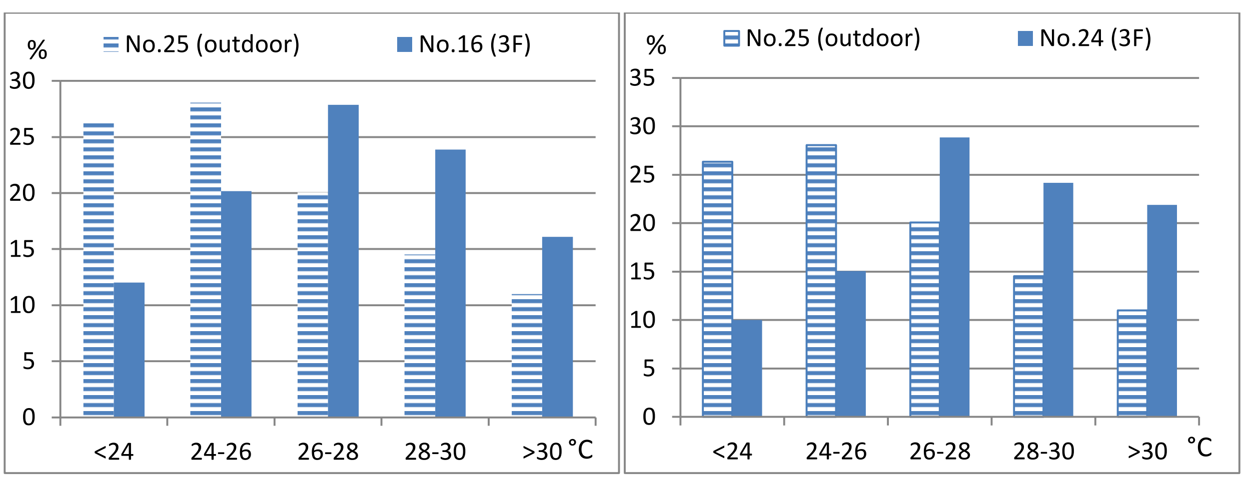


Figure 5-36. Frequency distribution of temperature from No.16 and No.24

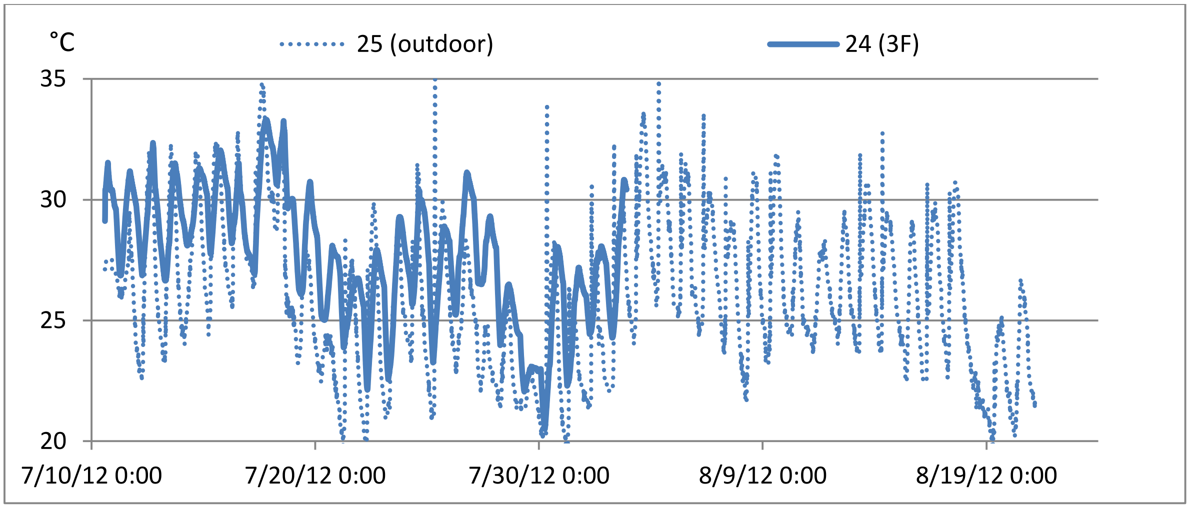


Figure 5-37. Frequency distribution of temperature from No.24

Discussion

Results show that comfort condition on the third floor is worse than outdoor condition and similar to office area on the second floor. This is because third floor has a high occupancy density when the space is scheduled. Additional fans (in additional to ceiling fans) have to be add to create internal air flow when the space is occupied by a number of people. The figure below shows is an example of the space being occupied. When the space is not in use, comfort condition is good because internal heat gain is very low.

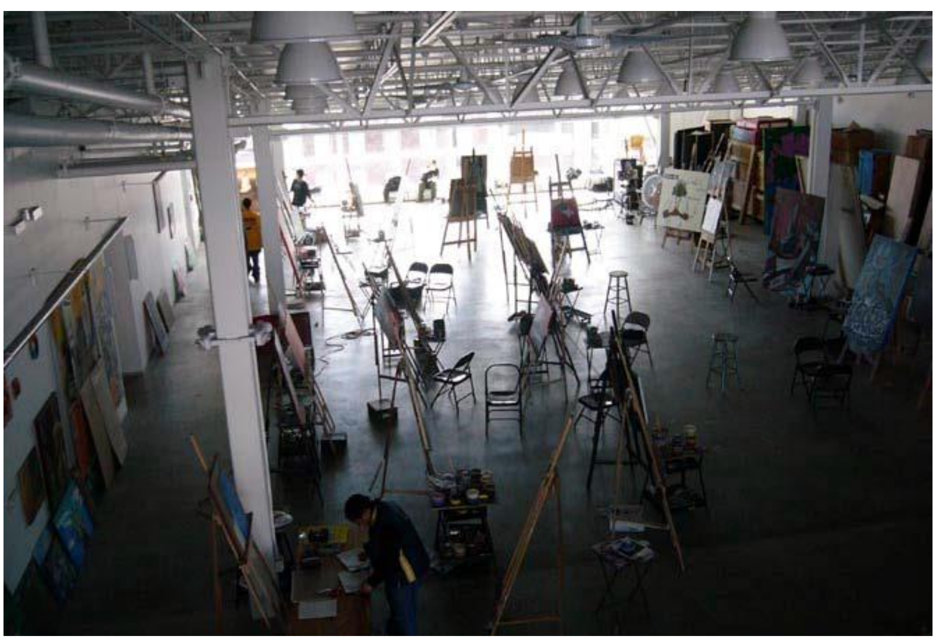


Figure 5-38. 3F space being occupied

5.4 Night Cooling Strategy

Sensors are placed in the vent of each floor. All three floors are connected to a central ventilation duct which is driven by the roof fan. When the fan is not in operation, vent temperature is close to indoor air temperature; when the fan is in operation, vent temperature is close to outdoor temperature due to high air flow

The figure below shows a typical example of night cooling by presenting a 24h temperature curve. Time period is from noon Aug. 15th to noon Aug. 16th. At daytime when the fan is not in operation, vent temperature is close to indoor temperature and exceeds outdoor temperature gradually because of internal heat gain. This trend lasts until midnight about 24:00 when the fan is turned on. Temperatures of the three vents drops suddenly due to the increased air flow from outdoor. During the entire night cooling period after 24:00, vent temperature is nearly equal to outdoor air temperature. When the fan is turned off at daytime, vent temperature is equal to indoor air temperature again.

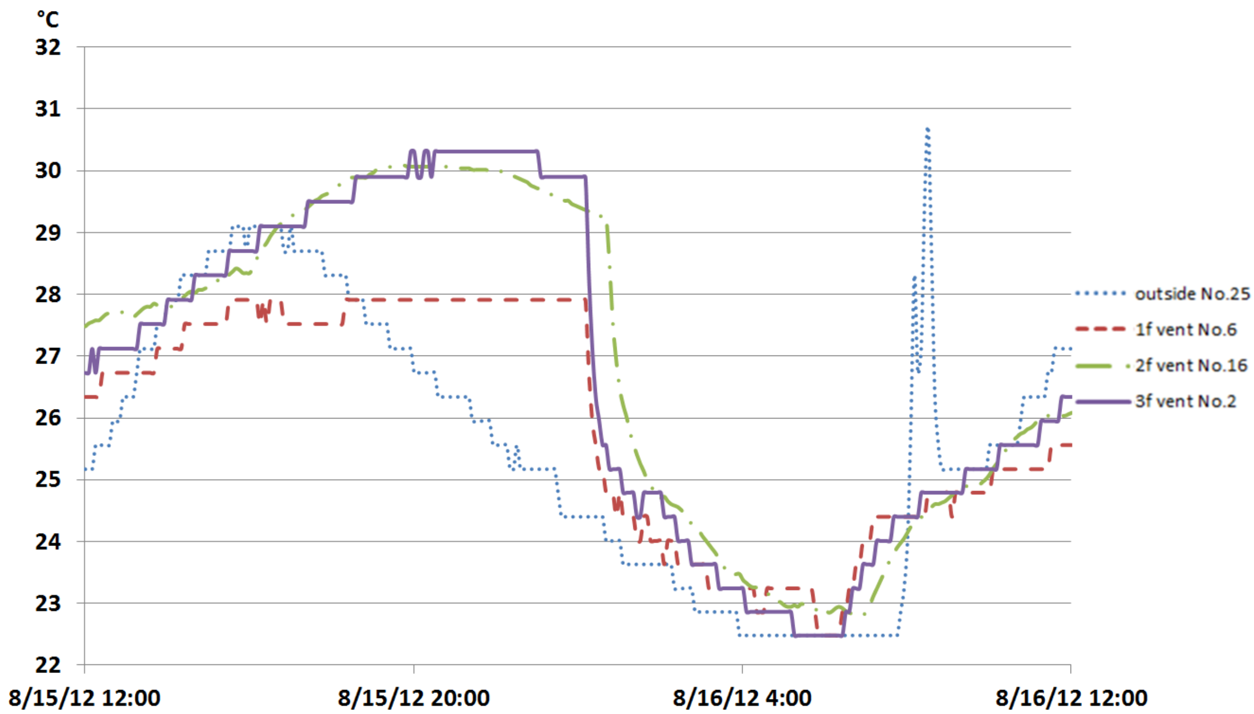
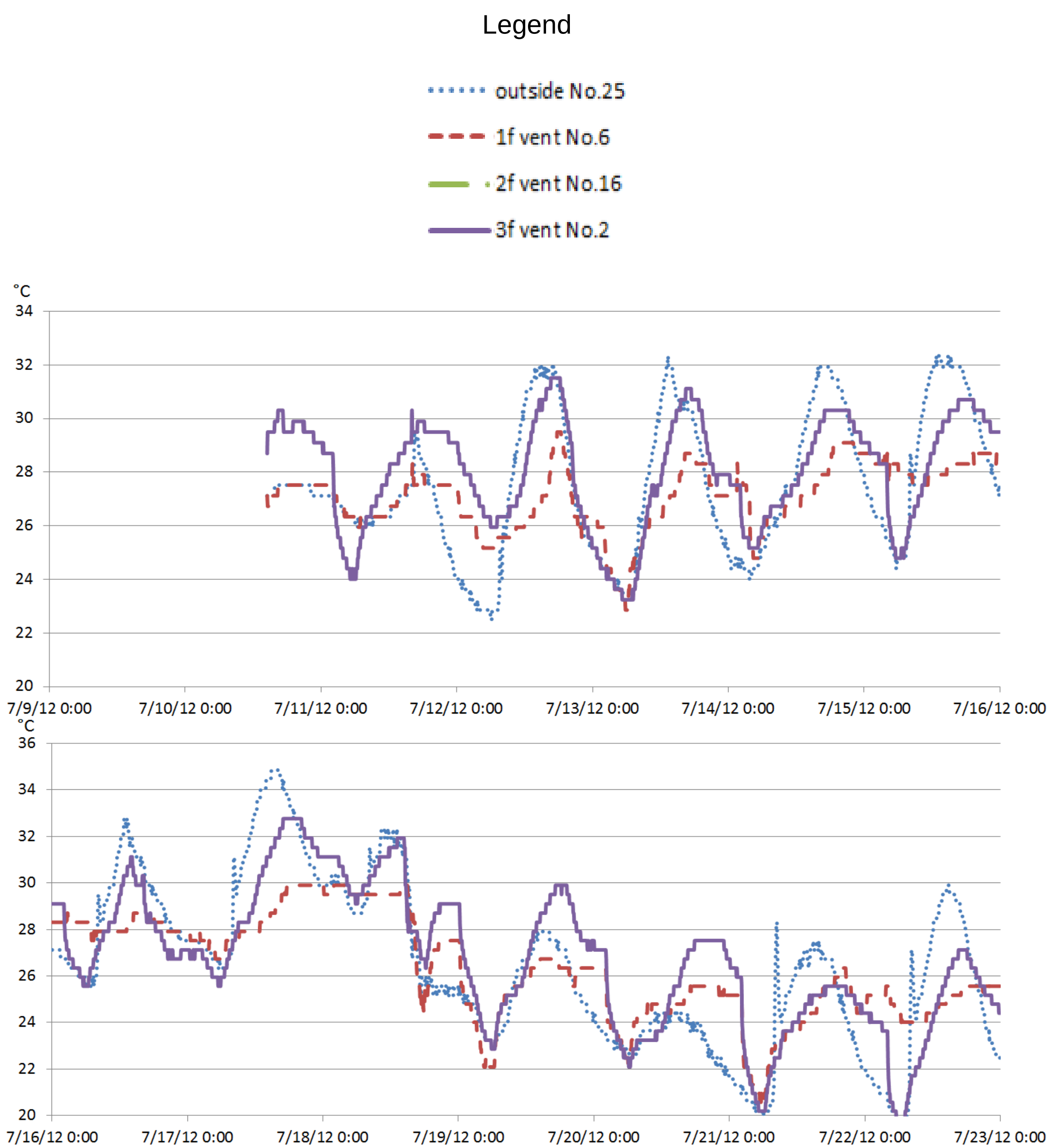
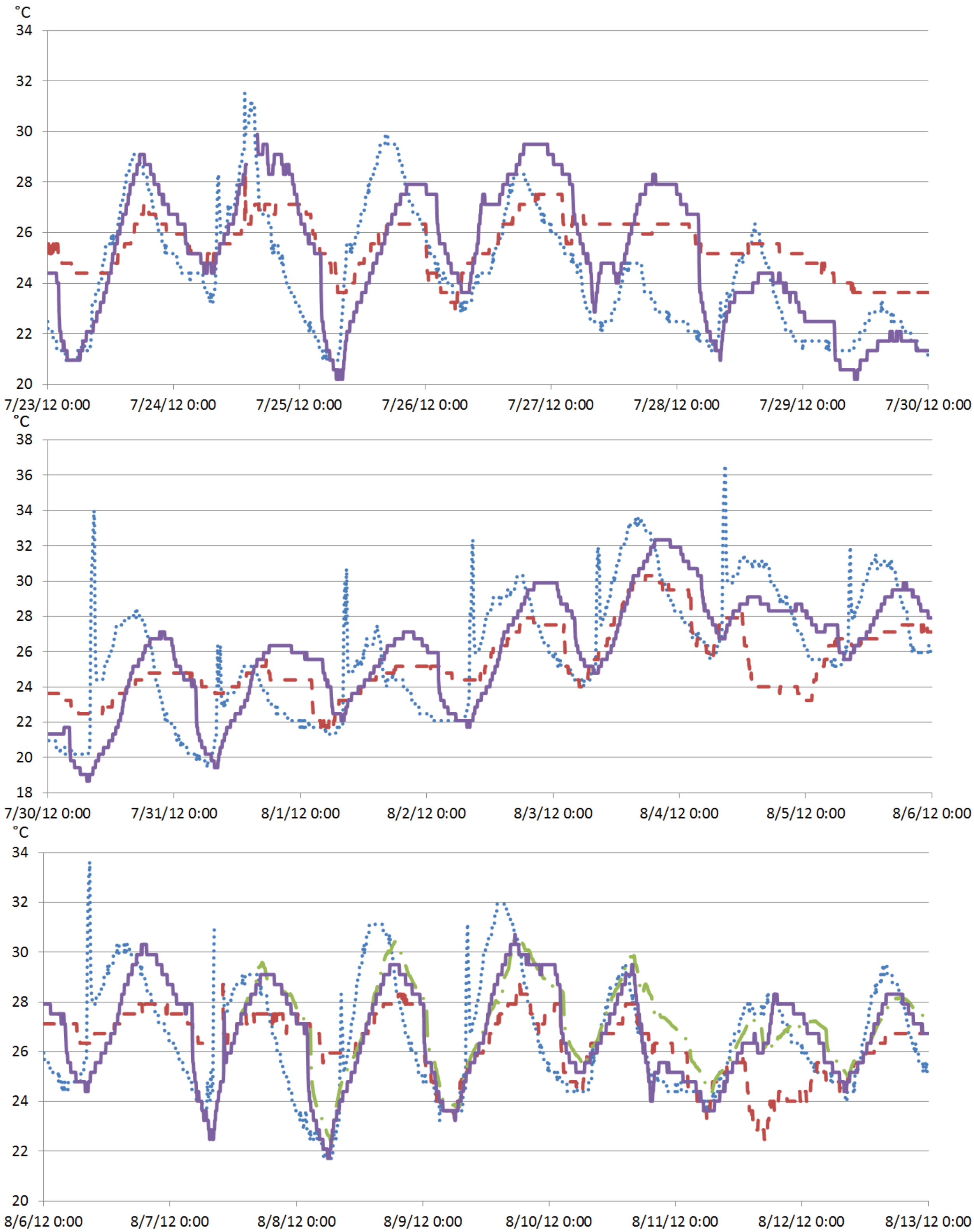


Figure 5-39. Typical daily temperature showing night cooling

The figure below summarizes this relationship of the entire monitoring period. It should be noted that there is an unreasonable peak of the outside temperature sensor happening every 24 hours. The reason for this is solar radiation at particular time of a day. Because a sensor is missing, ventilation temperature on the second floor is not complete.




°C
34
32
30
28
26
24
22
20
7/23/12 0:00
7/24/12 0:00
7/25/12 0:00
7/26/12 0:00
7/27/12 0:00
7/28/12 0:00
7/29/12 0:00
7/30/12 0:00
°C
38
36
34
32
30
28
26
24
22
20
18
7/30/12 0:00
7/31/12 0:00
8/1/12 0:00
8/2/12 0:00
8/3/12 0:00
8/4/12 0:00
8/5/12 0:00
8/6/12 0:00
°C
34
32
30
28
26
24
22
20
8/6/12 0:00
8/7/12 0:00
8/8/12 0:00
8/9/12 0:00
8/10/12 0:00
8/11/12 0:00
8/12/12 0:00
8/13/12 0:00

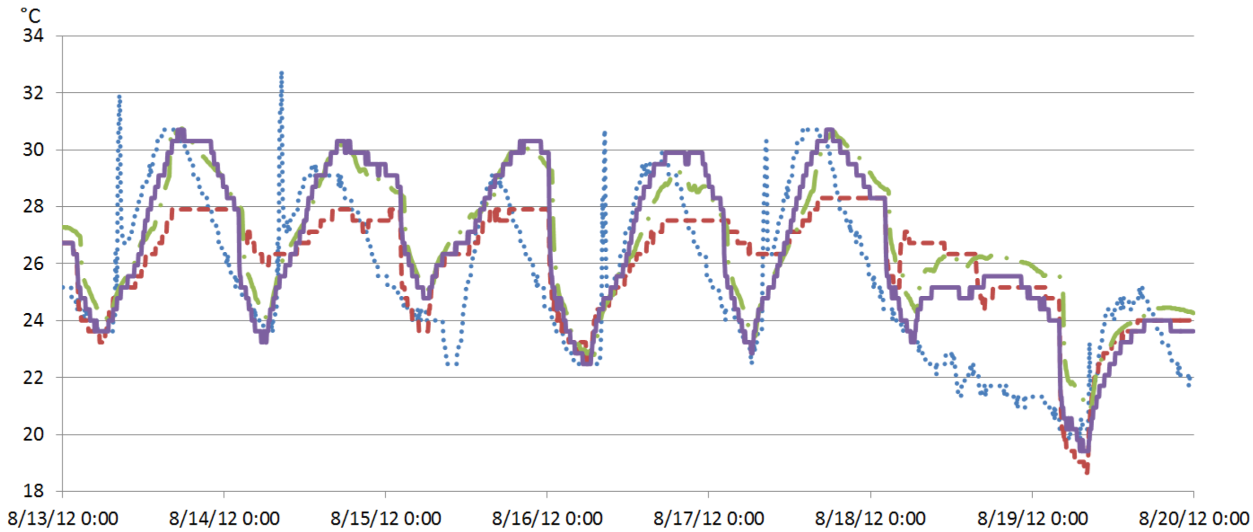


Figure 5-40. Relationship between outdoor and ventilation temperature

There is a timer controlling the roof fan. The timer is adjusted manually based on occupants' sensation. If occupants feel indoor air temperature uncomfortably high, the fan will set to be turned on earlier. Theoretically, when the fan is turned on, temperatures of the three vents will be close to outdoor air temperature soon because of the increased airflow. However, there are several exceptions showing inconsistency between temperatures on the three vents. Figure 5-41 below shows an example observed 2am July 25th.

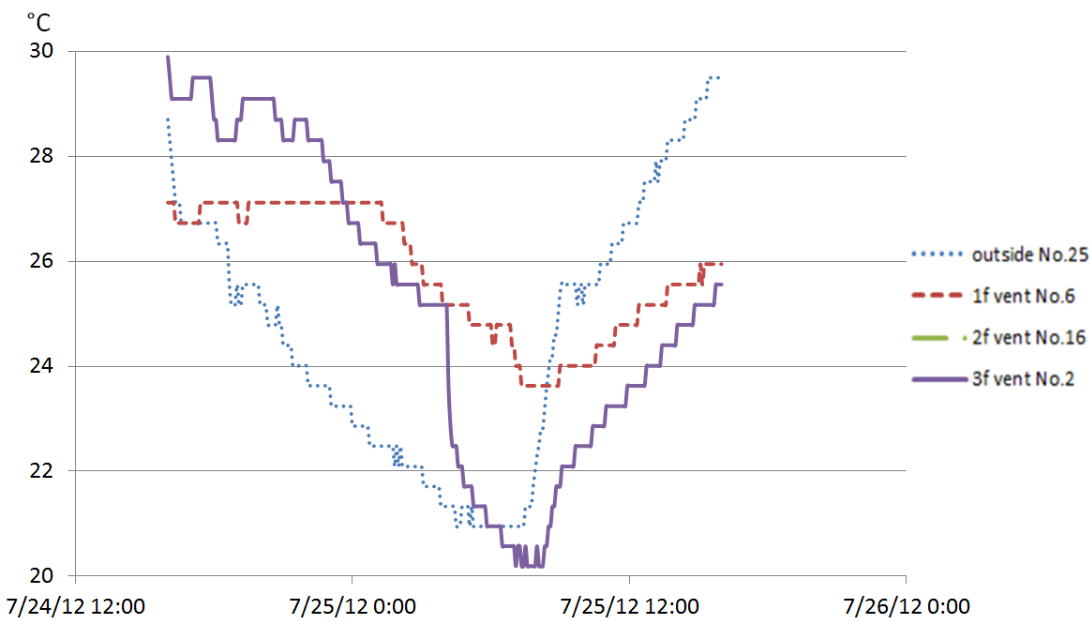


Figure 5-41. Typical daily temperature showing inconsistency between vent temperatures

The sudden drop of temperature of vent 3F shows that fan is turned on. However, temperature of vent on 1F did not drop simultaneously (potential reasons analyzed later). This is an example of inconsistency.

The chart below summarizes night cooling operation of the entire monitoring period. 'Time' is the time when the fan is turned on, which is represented by sudden drop of vent temperature of any floor. 'Y' or 'N' means whether ventilation on a certain floor is observed. Because sensor on 2F is missing and replaced at a later time, data on the 2F is not complete. For example, on Jul. 22nd, the fan is observed to be running on 3am. Ventilation is observed on 3F but not on 1F.

Time	1F vent	2F vent	3F vent	Time of fan on
7/11 Wed	Y	-	Y	1am
7/12 Thu	N	-	N	-
7/13 Fri	Y	-	Y	2am
7/14 Sat	Y	-	Y	3am
7/15 Sun	N	-	Y	2am
7/16 Mon	N	-	Y	1am
7/17 Tue	N	-	N	-
7/18 Wed	N	-	N	-
7/19 Thu	Y	-	Y	11pm
7/20 Fri	Y	-	Y	1am
7/21 Sat	Y	-	Y	1am
7/22 Sun	N	-	Y	3am
7/23 Mon	N	-	Y	0am
7/24 Tue	N	-	N	-
7/25 Wed	N	-	Y	2am
7/26 Thu	Y	-	Y	11pm
7/27 Fri	N	-	Y	3am
7/28 Sat	N	-	Y	3am
7/29 Sun	N	-	N	-
7/30 Mon	N	-	Y	3am
7/31 Tue	N	-	Y	1am

8/1 Wed	Y	-	Y	1am
8/2 Thu	N	-	Y	1am
8/3 Fri	Y	-	Y	0am
8/4 Sat	Y	-	Y	0am
8/5 Sun	N	-	N	-
8/6 Mon	N	-	Y	2am
8/7 Tue	N	-	Y	3am
8/8 Wed	N	Y	Y	0am
8/9 Thu	Y	Y	Y	0am
8/10 Fri	Y	Y	Y	0am
8/11 Sat	N	N	N	-
8/12 Sun	N	N	N	-
8/13 Mon	N	Y	Y	1am
8/14 Tue	N	Y	Y	1am
8/15 Wed	Y	Y	Y	1am
8/16 Thu	Y	Y	Y	0am
8/17 Fri	N	Y	Y	2am
8/18 Sat	Y	Y	Y	1am
8/19 Sun	Y	Y	Y	4am

Table 5-7. Night cooling of the entire monitoring period

Two potential reasons for this inconsistency are:

(1) Different internal resistance: the third floor is pretty empty and has the minimum internal resistance; the second floor might have the highest value of internal resistance because there are two separate rooms blocking the airflow; the first floor is also empty but only a single side window used for ventilation, whose effect is limited compared with the double side.

(2) The windows which should be open at night are closed. It can be seen from the table that ventilation effect of the second and third floor is identical; however, the effect on the first floor is comparatively limited. Although the night ventilation effect is not as obvious as it is for the other two floors, thermal condition for the first is still not bad because of its low level of internal load.

Another suggestion for operation is that sometimes fans could be turned on earlier to yield a better night cooling effect. Although temperature in the vent reaches the outside value quickly, it takes much more time to cool the building mass.

5.5 Slab Temperature

In this part we will analyze the relationship between slab and air temperature. This is important to natural ventilation at night in order to cool the thermal mass. The first floor has a low heat gain and cool temperature thus it might not be a good example of this analysis. Due to sensor loss, we did not get simultaneous vent, outdoor, indoor and slab temperatures for the second floor. Hence, the third floor is chosen for analysis. Data source is listed in the chart below.

No.	Location	Note
2	3F, fan vent	Ventilation temp.
9	3F, next to painting racks	Slab temp.
13	3F, metallic mezzanine on a column	Indoor temp.
25	Outdoor	Outdoor temp.

Table 5-8. Analysis for night cooling on thermal mass

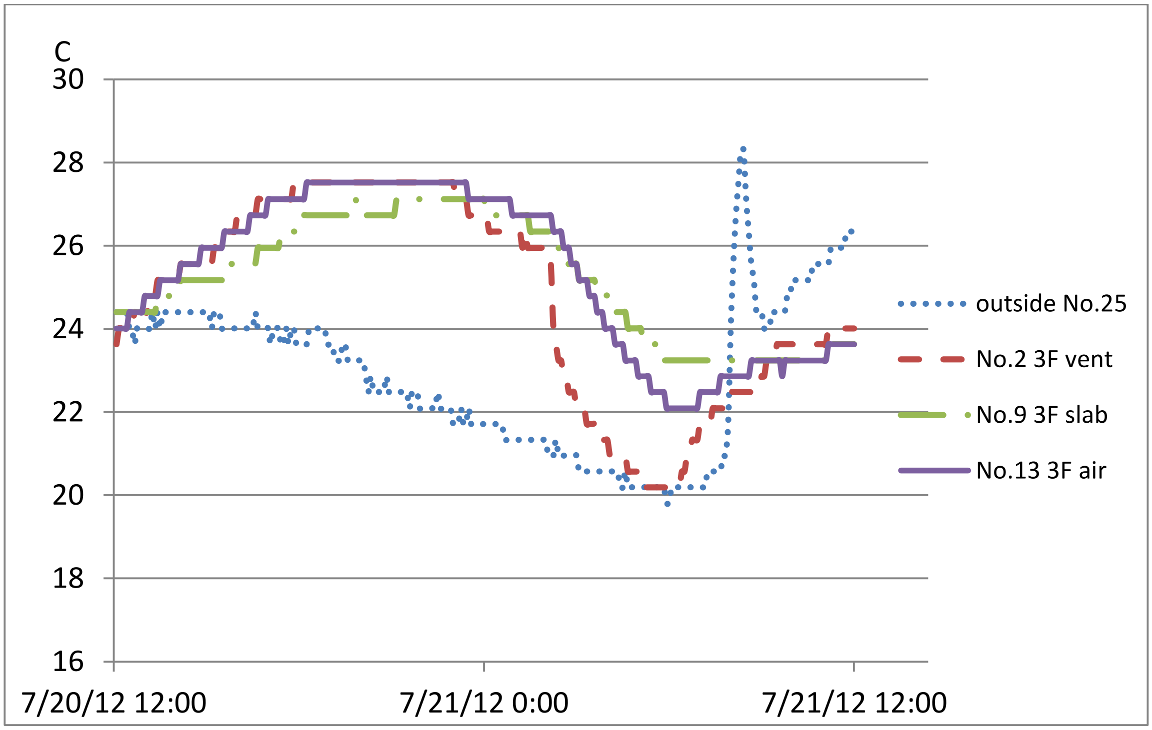
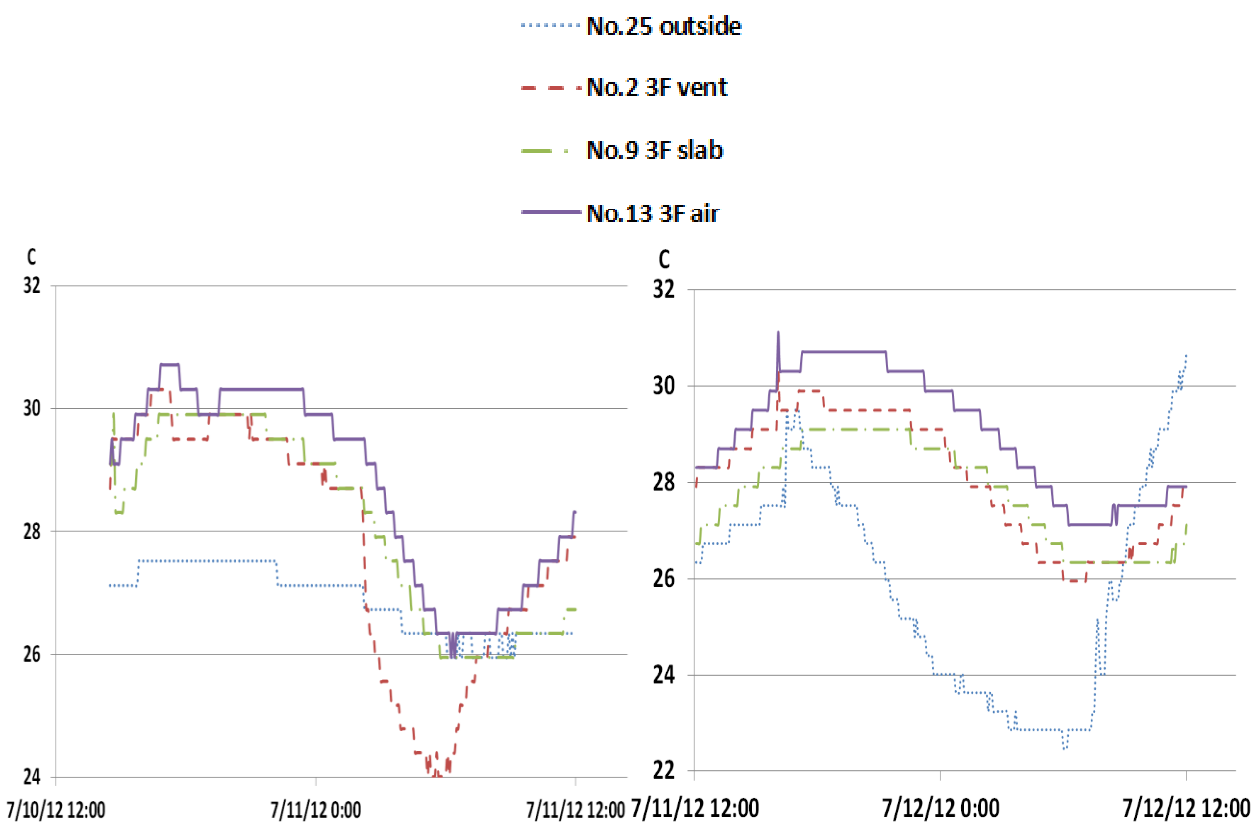


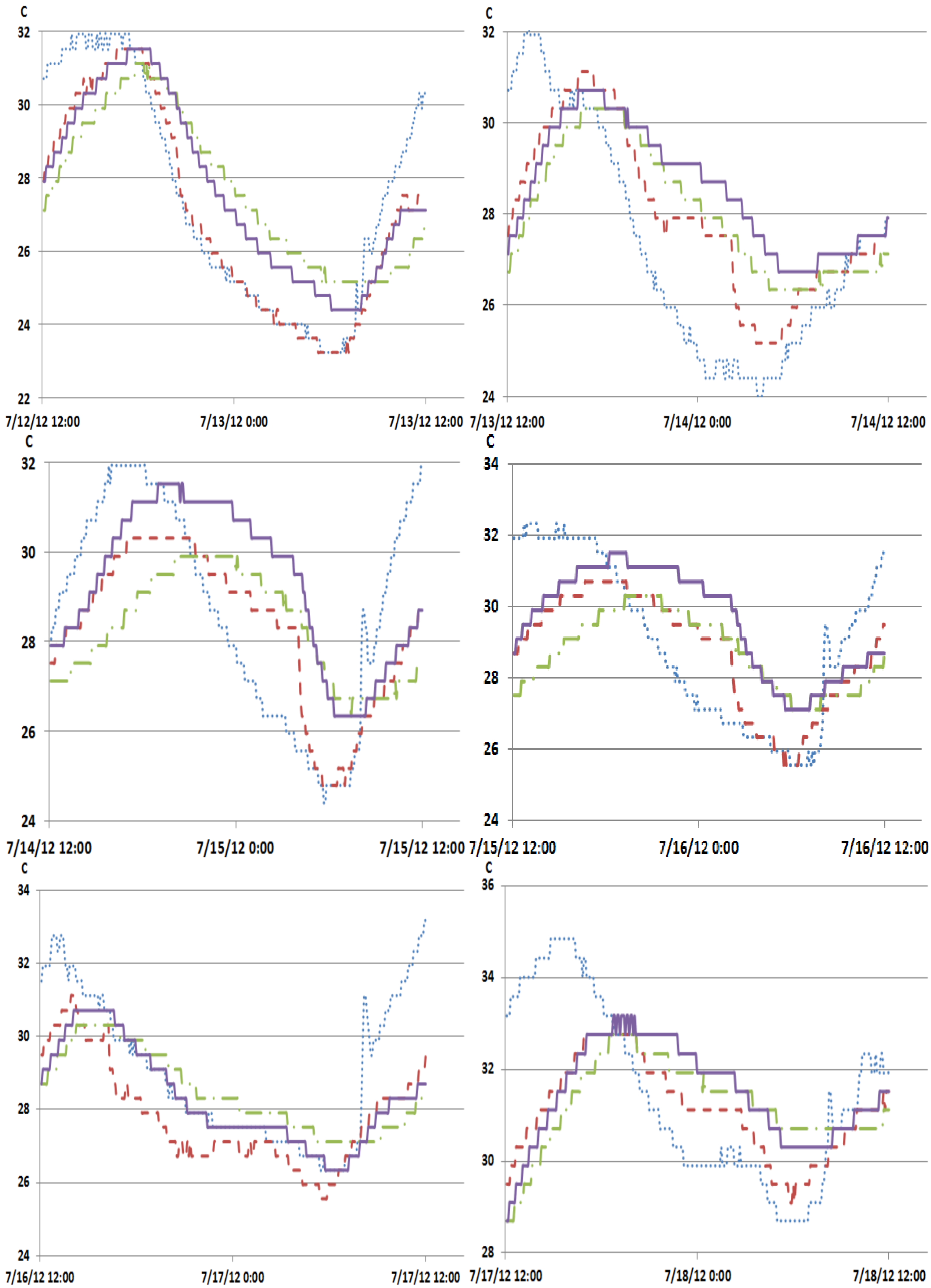
Figure 5-42. Example of night cooling effect on thermal mass

Here is an example showing the temperature of third floor's slab and temperature from July 20th noon to 21st noon. It can be shown during night time the fan is turned on, quickly lowering the temperature at the vent inlet to that of the outside. Then, the cooling outside air is ventilated to the indoor space, cooling the temperature of both the slab and the indoor air. Sensors measuring air and slab temperature are placed at the middle of the space's edge. Due to air flow pattern, indoor air temperature distribution is not uniform. In the space which is well ventilated, air temperature will be close to outdoor temp and vent temp very quickly. If the space is not well ventilated, there will be a lag between air temperature and outdoor/vent temp. At daytime when outdoor temperature rises and exceeds indoor one, by closing window the cooled thermal mass is capable of absorbing internal heat gain and maintain indoor temperature below outdoor one.

Daily relationship for third floor

Daily relationship for other days is plotted in the figure below. Similar results are observed. Legend is shown in the first part of the figure.




C
32
30
28
26
24
22
7/12/12 12:00
7/13/12 0:00
7/13/12 12:00
C
32
30
28
26
24
7/13/12 12:00
7/14/12 0:00
7/14/12 12:00
C
32
30
28
26
24
7/14/12 12:00
7/15/12 0:00
7/15/12 12:00
C
34
32
30
28
26
24
7/15/12 12:00
7/16/12 0:00
7/16/12 12:00
C
34
32
30
28
26
24
7/16/12 12:00
7/17/12 0:00
7/17/12 12:00
C
36
34
32
30
28
7/17/12 12:00
7/18/12 0:00
7/18/12 12:00

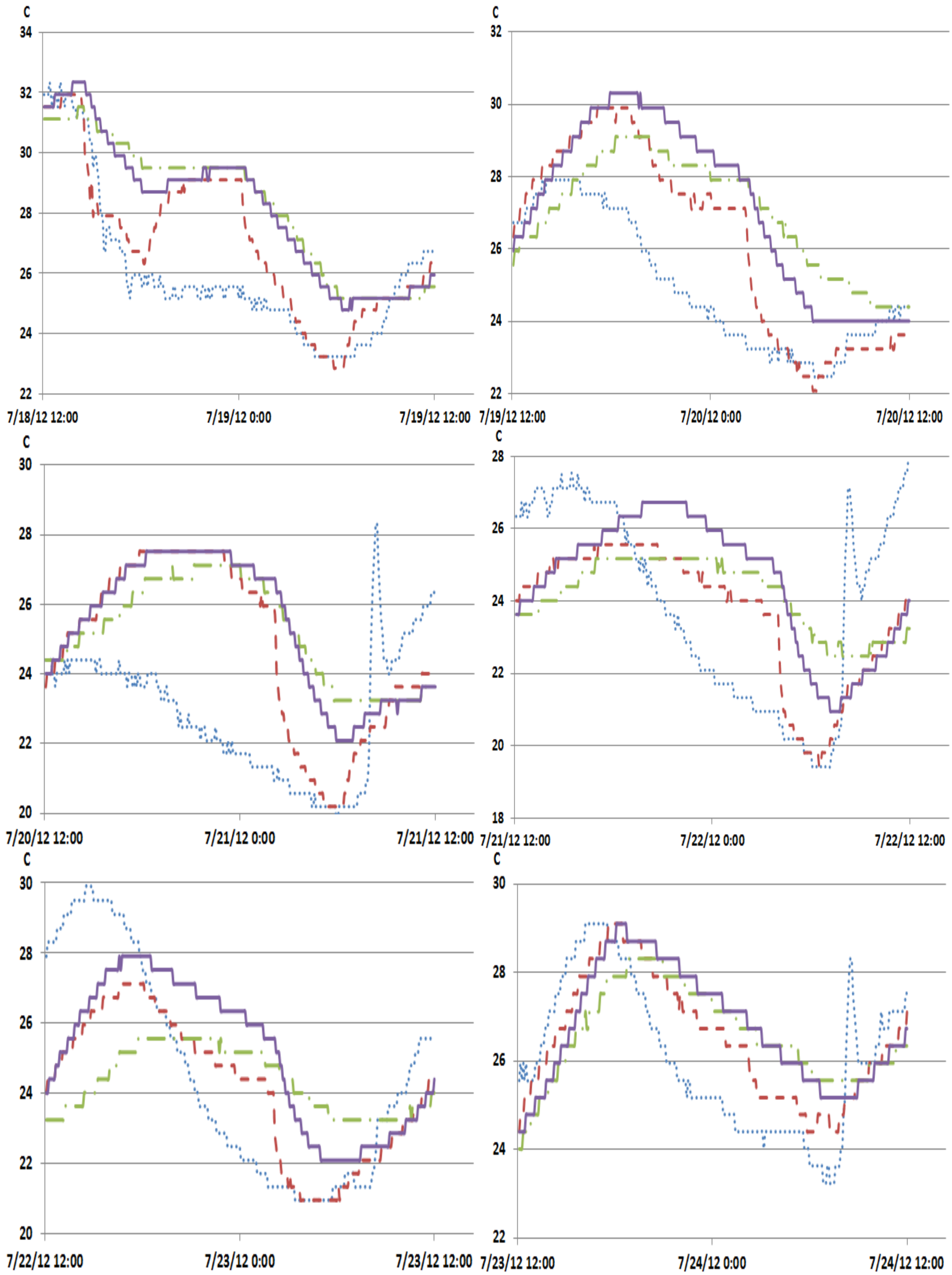


Figure 5-43. Daily relationship of night cooling

5.6 Airflow Analysis

Airflow of the building has been measured on Aug 21st on the third floor. Our intention is to compare this with the rated flow of fan so that airflow rate could be estimated in further simulation.

Measurement from window side

These four modes combine single/double opening and fan on/off. The results are shown below.

Mode	Average velocity (m/s)	Cross section area (m^2)	Flow rate (m^3/s)
Double side open, fan	3.37	1.28	4.31
Single side open, no fan	0.58	0.64	0.37
Single side open, fan	3.50	0.64	2.24
Double side open, no fan	1.50	1.28	1.92

Table 5-9. Total airflow measurement from windows of the third floor

Measurement from vent inlet

The cross section of the fan vent is divided into several sub-sections with equal area and the air velocity is measured. The direction of the air flow was found to be 60 degrees from the normal .The resulting normal component of the velocity is listed below.

Mode	Average normal velocity (m/s)	Cross section area (m^2)	Flow rate (m^3/s)
Double side, fan	3.26	1.60	5.21

Table 5-10. Airflow measurement from vent inlet

Discussion

There is a discrepancy between results from window and vent inlet. An important reason is that airflow pattern at the vent's inlet is not parallel. The inlet is close to a perpendicular duct. Hence, it might be hard to accurately measure air velocity perpendicular to the cross section. In comparison, air flow path and velocity profile at the side of window opening is more regular and easier to get accurate measurement. A figure of sketch showing floor geometry is attached below.

Rated total airflow rate of the fan is about 10.5m^3/s, the rated airflow per floor is 3.5m^3/s, which is about 20 percent less than the measurements from the window side. By assuming similar flow resistance of the three floors, the fan could be regarded as operating at rated condition. The slight difference between simulated and measured air flow rate has a negligible effect on convective heat transfer (between thermal mass and airflow) used in the simulation. Thus 3.5m^3/s is used in the simulation.

Figure 5-44. Opening geometry of the building

5.7 Electricity Consumption

Annual electricity consumption data is shown in the figure below.

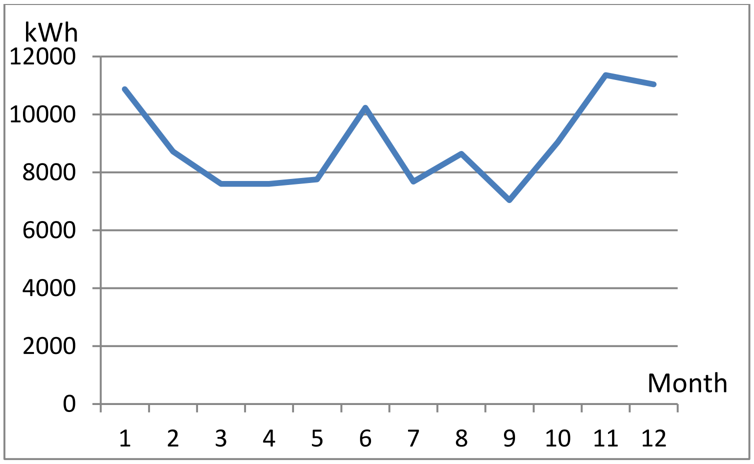


Figure 5-45. Monthly electricity consumption

It can be seen from the figure that electricity consumption peaks in winter and remains low in other time. The reason of this difference is that potential of natural lighting in summer is limited, although in summer fans also consume an addition part of energy. The energy consumption of the exhaust and ceiling fans needs estimation. Their rated power consumptions are attached below.

Ceiling fan

Mode	Power / W	Flow rate / CFM
Low	17.1	3447
Medium	35.6	5455
High 1	80.1	8188
High 2	107.1	10195

Table 5-11. Rated condition of ceiling fan

Roof fan

Power / kW	2.1
Flow rate / m^3 per s	10.5

Table 5-12. Rated condition of roof fan

Several assumptions concerning this estimation are:

(1) The power of the exhaust fan is set to be its rated power.

(2) The ceiling fans on the second floor are set to be running constantly on High 1 mode (according to the building manager) on office hour. Ceiling fans on other floors are rarely used. Hence only consider ceiling fans on the second floor, there are about 16 fans in total.

(3) The operating hour of the exhaust fan is derived from the temperature measurement (the roof fan is estimated to be turned when temperature of any vent drops suddenly at night). From the results we can get the time the fan is turned on at night. Its shut down hour is set to be 6am.

(4) The building is considered to be entirely shut down on weekends. The monthly electricity consumption of the exhaust fan is about 251 kWh. Energy consumption of the ceiling fans is estimated to be 290 kWh. Energy consumed by fans is about 7% of the total monthly consumption.

5.8 Thermal Comfort

We have distributed a questionnaire to occupants on the second floor, the floor which has the highest indoor temperature in summer due to high internal load (Note: occupants in the two separate rooms are not included). The questionnaire was distributed on October and could represent occupants' sensation for the past summer. Four basic questions are included and results are shown below.

Question 1: generally speaking, indoor temperature is:

Question 2: generally speaking, indoor humidity is:

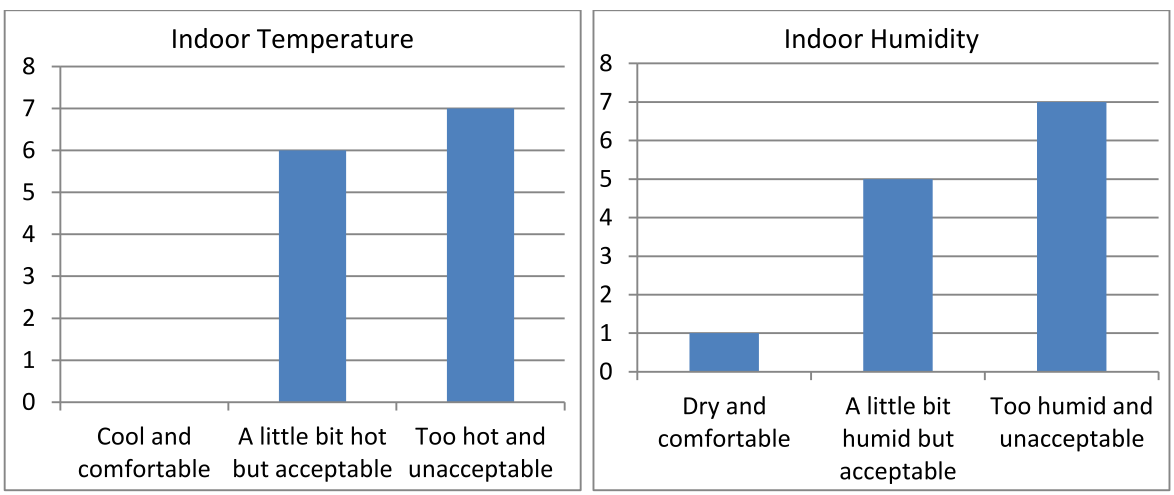


Figure 5-46. Occupants' response to question 1 and 2

Question 3: you think working efficiency in summer is:

Question 4: you think conventional air-conditioning for the working area is:

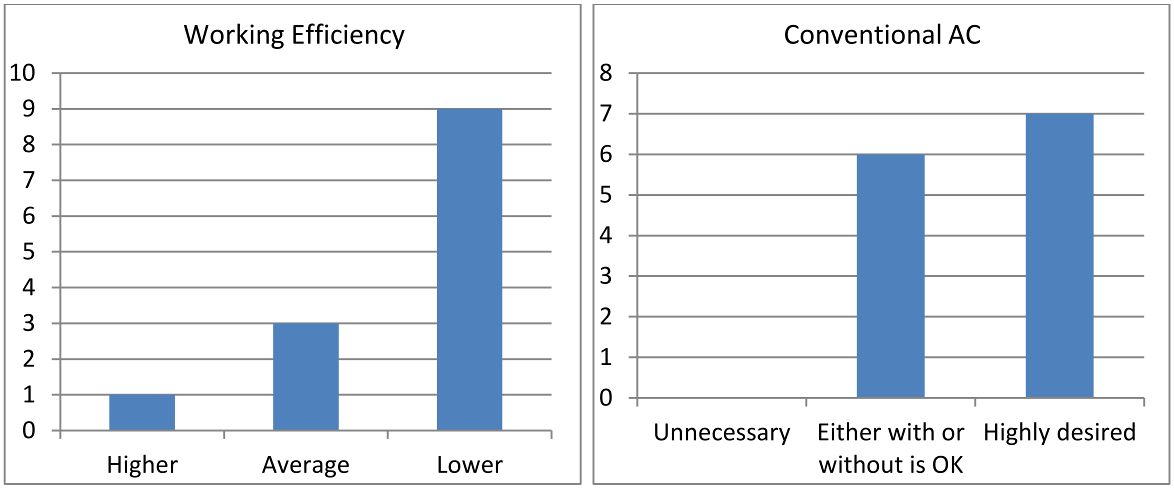


Figure 5-47. Occupants' response to question 3 and 4

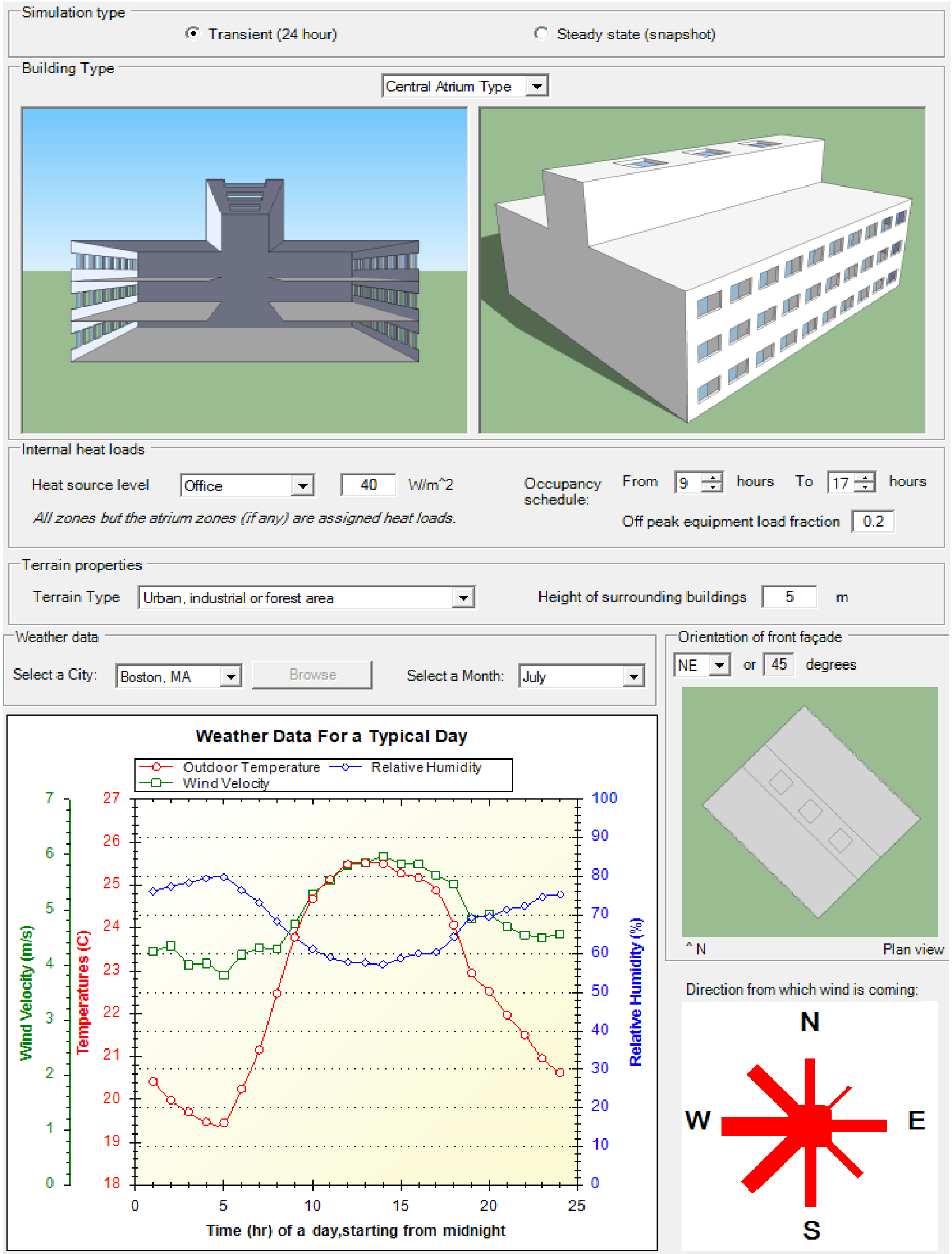
The above results show that occupants' thermal sensation ranges between acceptable and unacceptable, while almost no one feels comfortable in summer on the second floor. Many occupants feel working efficiency lowered in summer time and half occupants want a supplementary mechanical cooling system installed. The results reveal that energy efficiency is achieved at the cost of occupant comfort and some improvement of the current system should be proposed.

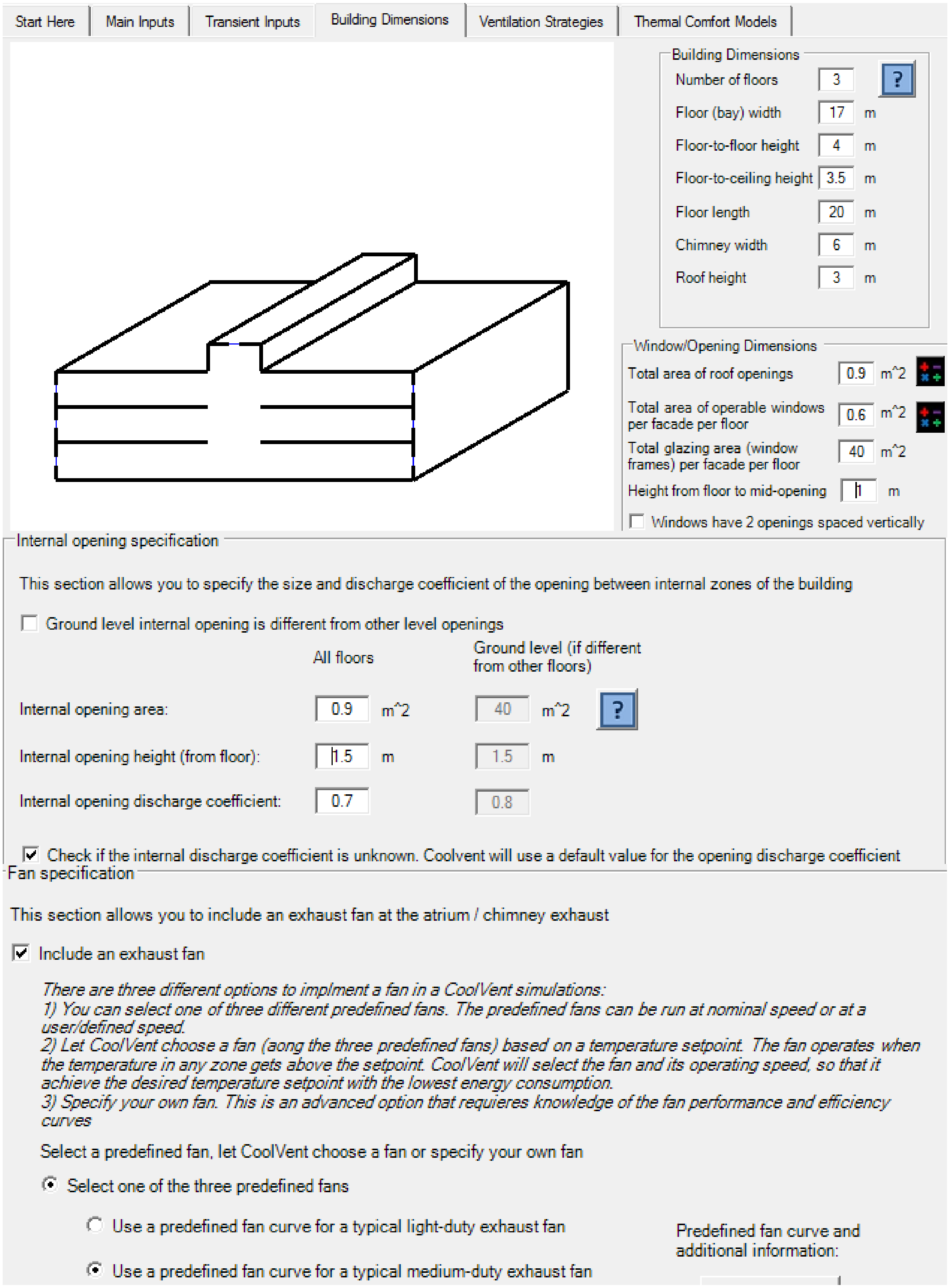
5.9 Simulation in CoolVent

The building is simulated in Coolvent as a three-story model. Internal heat gains of different floors vary a lot: the first floor is almost never occupied and has highest level comfort; the second floor is the main working area and has high internal heat gain and lowest comfort percentage; the third floor varies based on schedule (whether where will be activities, can be zero to very high).The building model assumes zero internal heat gain for the first floor and maximum internal gain for the third floor (40 W/m^2 based on estimation of occupants and utility). Internal heat gain on the second floor is relatively constant (40 W/m^2 based on estimation of occupants and utility).

There are two separate and closed rooms in the second floor which are used for manufacturing and painting. Due to high internal heat gain, separate exhaust fans and air conditioners are used in these two rooms. In this model, we will neglect the little amount of heat conduction from these two rooms to the other part of the floor, which is our major concern.

5.9.1 Building Model





Start Here
Main Inputs
Transient Inputs
Building Dimensions
Ventilation Strategies
Thermal Comfort Models
Building Dimensions
Number of floors 3
Floor (bay) width 17 m
Floor-to-floor height 4 m
Floor-to-ceiling height 3.5 m
Floor length 20 m
Chimney width 6 m
Roof height 3 m
Window/Opening Dimensions
Total area of roof openings 0.9 m^2
Total area of operable windows per facade per floor 0.6 m^2
Total glazing area (window frames) per facade per floor 40 m^2
Height from floor to mid-opening 1 m
Windows have 2 openings spaced vertically
Internal opening specification
This section allows you to specify the size and discharge coefficient of the opening between internal zones of the building
Ground level internal opening is different from other level openings
All floors
Ground level (if different from other floors)
Internal opening area: 0.9 m^2 40 m^2
Internal opening height (from floor): 1.5 m 1.5 m
Internal opening discharge coefficient: 0.7 0.8
Check if the internal discharge coefficient is unknown. Coolvent will use a default value for the opening discharge coefficient
Fan specification
This section allows you to include an exhaust fan at the atrium / chimney exhaust
Include an exhaust fan
There are three different options to implment a fan in a CoolVent simulations:
1) You can select one of three different predefined fans. The predefined fans can be run at nominal speed or at a user/defined speed.
2) Let CoolVent choose a fan (aong the three predefined fans) based on a temperature setpoint. The fan operates when the temperature in any zone gets above the setpoint. CoolVent will select the fan and its operating speed, so that it achieve the desired temperature setpoint with the lowest energy consumption.
3) Specify your own fan. This is an advanced option that requieres knowledge of the fan performance and efficiency curves
Select a predefined fan, let CoolVent choose a fan or specify your own fan
Select one of the three predefined fans
Use a predefined fan curve for a typical light-duty exhaust fan
Use a predefined fan curve for a typical medium-duty exhaust fan
Predefined fan curve and additional information:

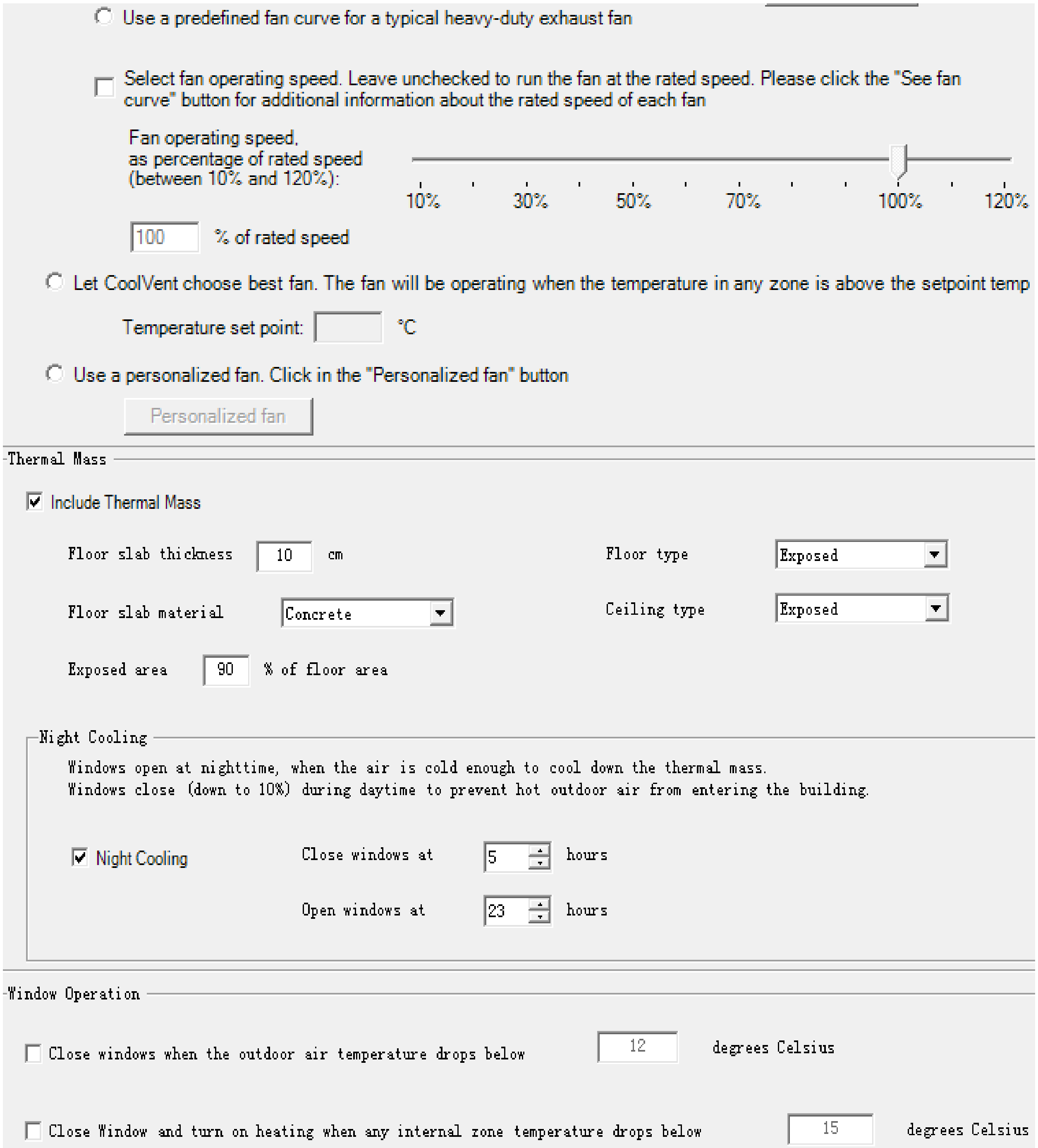


Figure 5-48. Building model in CoolVent

5.9.2 Verification of Fan

To get accurate simulation result, airflow in simulation and measurement need to be close to each other. The comparison is made in this section. Several key parameters that need to be listed are:

(1) Total area of roof openings 0.9 m^2

This is equal to the area of a medium-size fan (the blue one in the figure below). This fan is very similar to the fan in the building, which has a rated flow rate of $10m^3/s$.

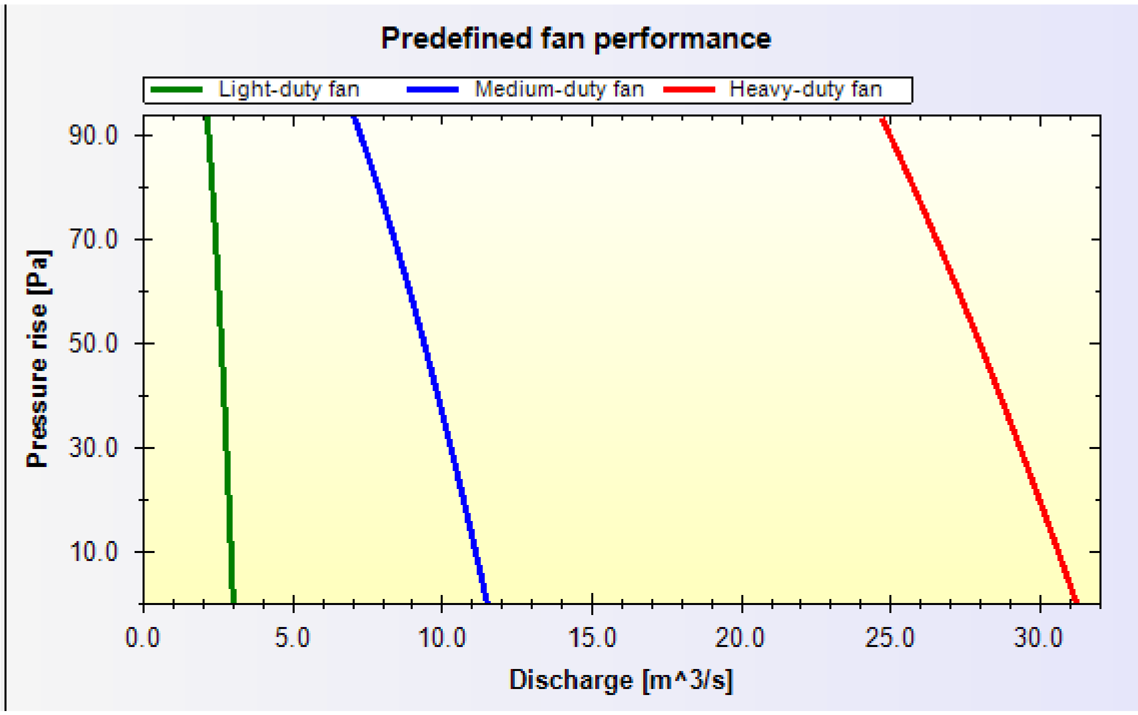


Figure 5-49. Fan curve in CoolVent simulation

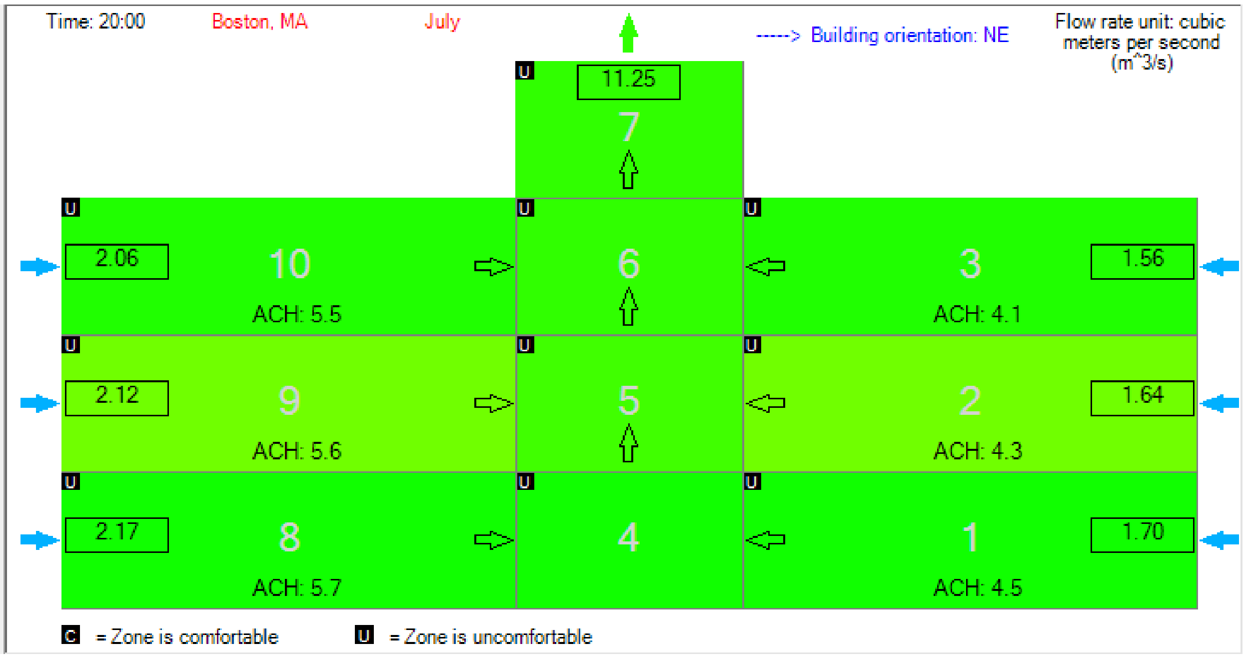
(2) Total area of operable windows per façade per floor 0.6 m^2

From the measurement result, total operable windows area per façade per floor.

(3) Internal opening area 0.9 m^2

This is in correspondence with the third floor measured. In that floor there is almost no internal resistance inside the space. This internal opening area is based on opening area of the ventilation duct, which has a total area of 1.8 m^2. Since it is central atrium building type, the corresponding internal opening area is half of this value and equals to 0.9 m^2.

Simulation result figure is shown in the following figures. Flow rate in different hours when fan is in operation are listed in the following table (a 24 hour typical day is simulated).



Time	Left side flow rate (m^3/s)	Right side flow rate (m^3/s)	Total flow rate (m^3/s)
0am	1.60	1.48	3.08
1am	1.60	1.48	3.08
2am	1.60	1.48	3.08
3am	1.59	1.48	3.07
4am	1.59	1.48	3.07
5am	1.59	1.49	3.07
6am	1.58	1.47	3.05
7am	1.58	1.46	3.04
8am	1.51	1.48	2.99

9am	1.59	1.43	3.02
10am	1.41	1.60	3.01
11am	1.41	1.61	3.02
12pm	1.40	1.62	3.02
1pm	1.62	1.40	3.02
2pm	1.40	1.62	3.02
3pm	1.40	1.62	3.02
4pm	1.41	1.62	3.02
5pm	1.42	1.61	3.03
6pm	1.44	1.60	3.04
7pm	1.48	1.51	2.99
8pm	1.59	1.45	3.04
9pm	1.58	1.45	3.03
10pm	1.60	1.47	3.07
11pm	1.52	1.49	3.01

Table 5-13. Simulation of fan in CoolVent

Due to wind and buoyancy effect, airflows on both sides of the building are not equal. However, the fan is capable of keeping the total airflow rate constant, independent of weather condition. This simulation adjusts the fan size to equal the actual rated fan capacity for the building. This value is 3.5 m^3/s.

One important condition for this assumption is the interaction of different floors. In the real building, there is very limited interaction between floors when fan is in operation: outdoor air is driven into the room and then into the central duct; there is heat transfer between floors through slab but the amount is very small. In the building model, it is necessary to evaluate airflow pattern of the central atrium model.

Airflow simulation for a typical day is attached in the figure below. Weather data is monthly average one for Boston in July. Monthly wind direction and velocity data for a specified time is averaged using the following method: firstly the direction with the highest frequency is determined; then velocities of that direction are averaged. Other parameters are arithmetic average value.

Results show that interactions between floors are limited. For the second floor, all air coming from the first floor passes through the central atrium directly without entering zone 2 or zone 9. This means that airflow from other floors will not affect temperature distribution of the second floor (zone 2 and zone 9). The only effect of setting uniform internal load might be air temperature in zone 5 might be higher than it actual value because of the increased internal load on the first floor. The relative effect of this compared with the fan could be estimated by calculation below.

Assuming a 5°C temperature rise due to increased thermal load, the corresponding density difference could be estimated with the following chart.

Temperature / °C	**Density / kg/m^3**
0	1.293
10	1.247
20	1.205
30	1.165

Table 5-14. Air density at standard atmosphere pressure

The corresponding density could be estimated to be 0.03 kg/m^3. Then the pressure difference could be calculated.

$$\Delta P = \frac{0.03kg}{m^3} \times 2m \times \frac{9.8m}{s^2} = 0.6Pa$$

This value is negligible compared with the fan's pressure (25 Pa corresponding to airflow rate 10.3 kg/m^3).

5.9.3 Comparison between Hybrid Mode and Air-conditioning Mode

This section aims to evaluate the potential of natural ventilation for this building under Boston weather condition. This potential could be reflected by the percentage of energy saving resulted if a hybrid use of natural ventilation and air conditioning was used rather than the purely air conditioning. Firstly, pure air conditioning load (thermal energy in kWh) is calculated and results are listed below.

Month	Thermal load (kWh)
May	3362
Jun	5307
Jul	11840
Aug	9673
Sep	6529
Oct	771

Table 5-15. Monthly cooling load (thermal energy in kWh)

We will focus on hybrid mode simulation on June, July and August, months in which cooling energy peaks. The building is operated initially at a natural ventilation mode. When indoor environment cannot meet the comfort standard, windows are turned off and the mode is switched into air conditioning one. The control logic is same as it is in chapter 4.5.

Different comfort standards are used. For natural ventilation the set upper limit value is 28°C for temperature and 14 g/kgda for humidity ratio. For air conditioning mode the limit is stricter: 26°C for temperature and 12 g/kgda for humidity ratio.

The electrical energy in kWh for hybrid mode is summarized below. COP of the chiller is assumed to be 3 (electricity consumption = thermal energy / COP). The fan is operated at the rated power.

Month	Hybrid chiller	Hybrid fan	Hybrid total
May	0	218	218
Jun	293	190	483

Jul	2608	135	2743
Aug	1114	155	1269
Sep	14	225	239
Oct	0	467	467

Table 5-16. Electricity consumption of the hybrid mode

Comparison between the hybrid mode total and air conditioning mode is summarized below.

Month	Hybrid total	AC total	Saving Percentage
May	218	1121	80%
Jun	483	1769	72%
Jul	2743	3947	31%
Aug	1269	3224	60%
Sep	239	2176	89%
Oct	467	257	-

Table 5-17. Comparison between hybrid and air conditioning mode (*since hybrid mode consumes more energy in October, it is not used at that month)

5.9.4 Impact of Night Cooling

In actual monitoring period, the building is ventilated during night period and closed at daytime. An important issue is that to what extent the night ventilation helps reduce cooling load at daytime. To evaluate this effect, we will simulate night cooling in CoolVent.

The strategy used here is air-conditioning with fan-assisted night cooling. Windows are open and the building is ventilated with fan at night; at day time it is closed and cooling load is calculated as a measurement of the effect of night cooling. The impact of night ventilation duration on fan energy at night and air conditioning energy at day time is added up. The results enable us comparing the trade-off between fan and chiller energy and deciding the optimal operation hours.

The result is listed below. The number is the electricity consumption (in kWh) of chiller and fan (The chiller COP is assumed to be 3). Numbers in the left column represent the time that windows are open for fan-assisted night cooling (21, 22 and 23).

Numbers in the upper row represent the time that windows are shut down at the end of night cooling.

	chiller	fan	chiller	fan	chiller	fan	chiller	fan	
	3:00		4:00		5:00		6:00		
21:00	125	232	77	270	46	309	27	348	May
	686	214	612	249	562	285	534	320	Jun
	3332	243	3220	283	3113	323	3015	364	Jul
	1868	233	1677	272	1543	311	1450	350	Aug
	294	236	184	276	122	315	88	355	Sep
22:00	185	193	115	231	69	270	42	309	May
	759	178	671	214	610	249	575	284	Jun
	3392	202	3278	243	3163	283	3062	323	Jul
	2025	194	1805	233	1647	272	1537	311	Aug
	429	197	261	237	169	276	123	315	Sep
23:00	272	154	175	193	106	232	65	270	May
	860	143	746	178	672	213	627	249	Jun
	3476	162	3343	202	3228	242	3117	283	Jul
	2212	156	1958	195	1769	233	1640	272	Aug
	627	158	390	197	241	237	172	276	Sep

Table 5-18. Electricity consumption of chiller and fan with night cooling mode

To find optimal operation hours for each month, add the electricity consumption of chiller and AC and compare monthly results. The optimal hour (corresponding to minimum energy consumption) is marked in bold.

	Time	3:00	4:00	5:00	6:00
May	21:00	357	347	355	375
Jun		900	861	**847**	854
Jul		3575	3503	3436	**3379**
Aug		2101	1949	1854	**1800**
Sep		530	460	**437**	443
May	22:00	378	346	339	351
Jun		937	885	859	859

Jul		3594	3521	3446	3385
Aug		2219	2038	1919	1848
Sep		626	498	445	438
May	23:00	426	368	338	**335**
Jun		1003	924	885	876
Jul		3638	3545	3470	3400
Aug		2368	2153	2002	1912
Sep		785	587	478	448

Table 5-19. Total electricity consumption with night cooling mode

Compare minimum chiller electricity consumption with that from pure air-conditioning without night cooling. The saving percentage reflects sensible cooling load reduction due to night cooling and could be representative of comfort condition at daytime. It shows that the potential of night cooling at July is minimal. Hence, comfort condition during July (which is the monitoring period) is not good.

	min.	**pure ac**	**Saving percentage**
May	27	1121	98%
Jun	534	1769	70%
Jul	3015	3947	24%
Aug	1450	3224	55%
Sep	88	2176	96%

Table 5-20. Minimum chiller electricity consumption and pure air conditioning

Compare total electricity consumption from optimal operation with that from pure air-conditioning without night cooling. (Ncool represents night cooling)

	Optmal Ncool	**No Ncool**	**Saving percentage**
May	335	1121	70%
Jun	847	1769	52%
Jul	3379	3947	14%
Aug	1800	3224	44%
Sep	437	2176	80%

Table 5-21. Electricity consumption of optimal night cooling and pure air conditioning

In order to discuss the potential of running the fan at a lower power or without the fan, two other cases are conducted and compared with the optimal results above. Night cooling period is set to be maximum (21:00 - 6:00) and other parameters are identical.

Comparison of total electricity consumption is shown in the chart below.

	1. Rated fan flow rate	**2. Half of rated fan flow rate**	**3. Zero fan flow rate**	**4. Closed all time**	**Optimal scheme**
May	335	202	476	1121	2
Jun	847	795	1161	1769	2
Jul	3379	3416	3673	3947	1
Aug	1800	2020	2697	3224	1
Sep	437	388	1087	2176	2

Table 5-22. Comparison of total electricity consumption for different fan speeds

Comparison of sensible cooling load (represented by chiller electricity in kWh) is shown in the chart below.

	1. Rated fan flow rate	**2. Half of rated fan flow rate**	**3. Zero fan flow rate**	**4. Closed all time**	**Optimal scheme**
May	27	143	476	1121	1
Jun	534	752	1161	1769	1
Jul	3015	3342	3673	3947	1
Aug	1450	1954	2697	3224	1
Sep	88	327	1087	2176	1

Table 5-23. Comparison of cooling load for different fan speeds

It is straightforward that maximum flow rate of fan will result in the lowest sensible cooling load. However, its total electricity consumption is close to that of case 2 and significantly lower than case 3 and 4. Lower sensible cooling load might be helpful to reduce the uncomfortable hours during daytime since the building does not have mechanical cooling system. It is suggested that maintain the rated working condition of the fan.

Several conclusions that can be made from this result are:

(1) Due to the relatively cool climate of Boston, night cooling can help reduce the energy required for the chiller. The longer the night cooling period, the lower the cooling load is. It can be seen that all minimum value of cooling load values are observed in the ’21:00 to 6:00’ case. Saving percentage ranges from 24% to 98%. It is suggested use longest operation of night cooling to minimized cooling load at daytime.

(2) When fan energy is taken into account, the optimal operation hour could be decided. Operation hour can be specified for each month. However, the results are close to longest operation hour because of the moderate power of roof exhaust fan.

(3) Below is a frequency distribution of the estimated beginning operation hour (Based on a sudden drop of temperature at any vent).

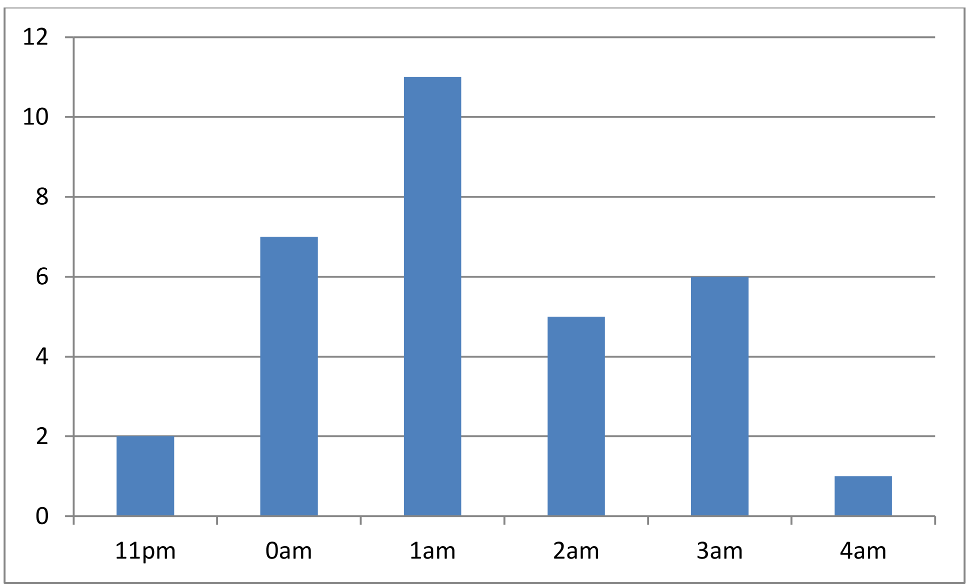


Figure 5-50. Monitored beginning time of night cooling

The night cooling is designed to begin at 4am [23]. But this time seems to be too late for the hottest period. The monitored operation hour begins mostly during 0am and 1am. It is suggested that this hour should be further extended, possibly to 11pm.

5.9.5 2F Simulation in CoolVent

Monitoring results between July 11st and July 14th are compared with simulation ones. The comparison shows good match between those two series of results: In most cases the difference is less than 2°C and trends of them are identical. Slab temperature is higher than air temperature, both in monitoring and simulating case. The figure below shows a comparison of office area in the second floor, the area of our major concern.

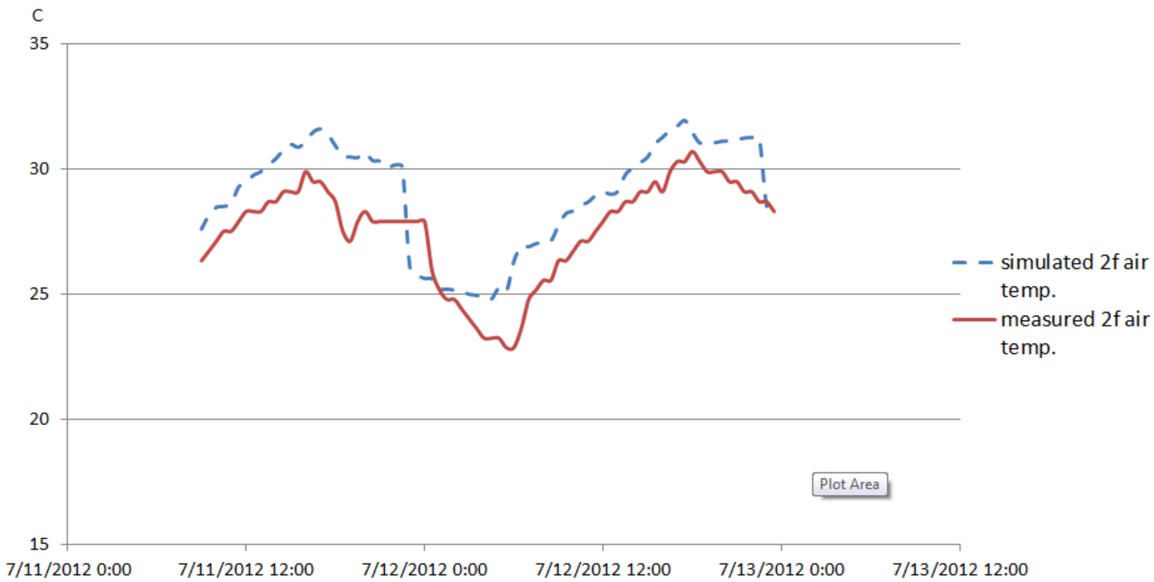


Figure 5-51. Comparison between simulated and measured 2F air temperature

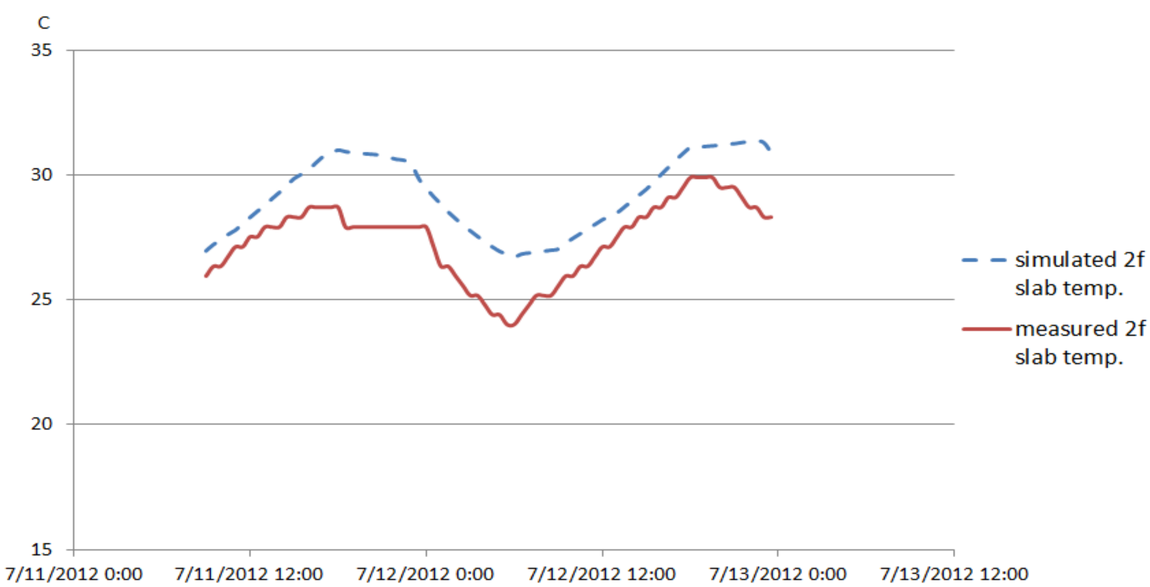


Figure 5-52. Comparison between simulated and measured 2F slab temperature

One important source of error is time period when the windows are open. Since not all opening time periods during the monitoring period is recorded, we only select a period of time when the openness of windows is certain. For this period of time, we have been to the building several times and observed all windows closed during daytime and ceiling fan used for creating indoor air flow when temperature got too high.

To evaluate the potential of opening window at daytime, simulation has been performed under two cases:

(1) Only fan assisted night cooling will be used. Night cooling period is set to be at maximum level. At daytime the building will be closed. This approximately corresponds to the current operating mode of the building.

(2) In addition to (1), fan assisted natural ventilation will also be used at daytime when outdoor temperature is lower than comfort limit value 28°C. This evaluates the potential of opening windows at daytime.

Comparison of comfort percentage under two cases is shown in the chart below.

Mode	**Overall comfort percentage (Office hour only)**
(1) Night cooling only, closed at day time Current Operating mode used	Jun - 79%; Jul - 44%; Aug - 55%
(2) NV with fan when appropriate	Jun - 90%; Jul - 70%; Aug - 79%

Table 5-24. Comparison of two operating modes in simulation

It shows that there is a significant improvement in performance if windows of the building could be opened at daytime when outdoor condition is suitable for natural ventilation.

6 REFERENCES

1. The Inside Story: A Guide to Indoor Air Quality. U.S. EPA/Office of Air and Radiation. Office of Radiation and Indoor Air (6609J) Cosponsored with the Consumer Product Safety Commission, EPA 402-K-93-007.

2. Buildings Energy Databook, 2006. US Department of Energy and Annual Energy Review 2007.DOE/EIA-0384 (2007).Energy Information Administration, U.S. Department of Energy. June 2008. http://www.eia.doe.gov/aer/pdf/aer.pdf.

3. Loftness, V. 2004. Improving Building Energy Efficiency in the US: Technologies and Policies for 2010 to 2050.Prepared for the PEW Center on Global Climate Change, http://www.house.gov/science/hearings/energyO4.

4. Yik, Francis WH, and Yu Fat Lun. "Energy saving by utilizing natural ventilation in public housing in Hong Kong." Indoor and built Environment 19.1 (2010): 73-87.

5. Cardinale, N., M. Micucci, and F. Ruggiero. "Analysis of energy saving using natural ventilation in a traditional Italian building." Energy and buildings 35.2 (2003): 153-159.

6. Wu, Li-Xia, Jia-Ning Zhao, and Zhao-Jun Wang. "Night ventilation and active cooling coupled operation for large supermarkets in cold climates." Energy and buildings 38.12 (2006): 1409-1416.

7. Yang, Lina, and Yuguo Li. "Cooling load reduction by using thermal mass and night ventilation." energy and buildings 40.11 (2008): 2052-2058.

8. Luo, Zhiwen, et al. "Estimating natural-ventilation potential considering both thermal comfort and IAQ issues." Building and environment 42.6 (2007): 2289-2298.

9. Schulze, Tobias, and Ursula Eicker. "Controlled natural ventilation for energy efficient buildings." Energy and Buildings (2012).

10. Ayata, Tahir, and Osman Yıldız. "Investigating the potential use of natural ventilation in new building designs in Turkey." Energy and buildings 38.8 (2006): 959-963.

11. Ji, Yingchun, Kevin J. Lomas, and Malcolm J. Cook. "Hybrid ventilation for low energy building design in south China." Building and Environment 44.11 (2009): 2245-2255.

12. Lomas, Kevin J., Malcolm J. Cook, and Dusan Fiala. "Low energy architecture for a severe US climate: design and evaluation of a hybrid ventilation strategy." Energy and Buildings 39.1 (2007): 32-44.

13. Olsen, Erik L., and Qinyan Yan Chen. "Energy consumption and comfort analysis for different low-energy cooling systems in a mild climate." Energy and buildings 35.6 (2003): 560-571.

14. Crawley, Drury B., et al. "Contrasting the capabilities of building energy performance simulation programs." Building and Environment 43.4 (2008): 661-673.

15. Pfafferott, Jens, Sebastian Herkel, and Matthias Wambsganß. "Design, monitoring and evaluation of a low energy office building with passive cooling by night ventilation." Energy and buildings 36.5 (2004): 455-465.

16. Cook, Malcolm, and Alan Short. "Natural ventilation and low energy cooling of large, non-domestic buildings–four case studies." International journal of ventilation 3.4 (2005): 283-294.

17. Wang, Zhaojun, et al. "Thermal comfort for naturally ventilated residential buildings in Harbin." Energy and Buildings 42.12 (2010): 2406-2415.

18. Tan, Gang. Study of natural ventilation design by integrating the multi-zone model with CFD simulation. Diss. Massachusetts Institute of Technology, 2005.

19. Yuan, Jinchao. Transition dynamics between the multiple steady states in natural ventilation systems: from theories to applications in optimal controls. Diss. Massachusetts Institute of Technology, 2007.

20. Menchaca Brandan, María Alejandra. Study of airflow and thermal stratification in naturally ventilated rooms. Diss. Massachusetts Institute of Technology, 2012.

21. Menchaca-B, M. A., and Leon Glicksman. "Coolvent: A multizone airflow and thermal analysis simulator for natural ventilation in buildings." Proceedings of SimBuild (2008): 1-8.

22. Torcellini, Paul, et al. "DOE commercial building benchmark models." Proceeding of 2008 ACEEE Summer Study on Energy Efficiency in Buildings, 2008.

23. Kollmuss Anja, Neely Dona and Kambli Shubhada. “EpiCenter headquarters for artists for humanity: demonstrating low-cost sustainable building strategies and integrated design process”.

24. Urban, Bryan James. The MIT Design Advisor: Simple and Rapid Energy Simulation of Early-Stage Building Designs. Diss. Massachusetts Institute of Technology, 2007.