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## **Influence of design parameters on the night ventilation performance in office buildings based on sensitivity analysis**

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# 1 **Influence of design parameters on the night ventilation** 2 **performance in office buildings based on sensitivity analysis**

## 3 **Abstract**

4 Overheating and energy-extensive consumption in buildings, especially in office buildings, are  
5 emerging challenges. Night ventilation (NV) is a promising technique. The performance of NV  
6 can be evaluated by a series of performance indicators. As many design parameters affect those  
7 indicators, it is beneficial to choose suitable indicators and identify the most important design  
8 parameters to develop more efficient design solutions at the early design stage. Sensitivity  
9 analysis makes it possible to identify the most important design parameters in relation to NV  
10 performance and to focus design and optimization of NV on these fewer, but most important,  
11 parameters. A holistic approach integrating sensitivity analysis and parametric simulation  
12 analysis is developed to explore the key design parameters on night cooling performance  
13 indicators and evaluate the applicability and limitations of those indicators. The results show  
14 that the climatic conditions and NV modes strongly affect the influence of design parameters  
15 on the performance indicators. The window-wall ratio, internal thermal mass level, internal  
16 convective heat transfer coefficient, and night mechanical air change rate are the most important  
17 design parameters. The indicators of ventilative cooling advantage, cooling requirement  
18 reduction, and percentage outside the range are recommended for the night cooling performance  
19 evaluation.

## 20 **Keywords**

21 Night ventilation; Performance indicators; Design parameters; Sensitivity analysis; Parametric  
22 simulation

**Nomenclature**

$T_{out}$	outlet air temperature (°C)
$T_{in}$	inlet air temperature (°C)
$\bar{T}_{surface}$	average building indoor surface temperature (°C)
$T_{o,max}$	maximum ambient air temperature (°C)
$T_{o,min}$	minimum ambient air temperature (°C)
$T_{i,max}$	maximum building indoor air temperature (°C)
$T_{i,min}$	minimum building indoor air temperature (°C)
$T_i(t)$	building indoor air temperature at time t (°C)
$T_o(t)$	ambient air temperature at time t (°C)
$\dot{m}_{air}$	airflow rate (kg/s)
$c_p$	specific heat capacity (kJ/kg.°C)
$P_e$	electric power of fan (W)
$t_i$	start time of night-time ventilation (h)
$t_f$	end time of night-time ventilation (h)
$Q_{el,c}^{ref}$	cooling system electrical energy consumption of the scenario without ventilative cooling (kWh/m <sup>2</sup> )
$Q_{el,c}^{scen}$	cooling system electrical energy consumption of the scenario with ventilative cooling (kWh/m <sup>2</sup> )
$Q_{el,v}$	electrical energy use of the night ventilation system
$Q_{t,c}^{ref}$	cooling demand of the reference scenario (kWh)
$Q_{t,c}^{scen}$	cooling demand of the analyzed scenario (kWh)
$wf_i$	weighting factor
$h_i$	occupied hours (h)
$T_{conf,sup}$	upper comfort temperature limit (°C)

**Abbreviations**

NV	Night ventilation
TE	Temperature efficiency
TDR	Temperature difference ratio
DF	Decrement factor
COP	Coefficient of performance
ADV	Ventilative cooling advantage
CRR	Cooling requirements reduction
POR	Percentage outside the range
DhC	Degree-hours criterion
DI	Weighted discomfort temperature index
SHGC	Solar heat gain coefficient
CHTC	Convective heat transfer coefficient
MCA	Monte Carlo analysis
LHS	Latin hypercube sampling
SRC	Standardized regression coefficient
SA	Sensitivity analysis
ACH	Air change rate per hour
WWR	Window-wall ratio
AC	Air conditioner

## 24 **1 Introduction**

25 During the last decades, there has been a trend of increasing cooling demand in buildings. This  
26 has especially been the case for commercial buildings, where high internal loads in combination  
27 with high solar gains through extensive glazing have led to considerable cooling loads, even in  
28 moderate and cold climates [1]. An additional rise of the cooling demand is caused by global  
29 climate warming, which is expected to increase summertime temperatures significantly [2][3].  
30 Night ventilation is a promising way to alleviate or solve the foregoing problem. The basic  
31 concept is to utilize the relatively low-temperature ambient air during the night time by the  
32 natural or mechanical ventilation systems to cool down the indoor air as well as the building  
33 construction components to provide a heat sink for the following day [4][5].  
34 Numerous night cooling projects have been successfully undertaken in the past decades [6–10].  
35 Despite the simplicity of the concept, architects and engineers are hesitant to apply this low-  
36 energy technology [11]. One reason is that the efficiency of night-time cooling is affected by  
37 many parameters, which makes the performance predictions uncertain. Another reason is that  
38 there are many different performance indicators used for night ventilation design and evaluation,  
39 which confuse designers. Some of these indicators focus on temperature performance, others  
40 evaluate the energy balance, and several of them pay attention to thermal comfort. The heat  
41 removal effectiveness of night ventilation is evaluated by the temperature performance of the  
42 building and its relationship to the outdoor temperature profile. Several researchers have  
43 proposed different indicators for heat removal, including ventilation effectiveness for heat  
44 removal [12], temperature efficiency [13], temperature difference ratio [14], decrement factor,  
45 and daily time lag [15]. The energy efficiency of night ventilation is evaluated by the ratio of  
46 ventilation energy saving and ventilation equipment energy use. The indicators for energy  
47 efficiency proposed by researchers are the coefficient of performance [16], potential energy  
48 efficiency index [17], ventilative cooling advantage, cooling requirement reduction [18], etc.

49 For thermal comfort evaluation when applying night ventilation, there are indicators like the  
50 degree-hours criterion[1] and the weighted discomfort temperature index[19]. Some of the  
51 indicators are independent of each other, others have a different level of dependency between  
52 each other. It is necessary to choose multiple indicators to have an overall evaluation of the  
53 night ventilation performance.

54 Sensitivity analysis is a useful tool to identify the most important parameters for the building  
55 design and energy analysis [20]. The methods for sensitivity analysis can be sorted into local  
56 sensitivity methods and global sensitivity methods [21]. Local sensitivity analysis is based on  
57 only varying one design parameter at a time, while the global sensitivity analysis is based on  
58 changing all the design parameters at the same time [22]. Therefore, the global method is more  
59 reliable but with a high computational calculation effort compared to the local method. Both  
60 local [23–26] and global methods [27–31] have been widely used in investigating the most  
61 important variables related to building energy performance. Among those, few research are  
62 about night ventilation performance. Artmann et al [1] conducted a local sensitivity analysis to  
63 investigate the most influential design parameters for night mechanical ventilation in an office  
64 room located in a moderate climatic location with the indicator of the number of overheat degree  
65 hours. The conclusion was that the climatic conditions and air flow rate at night-time were the  
66 most important parameters. Finn et al [32] examined the design and operational parameters in  
67 a night ventilated library building located in a maritime type climate. The result showed the  
68 building mass as the most significant parameter, followed by the internal heat gains and night  
69 air flow rates. Breesch and Janssens [29][33] analyzed the input parameters causing the  
70 uncertainty on the thermal comfort for a single-sided night natural ventilation in the moderate  
71 climate. The results showed that the top 3 important design parameters were the internal heat  
72 gains, the solar heat gain coefficient of the sun blinds, and the internal convective heat transfer  
73 coefficient. Encinas et al [34] found that for night cooling of a real estate market in a warm  
74 climate region, the most important input parameter for summer comfort is solar and light

75 transmittance of the solar protection devices, followed by the night ventilation flow rate.  
76 Goethals et al [30] investigated the sensitivity of convection algorithms on the night ventilation  
77 performance, showing that the selection of the convection algorithm strongly affects the energy  
78 and thermal comfort predictions. Ran et al [35] adopted the local sensitivity analysis method to  
79 investigate the influence of external wall insulation level, night ventilation airflow rate on the  
80 indoor air temperature reduction, showing that the increase of the insulation level and night  
81 airflow rate will enhance the night cooling performance.

82 The aforementioned sensitivity analyses for night ventilation performance are mostly only  
83 focused on one night ventilation mode with one daytime cooling method or limited to the  
84 amount of performance indicators and climate regions. To get an overall design guideline of  
85 night ventilation design parameters, research should include various night ventilation systems  
86 and performance indicators in different climatic conditions.

87 This paper firstly selects nine performance indicators for night ventilation performance  
88 evaluation. Then it investigates the performance of night mechanical and natural ventilation  
89 integrated with three different daytime cooling systems (air conditioning, mechanical  
90 ventilation, and natural ventilation) to do a global sensitivity analysis for an office room located  
91 in three climate zones (cold, medium, and hot climate regions). The night cooling performance  
92 is analyzed based on the parametric simulation results in consideration of the thermal comfort  
93 evaluation and energy-saving benefit. Finally, the evaluation of the applicability of performance  
94 indicators is conducted to propose the recommendation.

## 95 **2 Methodology**

### 96 **2.1 Outline of the quantitative study**

97 A systematic approach is proposed to evaluate and quantify the influence of different design  
98 parameters on the night ventilation performance alongside the evaluation of performance  
99 indicators as shown in Fig.1. *In the first step*, a suitable series of performance indicators for

100 night cooling are reviewed and selected. *In the second step*, a software designed for uncertainty  
 101 and sensitivity analysis by Monte Carlo method-SimLab v2.2 [36] generates samples based on  
 102 the input design parameters and sends the scenarios to the parametric simulation manger jEPlus  
 103 [37]. Then, the jEplus uses the model built by EnergyPlus to do parametric simulations before  
 104 transferring the simulation results back again to SimLab. Follow on, a global sensitivity analysis  
 105 is conducted in SimLab by regression method to investigate the influence of design parameters  
 106 on performance indicators. Finally, the parametric simulation results of night cooling  
 107 performance indicators are used to propose the application recommendations for those  
 108 performance indicators by mathematical analysis.

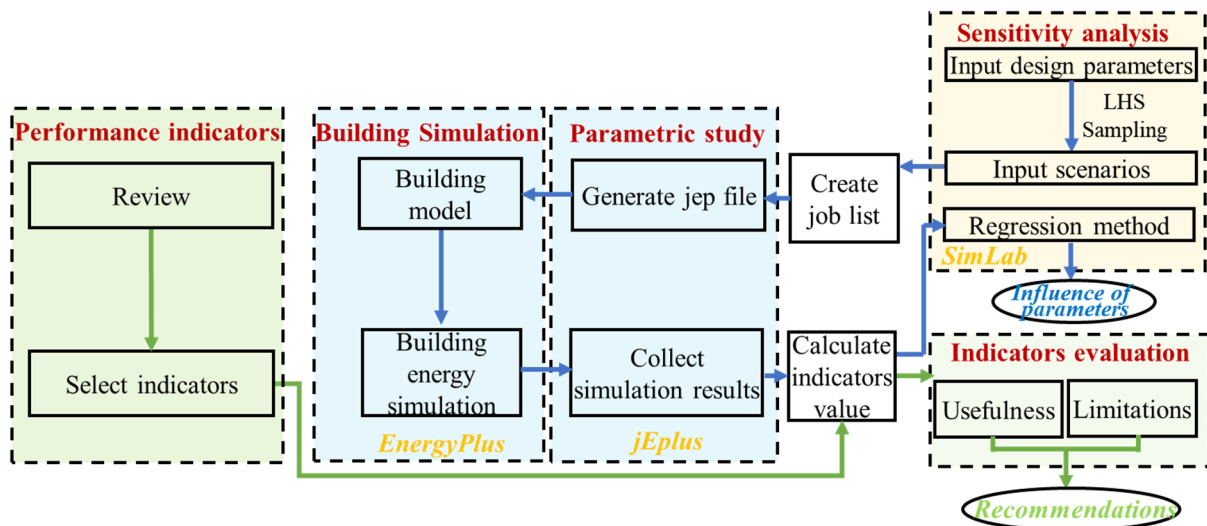


Fig.1. Flow chart of the systematic approach.

## 2.2 Performance indicators of night ventilation

112 Appropriate performance indicators should be chosen according to the application conditions  
 113 of the night ventilation, in order to provide guidelines for the measurement or simulation in the  
 114 design process to achieve those goals. It should be noted that the performance of night  
 115 ventilation cannot be well represented by a single indicator. It needs a combination of different  
 116 types of indicators. The performance of night ventilation can be quantified by the  
 117 thermodynamical effect (energy balance) and by its cooling effect (room temperature). Night  
 118 ventilation performance indicators can be sorted into the following four categories: 1) Heat

119 removal effectiveness, 2) Energy efficiency, 3) Ability to reduce cooling energy use, and 4)  
 120 Thermal comfort improvement [38]. Heat removal effectiveness quantifies the ability of the  
 121 night cooling system to remove excess heat stored in the building. Energy efficiency quantifies  
 122 the energy use required to reduce cooling demand. The ability to reduce cooling energy use  
 123 represents the ability of the night cooling system to provide energy saving for the daytime  
 124 mechanical cooling. Thermal comfort improvement shows the ability of the night cooling  
 125 system to reduce periods of thermal discomfort during the occupied time.

126 Some indicators are more suitable for simulation analysis because they can be easily calculated  
 127 by post-processing outcomes of building energy simulation runs of a reference scenario (e.g.  
 128 mechanically cooled building) and a ventilative cooling scenario (e.g. natural night cooling and  
 129 daytime mechanical cooling). However, other indicators are more suitable for experimental  
 130 analysis, since some data is easier to obtain in field studies. In addition, in experimental studies,  
 131 the thermal comfort improvement indicators are much more prevalent than the energy efficiency  
 132 indicators, probably because the indoor conditions are easier obtained than energy data, which  
 133 is often challenging to measure directly.

134 In this paper, we select nine performance indicators in total from the four categories mentioned  
 135 above to evaluate the influences of different design parameters. Table 1 summarizes the selected  
 136 performance indicators.

137 Table 1: Summary of the selected performance indicators.

Family of indices	Indicator name	Expression	Explanation	Source
Heat removal effectiveness	Temperature efficiency (TE)	$TE = \frac{T_{out} - T_{in}}{\bar{T}_{surface} - T_{in}}$	Originates from experimental studies. Mainly depends on the air distribution concept and the airflow rate. For mixing ventilation, the value of temperature efficiency is limited to 1, while in displacement ventilation the temperature stratification can result in an efficiency exceeding 1.	[13]
	Temperature difference ratio (TDR)	$TDR = \frac{T_{o,max} - T_{i,max}}{T_{o,max} - T_{o,min}}$	Used with good results to compare passive cooling systems with different configurations. A higher value of TDR indicates a larger temperature difference between indoors and outdoors and thus a more efficient night cooling strategy.	[14]
	Decrement factor (DF)	$DF = \frac{T_{i,max} - T_{i,min}}{T_{o,max} - T_{o,min}}$	Means the ratio of indoor air temperature fluctuation to the ambient air temperature fluctuation.	[15]
Energy efficiency	Coefficient of performance (COP)	$COP = \frac{\int_{t_s}^{t_e} \dot{m}_{air} c_p (T_i(t) - T_o(t)) dt}{\int_{t_s}^{t_e} P_e(t) dt}$	The ratio of the cooling energy delivered into the building to the auxiliary electric consumption by mechanical machines during the night period. The higher the COP, the better the performance for night-time ventilation.	[16]



Ventilative cooling advantage (ADV)		$ADV_{VC} = \frac{Q_{el,c}^{ref} - Q_{el,c}^{scen}}{Q_{el,v}}$	Defines the benefit of the night ventilative cooling in case which ventilation rates are provided mechanically. If $ADV_{VC}$ is lower than 1, the electrical energy use of the scenario is higher than the reference scenario. If $ADV_{VC}$ is higher than 1, the electrical energy use of the scenario is lower than the reference scenario.	[18]
Ability to reduce cooling energy use	Cooling requirements reduction (CRR)	$CRR = \frac{Q_{t,c}^{ref} - Q_{t,c}^{scen}}{Q_{t,c}^{ref}}$	Expresses the percentage of reduction of the cooling demand of a scenario with night cooling in respect to the cooling demand of the reference scenario. The value of CRR can range between -1 and +1. If CRR is positive, it means that the night ventilative cooling system reduces the cooling need of the building. If the value of CRR is negative or 0, it means that the night ventilative cooling system does not reduce the cooling requirements	[18]
Thermal comfort improvement in daytime	Percentage outside the range (POR)	$POR = \frac{\sum_{t=1}^{oh}(w_{fi} \cdot h_i)}{\sum_{t=1}^{oh}(h_i)}$	Accumulate the percentage of occupied hours when the thermal comfort parameters are outside a specified range. The comfort range can be expressed in terms of PMV when referring to the Fanger model or in terms of operative temperature when referring to the adaptive comfort model. If the thermal comfort parameters exceed the corresponding comfort range, the $w_{fi}$ would be 1, or the $w_{fi}$ would be 0. The lower value of POR is, the better thermal comfort improvement is provided by night ventilative cooling.	[39]
	Degree-hours criterion (DhC)	$DhC = \sum_{t=1}^{oh}(w_{fi} \cdot h_i)$	Accumulate overheating degree hours of the operative room temperature above 26°C during the occupied period. $w_{fi}$ here is calculated as the module of the difference between actual and calculated operative temperature. The lower the value of DhC is, the better the thermal comfort improvement is provided by night ventilation.	[1]
	Weighted discomfort temperature index (DI)	$DI = \sum(T_i - T_{comf,sup})$	Discomfort weighted on the distance of calculated operative temperature from the comfort temperature upper limit which is fixed at 28 °C. The lower the value of DI is, the better thermal comfort improvement is provided by night ventilation.	[19]

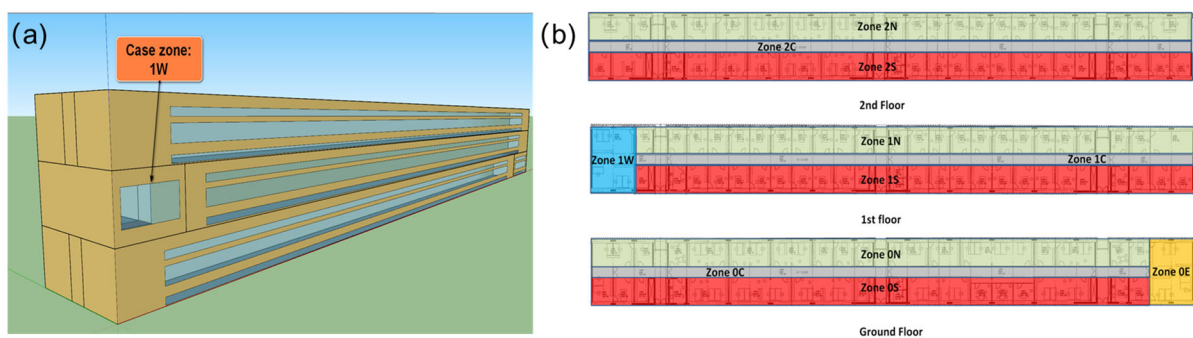
138

## 139 2.3 Case study

### 140 2.3.1 Building model

141 The EnergyPlus v.8.9 software was selected in this study to build a model and simulate its heat,  
142 energy, and thermal comfort performance. An office building located in Aarhus, Denmark was  
143 used for this study, as shown in Fig. 2 (a). The building is 103.7 m long and 9.5 m wide, with 3  
144 stories and a total area of 2924.1 m<sup>2</sup>. The layout of the office building can be seen in Fig. 2 (b),  
145 in which N, W, S, and C indicate the orientation as north, west, south, and center respectively.  
146 A typical office room 1W occupied by 6 persons was selected as the case zone, whose floor  
147 area is 51.3 m<sup>2</sup> and height is 2.8 m [40]. Internal partitions between the concerned zone 1W and  
148 adjacent zones were set as adiabatic to assume the similar conditions in all adjacent zones. The  
149 case was simulated in the hot (Rome), medium (Geneva), and cold (Copenhagen) climates

150 respectively to investigate the climate influence on night ventilation performance. The weather  
 151 data for the three locations originated from the World Meteorological Organization [41].  
 152 In order to evaluate the influence of building orientation on night ventilation performance, the  
 153 orientation was set with a uniform distribution from  $0^\circ$  to  $360^\circ$ . The European ventilation  
 154 standard for office building recommends that the airtightness should be below  $1.0 \text{ h}^{-1}$  in case of  
 155 buildings with more than three stories [42]. The infiltration of building airtightness was set with  
 156 triangular distribution with a minimum value of  $0.1 \text{ h}^{-1}$ , maximum  $1.0 \text{ h}^{-1}$ , and mean value  $0.6$   
 157  $\text{h}^{-1}$ .



158  
 159 Fig. 2. (a) View of the building model and (b) Layout of the case office building.  
 160

### 161 2.3.2 Thermal mass models

162 Thermal mass can be sorted as external and internal thermal mass. External thermal mass, such  
 163 as an external wall or roof, is affected by the ambient air temperature and solar radiation  
 164 directly. Internal thermal mass, such as internal walls or interior furniture, influences the indoor  
 165 air temperature through the process of absorbing and releasing heat [43]. For the concerned  
 166 zone 1W, the external thermal mass is the external wall, while the internal thermal mass  
 167 contains an internal wall, ceiling, floor, and interior furniture.

168 Three different levels (light, medium, heavy) were defined for external and internal thermal  
 169 mass, respectively. Table 2 shows the detailed composition of the thermophysical properties of  
 170 building materials and the thermal mass of the building components. The last column of Table  
 171 2 is the dynamic heat capacity per unit floor area, indicating the thermal mass level. The  
 172 dynamic heat capacity  $c_{dyn}$  is the ability to store energy per area when the building component

173 is exposed to a sinusoidal temperature variation for a period of 24 h with surface resistance, as  
 174 defined by EN ISO 13786 [44]. It should be noticed that for light, medium, and heavy internal  
 175 thermal mass levels, the interior furniture surface area is 10, 30, 50 m<sup>2</sup> respectively.

176 Table 2: Detailed composition of the thermal mass and thermophysical properties of building materials.

External thermal mass						
	$d$ (mm)	$\rho$ (kg/m <sup>3</sup> )	$c$ (J/kg/K)	$\lambda$ (W/m·K)	Total $c_{dyn}/A_{floor}$ (kJ/m <sup>2</sup> ·K)	
<b>External wall (Heavy)</b>						
Plaster	15	1400	936	0.7		
Sand-lime	150	2000	936	1.1		
Exp.polystyrene	120	40	1200	0.035		
Plaster ext.	20	1600	1000	0.87		77.5
<b>External wall (Medium)</b>						
Plasterboard (fire-resisting)	160	900	1000	0.25		
Concrete 200	200	2385	800	1.2		
PUR 210	210	40	1400	0.021		
Cement plate	15	2000	1500	0.35		42.0
<b>External wall (Light)</b>						
Gypsum board	25	1000	792	0.4		
Exp.polystyrene	120	40	1200	0.035		
Concrete 180	180	2400	1080	1.8		24.0
Internal thermal mass						
	$d$ (mm)	$\rho$ (kg/m <sup>3</sup> )	$c$ (J/kg/K)	$\lambda$ (W/m·K)	$R_{userdefined}$ (m <sup>2</sup> ·K/W)	Total $c_{dyn}/A_{floor}$ (kJ/m <sup>2</sup> ·K)
<b>Internal wall (heavy)</b>						
Plaster	15	1400	936	0.7		
Sand-lime	150	2000	936	1.1		
Plaster	15	1400	936	0.7		
<b>Ceiling (Heavy)</b>						
Concrete 180	180	2400	1080	1.8		
<b>Floor (Heavy)</b>						
Concrete 180	180	2400	1080	1.8		
Sound insulation	40	30	1404	0.04		
Plaster floor	80	2200	1080	1.5		
Carpet	5	80	930	0.05		
<b>Interior furniture (Heavy)</b>						
Wood 6inch	150	540	1210	0.12		238.1
<b>Internal wall (Medium)</b>						
Gypsum board	25	1000	792	0.4		
Mineral wool	70	1750	1000	0.56		
Gypsum board	25	1000	792	0.4		
<b>Ceiling (Medium)</b>						
Cast concrete	120	1800	1000	1.13		
<b>Floor (Medium)</b>						
Linoleum	3	1200	1470	0.17		

177

Cement screed (fiber reinforced)	50	1400	1000	0.8	
Acoustic insulation	9	556	1700	0.15	
OSB panels	25	600	2150	0.13	
Insulation glass wool	200	28	1030	0.032	
Wooden panels	60	250	2100	0.047	
<b>Interior furniture (Medium)</b>					
Wood 6inch	150	540	1210	0.12	160.1
<b>Internal wall (Light)</b>					
Gypsum board	25	1000	792	0.4	
Mineral wool	70	90	612	0.036	
Gypsum board	25	1000	792	0.4	
<b>Suspend ceiling (Light)</b>					
Acoustic panel	20	800	900	0.21	
Air gap	250				0.16
<b>Floor (Light)</b>					
Linoleum	3	1200	1470	0.17	
Acoustic insulation	9	556	1700	0.15	
OSB panels	25	600	2150	0.13	
Insulation glass wool	200	28	1030	0.032	
Wooden panels	60	250	2100	0.047	
<b>Interior furniture (Light)</b>					
Wood 6inch	150	540	1210	0.12	63.3

178

### 179 2.3.3 Internal heat gain models

180 Similar to the thermal mass, internal heat gains were also defined by three different levels, cf.  
 181 Table 3. The hourly operational schedules for people, lights, and electric equipment were  
 182 always 1.0 during the occupied hours (08:00-17:00) and 0 for the other hours. The people  
 183 clothing insulation was set to 0.5 clo in summer [45].

184

Table 3: Internal heat gains per unit floor area in zone 1W.

Internal heat gains		Low	Medium	High
People	W/pers.	70	75	80
Lights	W/m <sup>2</sup>	4	6	8
Electric equipment	W/m <sup>2</sup>	6	8	10
Total	W/m <sup>2</sup>	18.2	22.8	27.4

185

### 186 2.3.4 Window models

187 The windows in zone 1W were modeled as energy-efficient windows with a double pane  
 188 construction made by 3mm glass and a 13mm gap filled with argon. The window U-value is  
 189 1.062 W/m<sup>2</sup>·K, while the glass solar heat gain coefficient (SHGC) and visible transmittance are  
 190 0.579 and 0.698 respectively. In order to evaluate the influence of window-wall ratio on night

191 ventilation performance, the design parameter of the window-wall ratio for north and south  
192 windows of zone 1W was set with a discrete distribution from 10%, 20%,...,90%.

### 193 **2.3.5 Night ventilation systems**

194 Two typical concepts of night ventilation were selected for the investigation, which are  
195 mechanical ventilation and natural ventilation. The night venting schedule is during 17:00-  
196 08:00 (+1) from 1<sup>st</sup> July to 1<sup>st</sup> September, except for weekends.

197 The night mechanical ventilation system is a balanced system with a supply fan and an exhaust  
198 fan. The night natural ventilation has been modeled using a wind and stack model in EnergyPlus,  
199 in which the ventilation air flow rate is a function of wind speed and thermal stack effect, along  
200 with the area of the opening being modeled [46].

201 To prevent the overcooling and to store more cooling energy in building thermal mass, the  
202 minimum indoor air temperature setpoint for both night ventilation systems was 18°C [47].

203 Night ventilation is only activated when the indoor air temperature exceeds the ambient  
204 temperature at a certain temperature which was set with a discrete value of 1, 2, and 3 °C.

205 Because the maximum airflow rate for the design of night ventilation should be further  
206 increased corresponding to an air change rate of 10 h<sup>-1</sup>[18], the design air flow rate for night  
207 mechanical ventilation was set with uniform distribution from 1 to 10 h<sup>-1</sup>. For night natural  
208 ventilation, the opening area is 0.4 m<sup>2</sup>, and the discharge coefficient of the opening was set with  
209 a typical uniform distribution from 0.5 to 0.7 [48]. The opening effectiveness for natural  
210 ventilation was calculated automatically in EnergyPlus so that the window can be assumed to  
211 adjust its angle to make the most use of wind under different wind direction. Table 4 shows the  
212 detailed setup information of night ventilation.

213 **Table 4: Detailed setup information of night ventilation systems.**

---

<i>Night mechanical ventilation</i>	
System	Supply fan + exhaust fan
Design pressure rise	600 Pa (Both for supply and exhaust fan)
Fan total efficiency	0.9
Design flow rate	U[1-10]
Minimum indoor temperature	18°C

---

---

Activation requirements	$T_{in}-T_{out}> \mathbf{D}[1, 2, 3] \text{ } ^\circ\text{C}$
<i>Night natural ventilation</i>	
System	Natural ventilation driven by wind and stack effect
Minimum indoor temperature	18°C
Activation requirements	$T_{in}-T_{out}> \mathbf{D}[1, 2, 3] \text{ } ^\circ\text{C}$
Opening area	0.4 m <sup>2</sup>
Discharge coefficient	$\mathbf{U}[0.5-0.7]$
Opening effectiveness	Automatic calculation by EnergyPlus

---

$T_{in}$ : indoor air temperature (°C);  $T_{out}$ : ambient temperature (°C); D: discrete distribution (levels); U: uniform distribution (lower value, upper value);

214  
215  
216

### 217 **2.3.6 Daytime cooling systems**

218 Three typical methods were selected to cool the building at daytime, which are air conditioner  
219 (AC), mechanical ventilation, and natural ventilation. The operating period for daytime cooling  
220 is 08:00-17:00 on weekdays from 1<sup>st</sup> July to 1<sup>st</sup> September.

221 A packaged thermal heat pump with a dedicated outdoor air system was modeled as the air  
222 conditioning system with COP (coefficient of performance) 3.0 for cooling in summer with the  
223 HVAC template module of EnergyPlus. The setpoint for the air conditioning system is 24.5°C  
224 which is a middle point of the temperature range for cooling EN 15251 [45]. The outdoor air  
225 flow rate was set to 30 m<sup>3</sup>/h per person [45].

226 The setups for daytime mechanical ventilation and natural ventilation are similar to that of night  
227 mechanical and natural ventilation systems respectively, but with some differences. The first  
228 difference is the design flow rate for daytime mechanical ventilation and maximum flow rate  
229 for daytime natural ventilation is 6 h<sup>-1</sup>. It is because the typical maximum air flow rate used in  
230 the design of daytime ventilative cooling is 6 h<sup>-1</sup> [18]. The second difference is that when the  
231 indoor and outdoor air temperature difference is smaller than 2°C, the outdoor air flow rate is  
232 30 m<sup>3</sup>/h per person to fulfill the human hygiene requirements. Table 5 shows the detailed setup  
233 information of daytime cooling methods.

234 **Table 5: Detailed setup information about daytime cooling methods.**

---

<i>Daytime air conditioning</i>	
System	Packaged terminal heat pump + dedicated outdoor air system
Setpoint	24.5°C
Design fan pressure rise	75 Pa

---

Outdoor air flow rate	30 m <sup>3</sup> /h/person
<i>Daytime mechanical ventilation</i>	
System	Supply fan + exhaust fan
Design fan pressure rise	1000 Pa (Both for supply and exhaust fan)
Fan total efficiency	0.9
Minimum indoor temperature	24.5°C
Design flow rate	6 h <sup>-1</sup> or 30 m <sup>3</sup> /h/person
Control strategy	If T <sub>in</sub> -T <sub>out</sub> >2°C air flow=6 h <sup>-1</sup> or flow=30 m <sup>3</sup> /h /person
<i>Daytime natural ventilation</i>	
System	Natural ventilation driven by wind and stack effect
Minimum indoor temperature	24.5°C
Opening area	0.4 m <sup>2</sup>
Discharge coefficient	U[0.5- 0.7]
Control strategy	If T <sub>in</sub> -T <sub>out</sub> >2°C air flow<6 ACH or flow=30 m <sup>3</sup> /h /person
Opening effectiveness	Automatic calculation by EnergyPlus

235

### 236 2.3.7 Internal convective heat transfer coefficient

237 Several research indicated different convective heat transfer coefficient (CHTC ) correlations  
 238 or values for different types of the internal surface [49][50]. According to the EN ISO 13791  
 239 [51], the standard convective heat transfer coefficient for vertical, horizontal (upward), and  
 240 horizontal(downward) are 2.5, 5.0, 0.7 W/m<sup>2</sup>·K respectively. As a consequence, the CHTC of  
 241 internal surfaces were both set with uniform distribution from 0.5 to 5 W/m<sup>2</sup>·K.

242

### 243 2.3.8 Summary of the independent design parameters

244 Table 6 summarizes the independent design parameters for night mechanical/natural ventilation.  
 245 P6 has two meanings, of which night air change rate per hour (ACH) is for mechanical  
 246 ventilation and discharge coefficient for the opening of natural ventilation.

247

Table 6: Design parameters for sensitivity analysis, their range, and distribution.

Parameter	Unit	Distribution
P1 External thermal mass	kJ/m <sup>2</sup> ·K	D[24.0, 42.0, 77.5]
P2 Internal thermal mass	kJ/m <sup>2</sup> ·K	D[63.3, 160.1, 238.1]
P3 Internal heat gains	W/m <sup>2</sup>	D[18.2, 22.8, 27.4]
P4 Window-wall ratio (WWR)	%	D[10, 20, 30, 40, 50, 60, 70, 80, 90]
P5 Internal CHTC	W/m <sup>2</sup> ·K	U[0.7-5]
P6 Night ACH	h <sup>-1</sup>	U[1-10]
Discharge coefficient for opening	-	U[0.5-0.7]
P7 Building airtightness	h <sup>-1</sup>	T[0.1, 0.6, 1]
P8 Building orientation	°	U[0-360]

## 250 **2.4 Sensitivity analysis**

251 Sensitivity analysis (SA) can be divided into three different types: screening methods, local  
252 sensitivity methods, and global sensitivity methods [52]. In this paper, the global sensitivity  
253 analysis methods were selected to quantify the influence of a single input variable on the outputs  
254 while all other input variables also vary simultaneously. Monte Carlo Analysis (MCA) is the  
255 most prevalent variance-based method because it provides approximate solutions only with a  
256 restricted number of simulations and the input variables have uncertainties of a different order  
257 of magnitude [29]. Different sampling methods exist in MCA studies: random sampling,  
258 importance sampling, quasi-random sampling, and Latin hypercube sampling (LHS). The LHS  
259 method was selected because this method is a powerful tool in building performance analysis  
260 and it fully covers the range of each variable [20]. The sample size based on LHS was chosen  
261 to be 400 as the minimum number of model executions should be higher than 10 times the  
262 number of variables [53]. SimLab v2.2 generated the 400 samples by LHS method [53], then  
263 those samples were sent to jEPlus to do parametric simulations before transferring the  
264 simulation results back again to SimLab to do the sensitivity analysis. The Standardized  
265 Regression Coefficient (SRC) based on regression analysis was used as the global sensitivity  
266 analysis indicator when the input variables are independent. The sign of SRC indicates whether  
267 the output increases (positive value) or decreases (negative value) with the related input variable  
268 increases. The bigger the absolute value of SRC, the more influential the input variable is.  
269 Calculating the SRCs involves a linear multidimensional model based on an  $m \times k$  samples,  
270 with  $m$  the total number of samples and  $k$  the total number of input variables:

$$271 \hat{y}_i = \beta_0 + \sum_{j=1}^k \beta_j x_j \quad (1)$$



272 where  $\hat{y}_i$  represents the estimate of the output  $y_i$ ,  $x_j$  the input variable,  $i$  is the sample size,  $j$  is  
 273 the number of variables and  $\beta_j$  the regression coefficient. This regression model can be  
 274 standardized by subtracting the mean value from each input and output factor and successively  
 275 dividing this result by its standard deviation:

$$276 \quad \frac{\hat{y}_i - \bar{y}}{\hat{\sigma}} = \sum_{j=1}^k \frac{\beta_j \hat{\sigma}_j}{\hat{\sigma}} \frac{(x_j - \bar{x}_j)}{\hat{\sigma}_j}$$

277 where

$$278 \quad \bar{y} = \sum_{i=1}^m \frac{y_i}{m}, \bar{x}_j = \sum_{i=1}^m \frac{x_{ij}}{m}, \hat{\sigma} = \sqrt{\left[ \sum_{i=1}^m \frac{(y_i - \bar{y})^2}{m-1} \right]}, \hat{\sigma}_j = \sqrt{\left[ \sum_{i=1}^m \frac{(x_{ij} - \bar{x}_j)^2}{m-1} \right]} \quad (2)$$

279 The SRC for the input variable  $j$  is defined as:

$$280 \quad SRC_j = \frac{\beta_j \hat{\sigma}_j}{\hat{\sigma}} \quad (3)$$

281 The model coefficient of determination  $R_y^2$  measures how well the linear regression model  
 282 matches the data, which can be calculated by:

$$283 \quad R_y^2 = \frac{\sum_{i=1}^m (\hat{y}_i - \bar{y})^2}{\sum_{i=1}^m (y_i - \bar{y})^2} \quad (4)$$

284 where  $R_y^2$  represents the fraction of the variance of the output explained by the regression. The  
 285 closer it is to 1, the better the model performance is.

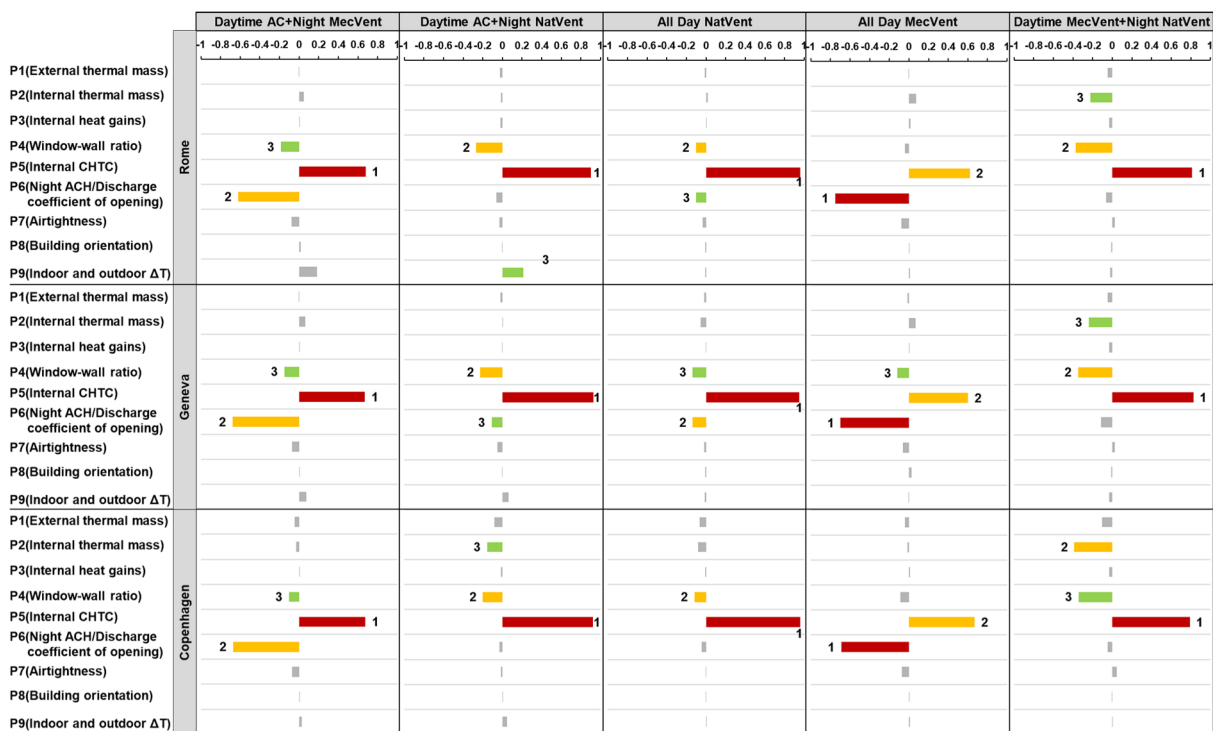
## 286 **3 Results**

### 287 **3.1 SA for temperature efficiency (TE)**

288 Fig. 3 illustrates the results of the sensitivity analysis ( $R^2=0.95$ ) for TE where the three top (and  
 289 the absolute value of SRC greater than 0.1) influential parameters are labeled. It can be  
 290 concluded that the internal CHTC is the most influential parameter for all climates and systems,  
 291 except for the all-day mechanical ventilation system, but still ranking second. P6 (Night ACH)  
 292 is important for the systems with night mechanical ventilation, while P6 (Discharge coefficient  
 293 of opening) is not obvious in cases with night natural ventilation. The risk of this happening for

294 the range of discharge coefficient is relatively small and will not influence the level of night  
 295 natural ventilation rate. However, it is acceptable because the range has been defined according  
 296 to the bibliography. Increasing window-wall ratio (WWR) always decreases the value of this  
 297 indicator considerably, except for the all-day mechanical ventilation system. In the daytime  
 298 mechanical ventilation with night natural ventilation system, the internal thermal mass becomes  
 299 more influential. Additionally, the colder the weather is, the larger the influence of the internal  
 300 thermal mass on TE.

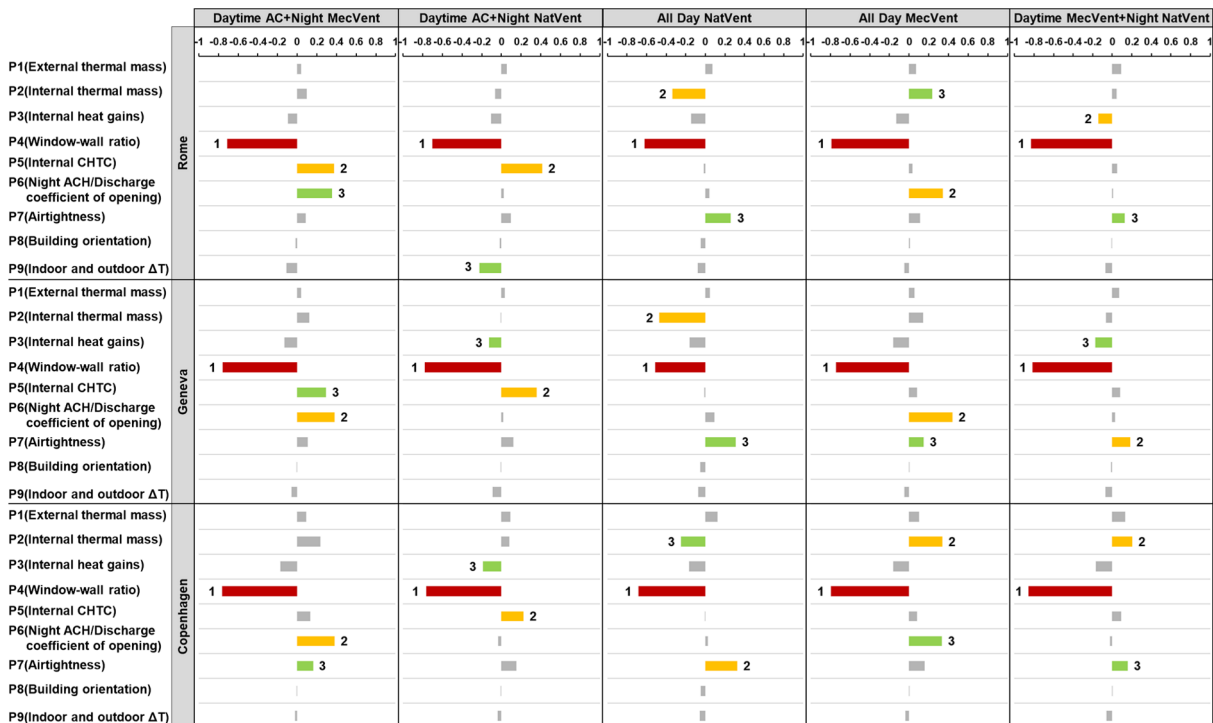
301 It may confuse people that the higher the night ACH is, the lower the value of TEs. Artmann  
 302 updated the indicator by multiplying TE with daily climatic cooling potential, ACH, and  
 303 physical parameters of room and air to evaluate the amount of heat removed by night  
 304 ventilation, demonstrating that increasing ACH will remove more heat [13]. Therefore, the  
 305 temperature efficiency is not suitable to evaluate the heat removal effectiveness affected by  
 306 different night ACH, but available to evaluate the performance of night ventilation for different  
 307 scenarios with the same air flow rate.



308  
309 Fig. 3. Sensitivity analysis for TE.

310 **3.2 SA for temperature difference ratio (TDR)**

311 Fig. 4 shows that the WWR is the most important design parameter for TDR for all systems in  
 312 all climates. Similar to the SA for TE, P6 is important for the systems with night mechanical  
 313 ventilation, while not obvious for the systems with night natural ventilation. In cases with the  
 314 daytime AC system, the internal CHTC tends to have a large influence with a positive SRC.  
 315 Moreover, the TDR appears to be sensitive to the building airtightness for the systems with  
 316 night natural ventilation. Increasing the infiltration rate will raise the value of TDR, as it can  
 317 lower the maximum indoor air temperature. As expected, the colder the weather is, the more  
 318 influential the building airtightness. For the all-day natural ventilation system and all-day  
 319 mechanical ventilation system, the internal thermal mass becomes influential, but the sign of its  
 320 SRC is negative for the former system while positive for the latter system. The reason is that  
 321 for the former system, the increase of internal thermal mass raises the maximum indoor air  
 322 temperature while decreases it for the latter system.



323  
324 Fig. 4. Sensitivity analysis for TDR.

### 325 **3.3 SA for decrement factor (DF)**

326 Fig. 5 shows the sensitivity analysis for the DF. Generally, the most influential design  
327 parameters are the internal thermal mass and WWR, whose rank vary slightly in some cases.

328 The increase of WWR raises the fluctuation of indoor air temperature, while the augment of the  
329 internal thermal mass level decreases the fluctuation. P6 is also important for the systems with  
330 night mechanical ventilation systems and insignificant in the systems with night natural  
331 ventilation. Moreover, the value of SRC ranges from -0.4 to -0.2, indicating that the internal  
332 CHTC generally has a big influence on DF. Even though the external thermal mass does not  
333 have the same obvious influence with the internal thermal mass, some attention should be paid  
334 on it, as the value of its SRC ranges from -0.4 to -0.1.

335 In general, the lower the value of DF is, the less the indoor air is affected by the local weather,  
336 which is beneficial for the climate region with high diurnal temperature range and has a great  
337 potential for night ventilation. Although the night ventilation can lower the indoor air  
338 temperature, it also enlarges the indoor air temperature fluctuation which increases the value of  
339 DF since the minimum indoor air temperature reduces more.

340 In such cases, it may be also confusing whether the bigger the value of DF means a better night  
341 ventilation performance. Therefore, the DF may be only suitable for the cases with the same  
342 building information to compare the scenarios with and without night ventilation or the  
343 scenarios with different night airflow rates.

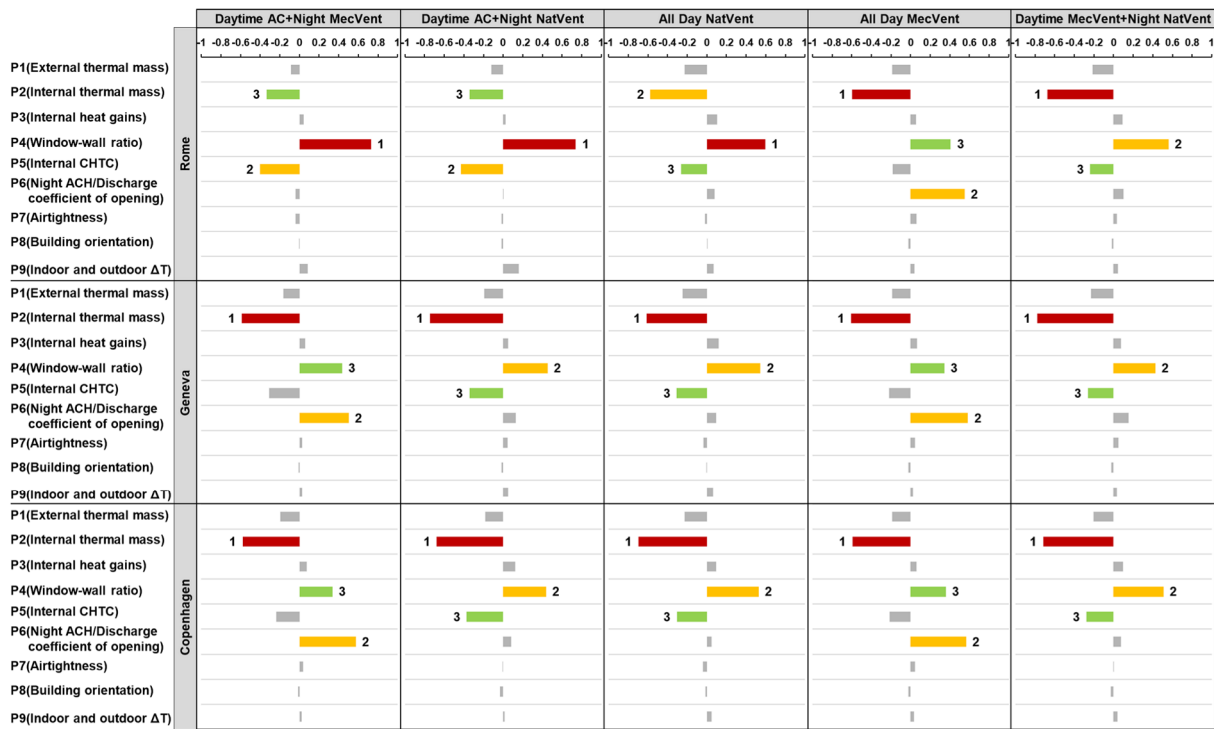


Fig. 5. Sensitivity analysis for DF.

### 3.4 SA for coefficient of performance (COP) and ventilative cooling advantage (ADV)

Fig. 6 shows the influence of design parameters on COP (Fig. 6 (a)) and ADV (Fig. 6 (b)). The COP and ADV are only available for the systems with night mechanical ventilation. It can be concluded that the influence of parameters on COP is almost the same for the two systems. The night ACH is the most important design parameter with a negative SRC, followed by the WWR and internal thermal mass whose signs of SRC are both positive. The reason why the night ACH has a negative SRC on COP is that increasing the air flow rate result in more fan electric consumption, while the amount of cooling energy supplied by the fan does not increase linearly with the fan electric consumption. When increasing the WWR and internal thermal mass level, there will be more excess heat stored during the daytime to be removed by the same night ventilation consumption. Attention should be paid on the building airtightness, as its SRC value is about -0.2, indicating that this parameter has some influence on COP.

The influence of design parameters on ADV varies a lot for different systems and locations. The WWR is important for both systems. However, it has a positive SRC on ADV for daytime

361 AC with night mechanical ventilation system while has a negative SRC for the all-day  
362 mechanical ventilation system. Undoubtedly, increasing the WWR will increase the cooling  
363 system electrical energy consumption of both the scenarios without and with night ventilative  
364 cooling which are  $Q_{el,c}^{ref}$  and  $Q_{el,c}^{scen}$  respectively. The reason why the WWR has a different effect  
365 on ADV for two systems may be that increasing WWR will increase  $Q_{el,c}^{ref}$  more for the former  
366 system while increase  $Q_{el,c}^{scen}$  more for the latter system. Night ACH plays an important role in the  
367 former system, especially in the medium and cold climate regions, but it is not influential for  
368 the latter system. Internal thermal mass ranks second among all design parameters for the  
369 former system but is not important for the latter system. It should be noticed that the P2 has a  
370 negative SRC on ADV for the former system in Rome, while has a positive SRC for the former  
371 system in Geneva and Copenhagen. This indicates that in hot climates, the internal thermal mass  
372 level should not be increased without limit, because the night cooling with relatively high-  
373 temperature ambient air may not be able to remove all the stored excess heat in the thermal  
374 mass during the daytime. Additionally, internal CHTC and internal heat gains have a limited  
375 effect on ADV for both systems.

376



377

Fig. 6. Sensitivity analysis for (a) COP and (b) ADV.

### 378 3.5 SA for cooling requirements reduction (CRR)

379

CRR is not available for the all-day natural ventilation system, because this system does not

380

have daytime mechanical cooling method. Fig. 7 shows that the design parameters have various

381

effects on CRR for different systems and locations. WWR is the most influential parameter in

382

the systems with daytime mechanical ventilation, but not the same influential in the cases with

383

daytime AC. The colder the weather is, the more influential the WWR is for systems with

384

daytime mechanical ventilation. This is probably due to the increasing P4 leads to a different

385

cooling demand increment of the reference scenario without ventilation and the analyzed

386

scenario with ventilation. Generally, the internal thermal mass has a big influence on CRR for

387

the systems with daytime AC, but the influence varies a lot in different locations. It indicates

388

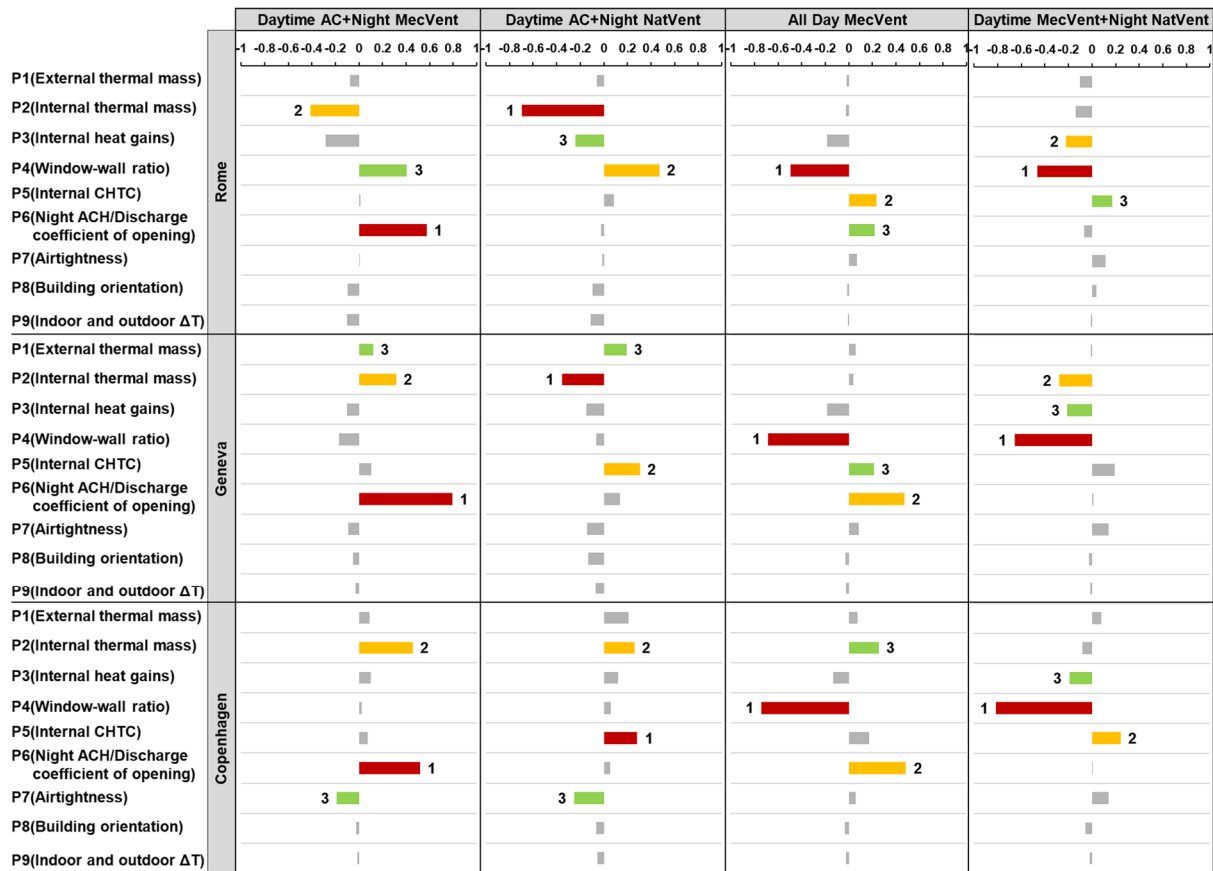
that the internal thermal mass should be arranged properly based on climate conditions and

389

system configurations. Similar to other indicators, the P6 is only significant in the cases with

390 night mechanical ventilation, with a positive SRC. Moreover, the internal CHTC always has a  
 391 small negative SRC on CRR.

392



393 Fig. 7. Sensitivity analysis for CRR.

### 394 3.6 SA for percentage outside the range (POR)

395 Two comfort models from EN 15251 Category II [45] and ASHRAE 55 [54] were applied to  
 396 calculate POR EN 15251 adaptive model category II refers to whether the operative temperature  
 397 falls into the 80% acceptability limits, while ASHRAE 55 simple model indicates whether the  
 398 combination of humidity ratio and the operative temperature is in the ASHRAE 55-2004  
 399 summer clothes region. Fig. 8 shows the sensitivity analysis for the POR based on the two  
 400 comfort models. The PORE and PORA refer to the POR with CEN 15251 Category II and  
 401 ASHRAE 55 simple model respectively.

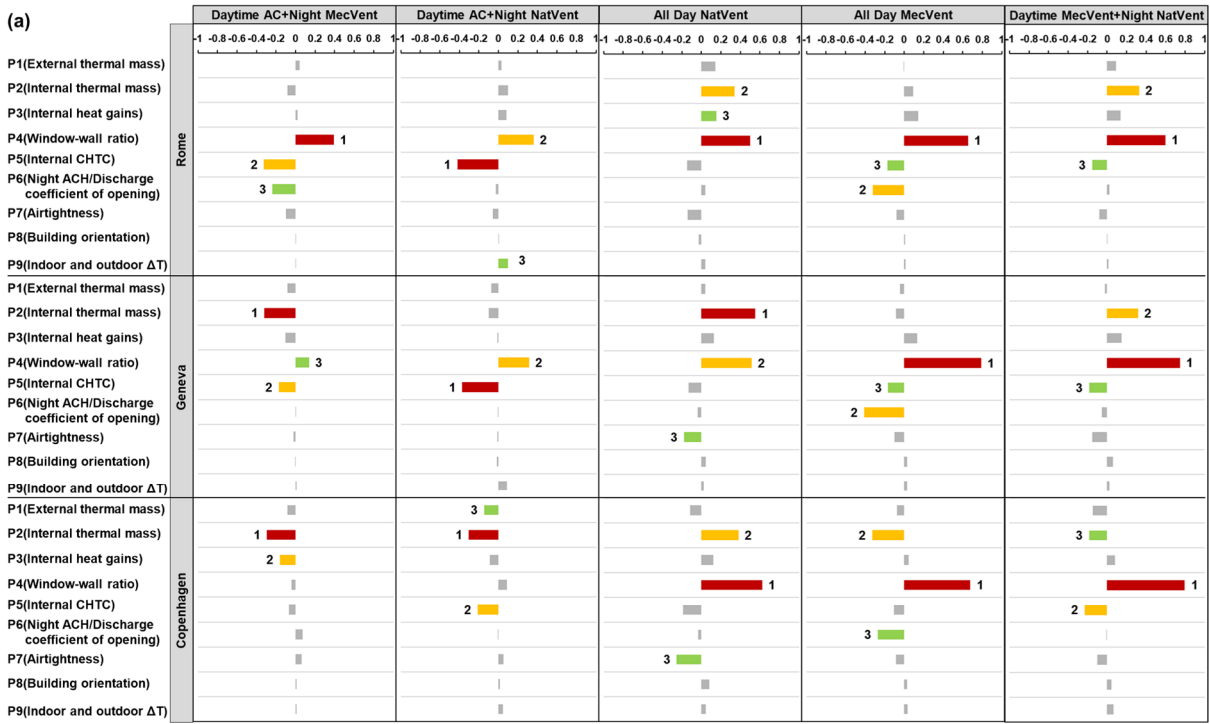
402 For EN 15251 model, the WWR is most influential for the last three systems, while its influence  
 403 is not as obvious for the first two systems which have daytime AC, especially in the cold climate



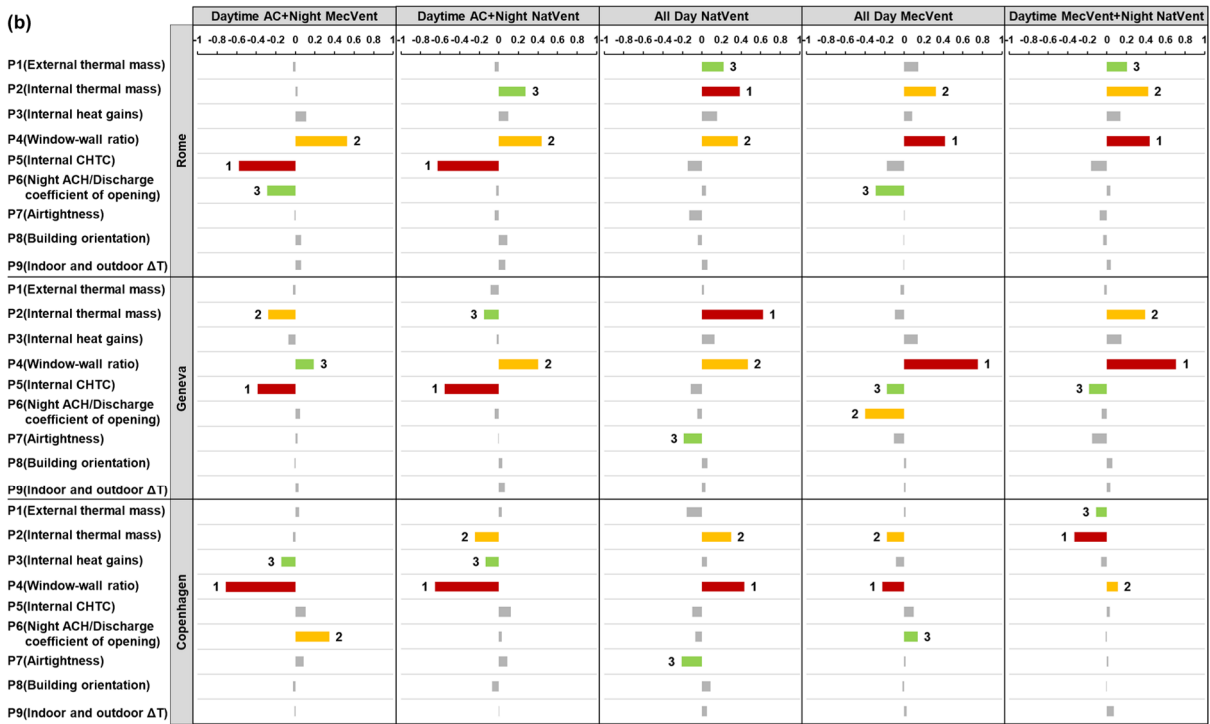
404 region. The effect of the internal thermal mass on POR varies a lot for different systems and  
405 locations. In general, P2 is more influential in medium or cold climate regions, but whether its  
406 SRC for the indicator is positive or negative depends on the systems. On the contrary, the PORE  
407 is more sensitive to the internal CHTC in non-cold climate regions, and the POR always  
408 declines with increasing the internal CHTC. P6 can only make a great difference in this indicator  
409 for the all-day mechanical ventilation system. Additionally, some attention should be paid for  
410 the building airtightness in the all-day natural ventilation system, as its SRC value ranges from  
411 -0.3 to -0.2.

412 Generally, the influence of design parameters on the ASHRAE 55 simple model is similar to  
413 those in EN 15251 adaptive model in most scenarios. However, the influences of WWR,  
414 internal CHTC, and night ACH on PORA are quite different or even reverse between the two  
415 comfort models for the systems with daytime AC and mechanical ventilation system in  
416 Copenhagen. The WWR does not play the same important role in PORA for the last three  
417 systems but is more influential for the first two systems when in comparison with PORE in  
418 Copenhagen, shown in Fig. 8 (b). This might because the ASHRAE 55 simple model takes the  
419 humidity ratio into account, while the EN 15251 adaptive model only considers operative  
420 temperature.

421



422



423

Fig. 8. Sensitivity analysis for (a) POR EN 15251 model and (b) POR ASHRAE 55 model.

424

### 3.7 SA for degree-hours criterion (DhC) and weighted discomfort temperature index (DI)

425

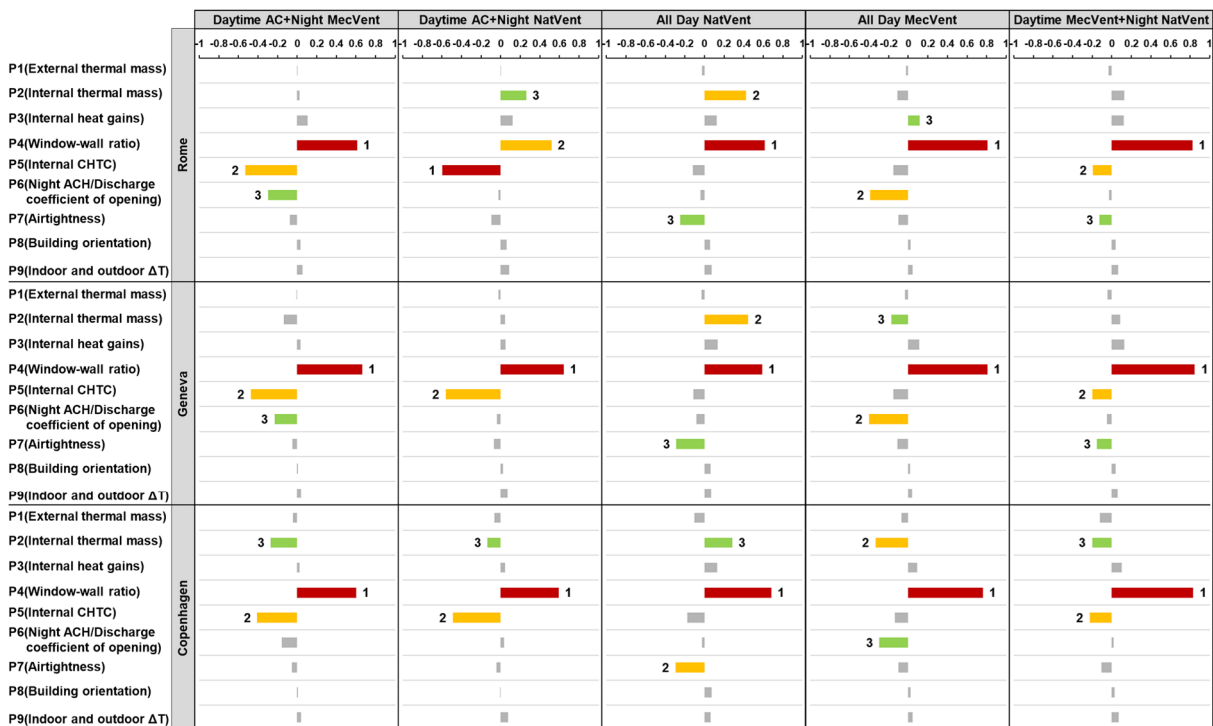
426

As the influence of design parameters on DI are quite similar to those on DhC, the SA results

427

for DI in Fig. 9 are also represented for DhC. The difference between the SA results of DI and

428 DhC is mainly the magnitude of SRC value for some design parameters in some scenarios.  
 429 Generally, for the two thermal comfort indicators, the WWR is most influential, followed by  
 430 the internal CHTC. The influence of internal thermal mass on DhC and DI varies a lot in  
 431 different systems and locations, indicating that the internal thermal mass should be designed  
 432 properly. P6 is important for the systems with night mechanical ventilation but not obvious for  
 433 the systems with night natural ventilation. For all-day natural ventilation system, the building  
 434 airtightness has some impact on the two indicators with negative SRCs. Besides, as expected,  
 435 the colder the weather is, the larger the influence of the building airtightness is.



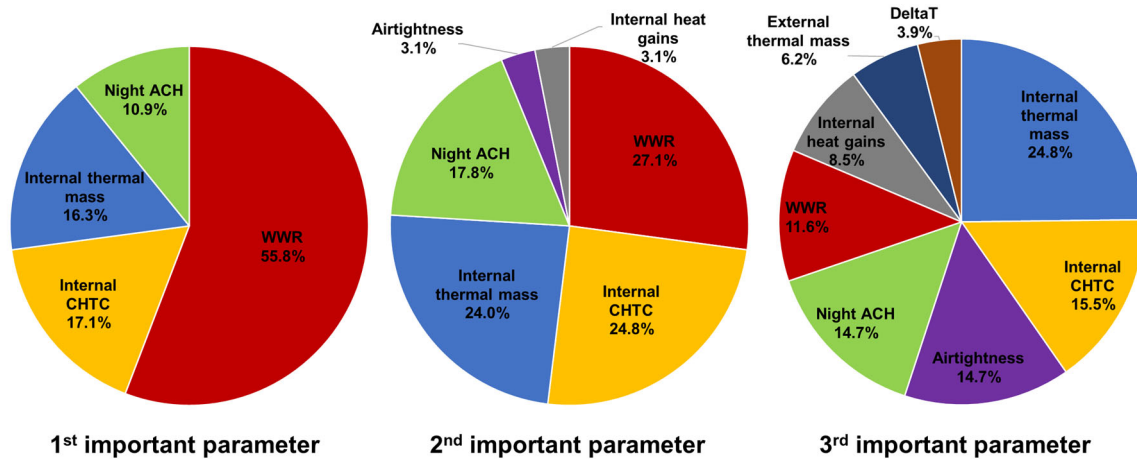
436  
437 Fig. 9. Sensitivity analysis for DhC and DI.

## 438 4 Discussions

### 439 4.1 Importance of design parameters

440 Fig. 10 shows the proportions of the design parameters in the corresponding first, second, third  
 441 important design parameter for all performance indicators. The 1<sup>st</sup> important parameter results  
 442 show that the WWR, internal CHTC, internal thermal mass, and night mechanical ACH are the  
 443 most important design parameters. The 2<sup>nd</sup> important parameter results mean that building

444 airtightness and internal heat gains should be taken into consideration when concerning some  
 445 performance indicators. Apart from the aforementioned six parameters, the results of the 3<sup>rd</sup>  
 446 important parameter show that the external thermal mass and threshold temperature  $\Delta T$  for night  
 447 ventilation should be paid some attention in certain cases.



448 **1<sup>st</sup> important parameter**      **2<sup>nd</sup> important parameter**      **3<sup>rd</sup> important parameter**  
 449 Fig. 10. Pie chart for the top three influential parameters.

450 In the perspective of the influence of each design parameter on all night cooling performance  
 451 indicators based on sensitivity analysis results from Section 3, it can be concluded that the  
 452 WWR always has significant negative SRCs on TE and TDR, but positive SRCs on DF, COP,  
 453 and the thermal comfort indicators. But there is an exception that WWR has a negative SRC on  
 454 the PORA for the systems with daytime AC and the all-day mechanical ventilation system in  
 455 cold climate region. Meanwhile, the signs and values of SRC of WWR on ADV and CRR vary  
 456 a lot depending on the climates or system configurations. Increasing the WWR will raise the  
 457 value of ADV and CRR for the systems with daytime AC, while reduces those value for the  
 458 systems with daytime mechanical ventilation.

459 The internal CHTC have uniform signs of SRCs for each indicator. Increasing the internal  
 460 CHTC will decrease the value of thermal comfort indicators to improve thermal comfort, as  
 461 well as the value of DF to keep the indoor air temperature steadier. On the other hand, increasing  
 462 the internal CHTC will augment heat removal effectiveness (TE & TDR), energy efficiency  
 463 (COP & ADV), and cooling energy use reduction (CRR). It means that increasing the CHTC is

464 always beneficial, which can be achieved by selecting appropriate night ventilation mode or  
465 optimizing the indoor air distribution to enhance the heat transfer area between the cold air and  
466 building elements.

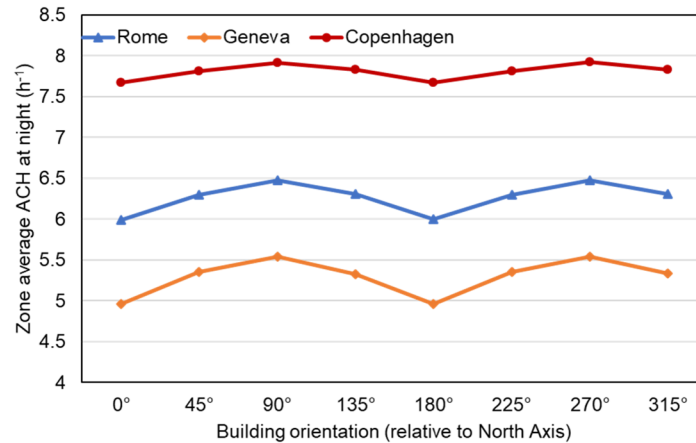
467 The external thermal mass is much less influential than the internal thermal mass. The former  
468 one is only slightly important on the CRR, POR, and DI in some scenarios. The latter one has  
469 positive SRCs for COP and negative SRCs on the DF all the time. But the signs of its SRCs for  
470 the rest of indicators vary a lot based on the night cooling solutions and climates.

471 Night ACH always has positive SRCs on TDR, DF, and CRR, but negative SRCs on TE, COP,  
472 DhC, DI, PORE, and PORA except for the daytime AC with night mechanical ventilation  
473 system in cold climate region. Commonly, increasing the night ACH will reduce the value of  
474 ADV. However, the ADV of the all-day mechanical ventilation system in the medium and hot  
475 climate regions will benefit from the increase of night ACH.

476 The building airtightness is only important on the TDR, COP, ADV, CRR, and the thermal  
477 comfort improvement indicators in some cases. In general, the colder the weather is, the more  
478 influential the building airtightness is. The internal heat gains always have negative SRCs on  
479 TDR. Moreover, it will influence the ADV, CRR, and POR for several scenarios a lot.  $\Delta T$  only  
480 has a limited influence on the TE, TDR, and thermal comfort improvement indicators for the  
481 daytime AC with night natural ventilation system in the hot or medium climate regions.  
482 Increasing the  $\Delta T$  will raise the value of thermal comfort improvement indicators and TE, but  
483 reduce the value of TDR.

484 Building orientation can affect the solar heat gains of the room, and the air flow rate of natural  
485 ventilation. However, the influence of building orientation on the night cooling performance is  
486 quite low, because the solar heat gains were generally low when compared with the internal  
487 heat gains, and the air flow rate does not have a big difference with the orientation changing  
488 (shown in Fig. 11). The reason why the orientation has little influence on the change of air flow  
489 rate is that the opening effectiveness in natural ventilation model is calculated automatically in

490 EnergyPlus, which assumes the window can adjust its angle to make the most of wind under  
491 different wind directions.



492

493 Fig. 11. Zone average ACH at night under different orientations for the daytime AC with night natural  
494 ventilation system in three cities.

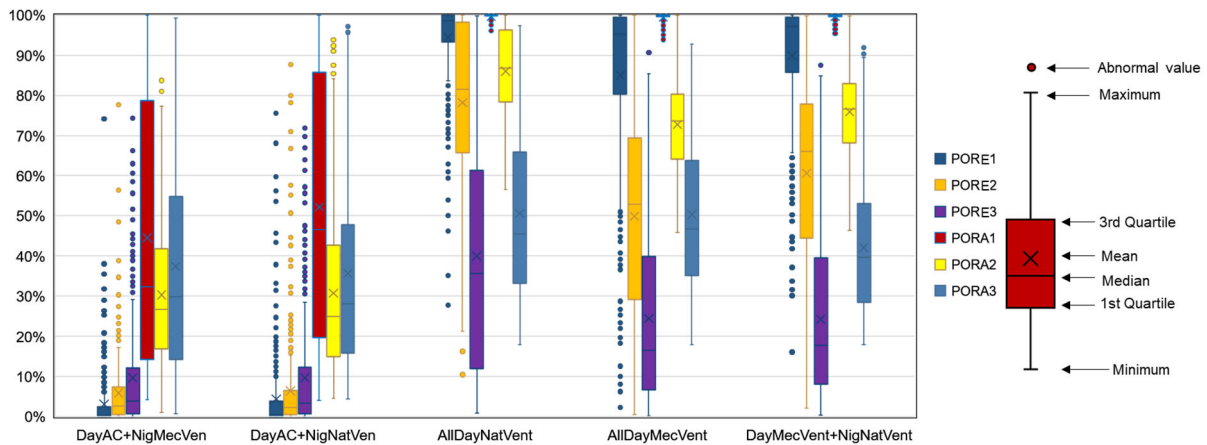
## 495 4.2 Night cooling performance

### 496 4.2.1 Thermal comfort evaluation

497 The ability of night cooling to improve thermal comfort performance depends on the night  
498 cooling solutions as well as the climate. As the magnitude of DhC and DI for different night  
499 cooling solutions varies a lot, the Fig. 12 only shows an overview of the PORE and PORA for  
500 the modeled cases. The numbers 1, 2, and 3 represent Rome, Geneva, and Copenhagen,  
501 respectively.

502 The comparison of the mean and median value of POR between different night cooling solutions  
503 demonstrates that the all-day natural ventilation system has the highest POR, followed by the  
504 daytime mechanical ventilation system with night natural ventilation system, all-day  
505 mechanical ventilation system, daytime AC with night natural ventilation system, and daytime  
506 AC with night mechanical ventilation system. It also can be concluded that the night mechanical  
507 ventilation can provide better thermal comfort with lower POR than night natural ventilation.  
508 Both for night natural and mechanical cooling solutions the best performance in the EN 15251  
509 model are obtained with the daytime AC system in Rome, reaching 0%. While in the

510 ASHARE55 model the best performance of night natural and mechanical ventilation are also  
 511 obtained with the daytime AC system, but in Copenhagen, close to 0% and 5% respectively.  
 512 The value difference between PORE and PORA for the same system in the same climate region  
 513 shows that the thermal comfort criterion selected will come to different results. The ASHRAE  
 514 55 model seems stricter than EN 15251 model, as the PORA is higher than PORE for the same  
 515 system in the same city. There is a clear trend that the PORE for all systems and PORA for the  
 516 latter three systems decrease with the location varying from Rome to Copenhagen. This  
 517 indicates that night ventilation has more application potential in cold climate regions. However,  
 518 no clear trend exists for the PORA for the first two systems in the same condition. The lowest  
 519 median and average value of PORA is in Geneva rather than in Copenhagen. One reason may  
 520 be that the system with daytime AC leaves less excess heat during daytime than other night  
 521 cooling solutions, leading to the overcooling phenomenon caused by night cooling in cold  
 522 climate region. Another reason is that the summer comfort range in ASHRAE 55 simple is  
 523 fixed. Consequently, the zone operative temperature in Rome tending to be higher than the  
 524 comfort range but lower than the comfort range in Copenhagen.



525  
 526 Fig. 12. Box-and-whisker plot of POR for different night cooling solutions.

#### 527 4.2.2 Energy-saving benefit

528 The energy efficiency and ability to reduce the cooling energy use of the different night cooling  
 529 solutions are also very different. Fig. 13 shows the values of COP, ADV, and CRR for different  
 530 night ventilation solutions. Night mechanical ventilation with daytime AC system tends to have

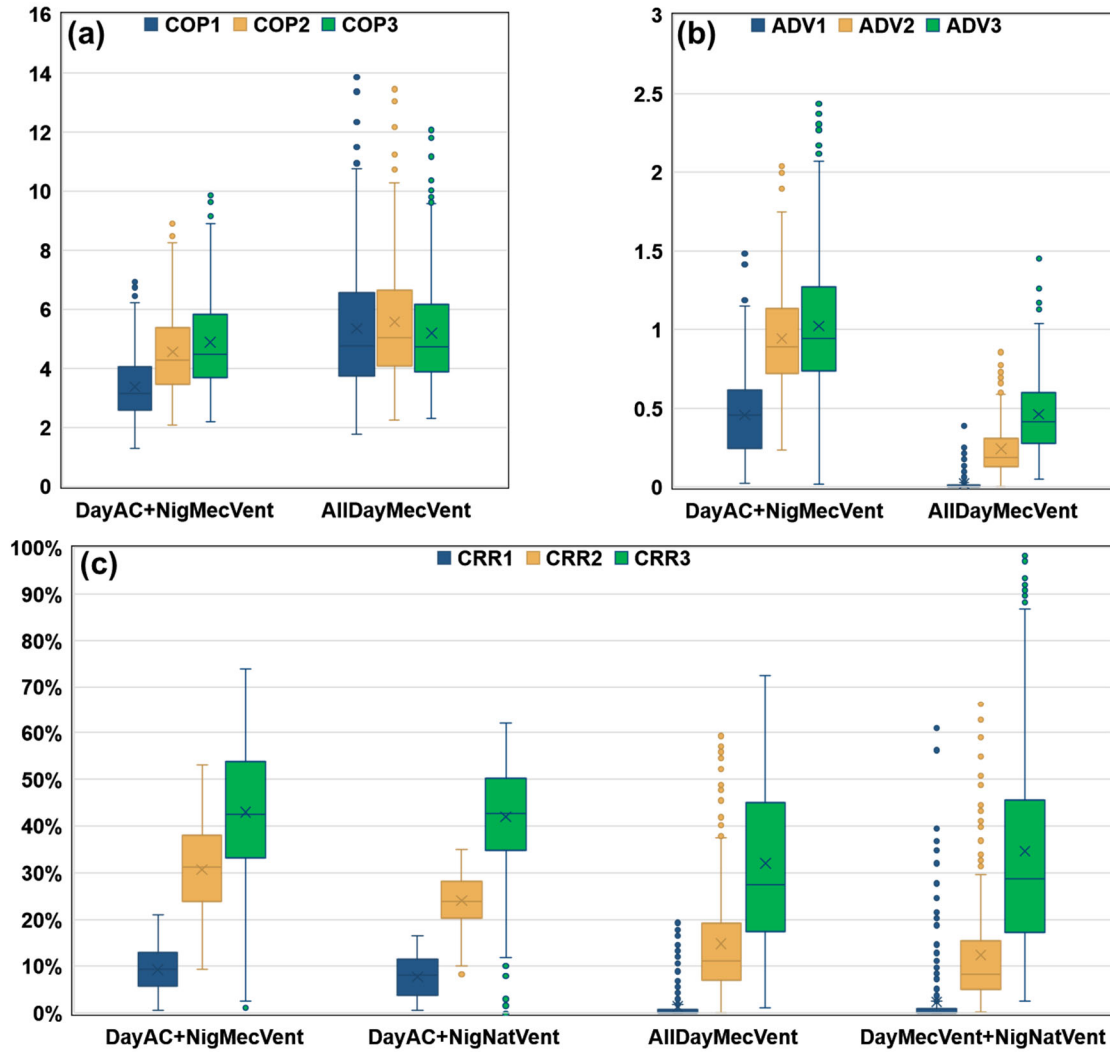
531 a lower COP but higher ADV than with daytime mechanical ventilation system. This is due to  
532 the fact that the daytime AC system can remove more heat and maintain the indoor temperature  
533 at the designed level when compared with the daytime mechanical ventilation system.  
534 Therefore, less excess heat stored at daytime with AC system will lead to lower COP and higher  
535 ADV for night mechanical cooling. ADV can evaluate directly whether the night mechanical  
536 cooling is energy saving or not. However, through the comparison of COP with ADV for all-  
537 day mechanical ventilation in different climate regions, it can be concluded that high COP does  
538 not result in high ADV. COP is not the key indicators to determine whether the night cooling  
539 can save energy or not. The result of CRR clearly demonstrates that there is a trend that the  
540 value of CRR increases with the climate becoming colder.

541 For night natural cooling solutions, the best performance for CRR is obtained with the daytime  
542 mechanical ventilation system in Copenhagen, reaching 97.1%. For night mechanical cooling  
543 solutions, the best performance for ADV and CRR are obtained with the daytime AC system in  
544 Copenhagen, reaching 2.4 and 73.8% respectively. While for the COP of night mechanical  
545 cooling, the best performance is obtained with daytime mechanical ventilation in Rome,  
546 reaching 13.9.

547 In hot climate region, even though the all-day mechanical ventilation can get a value of COP  
548 higher than 10, the night mechanical ventilation does not save energy. Because the ADV is less  
549 than 1. However, the CRR of night natural cooling system indicates that this system can be  
550 energy-saving, with the highest value of more than 60% for the all-day mechanical ventilation  
551 system. While in the cold climate region, all the night ventilation systems can achieve better  
552 performance with a higher value of COP, ADV, and CRR, except for the COP of the all-day  
553 mechanical ventilation system in Copenhagen. Besides, it is easier to save energy for night  
554 mechanical ventilation, with highest and mean value of ADV is 2.4 and 1.1 respectively. For  
555 the medium climate region of Geneva, all the values of three indicators are between that in  
556 Rome and Copenhagen. The result indicates that the colder the climate, the better performance



557 the night cooling can achieve. However, it should be noticed that the ADV of daytime AC with  
 558 night mechanical ventilation could be higher than 1 even in Rome, while close to 0 in  
 559 Copenhagen. Therefore, the night ventilation system should be designed properly based on the  
 560 climate in order to maximize the energy-saving benefit.



561  
 562 Fig. 13. Box-and-whisker plot of COP (a), ADV (b) and CRR (c) for different night cooling solutions

563 **4.3 Applicability of the different performance indicators**

564 The heat removal effectiveness indicators should be used with caution. Firstly, the lack of  
 565 modeling of the temperature distribution in spaces leads to inaccurate values of the temperature  
 566 efficiency. Secondly, a comparison of night cooling performance can only be carried out for  
 567 systems with similar airflow rates by the indicator of TE or with similar building information  
 568 by the indicator of DF. Under the application conditions, the higher the value of TE or DF, the

569 better the performance of night cooling. According to the definition of TDR, the denominator  
570 is the ambient temperature swing which is dependent on the local climate condition. Therefore,  
571 the TDR is not suitable for the same night cooling system to compare the heat removal  
572 effectiveness in different climate regions, but only suitable for the comparison of different  
573 system configurations in the same climate region.

574 The energy-related indicators of COP, ADV, and CRR are used to evaluate energy efficiency  
575 and cooling energy use of night cooling. ADV is very useful for night mechanical ventilation  
576 systems, while CRR is useful for night natural ventilation systems. Though COP provides a first  
577 evaluation of the thermal behavior, the night ventilation energy-saving effect cannot be  
578 quantified. Because for the all-day mechanical ventilation system, the high COP does not result  
579 in high ADV. Therefore, the COP only evaluates the energy efficiency of ventilation at night  
580 time, rather than the energy efficiency for an entire day.

581 For evaluation of the thermal comfort improvement in the daytime, the best performance  
582 indicator is POR because it gives a direct explanation of the percentage outside the comfort  
583 range. Furthermore, it can accompany different thermal comfort models or parameters, such as  
584 PMV, operative temperature, and dry resultant temperature. Both DhC and DI have some  
585 limitations and disadvantages. The biggest limitation is that the thermal comfort threshold value  
586 is too simple, such as the operative temperature 26°C or the indoor air temperature 28°C. In  
587 addition, the two indicators belong to the cumulative index, of which it may be difficult to  
588 evaluate the thermal comfort intuitively.

## 589 **5 Conclusion**

590 This paper applies a global sensitivity analysis to identify the key design parameters affecting  
591 the night ventilation performance. Besides, the applicability and limitations of the performance  
592 indicators are evaluated by the results from the parametric simulation. Based on the results of  
593 the case study, conclusions can be made as follows.

- 594 • The sensitivity analysis shows that the influence of design parameters depends much on the  
595 climate conditions and night ventilation system modes. The WWR, internal CHTC, internal  
596 thermal mass level, and night mechanical ACH of are the most important design parameters.  
597 However, the building airtightness, internal heat gains, external thermal mass level, and  
598 threshold temperature  $\Delta T$  also have limited effect on some indicators in several scenarios.  
599 Small differences on the night cooling performance can be noticed for various building  
600 orientations and different discharge coefficients of the opening.
- 601 • The parametric simulation results show that the way to get the best thermal comfort and  
602 energy-saving benefit for night ventilation is equipped with daytime AC. The colder the  
603 climate, the better performance the night cooling can achieve. Nevertheless, some measures  
604 should be taken to avoid the overcooling effect in cold climate region for the night  
605 ventilation with the daytime AC system.
- 606 • Some performance indicators have limitations and disadvantages. TE is only suitable to  
607 evaluate the performance of different scenarios with similar night ACH, while the DF can  
608 be only applied to evaluate the performance of different night ventilation with similar  
609 building information. TDR is only available to compare the different night cooling systems  
610 in the same climate region. COP is not able to evaluate the energy-saving benefit. DhC and  
611 DI are too simple and not able to evaluate the thermal comfort intuitively. Therefore, the  
612 ADV, CRR, and POR are recommended to evaluate the night ventilation performance.

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