Chapter 2
Ventilative Cooling Principles, Potential and Barriers

Per Heiselberg

Abstract This chapter introduces the main principles of ventilative cooling and the key performance indicators (KPI) to evaluate performance. It also presents and discusses the application potential and limitations as well as includes a critical discussion of barriers to ventilative cooling usage. The chapter is based on the outcome of the international research Annex 62—Ventilative Cooling developed under the Energy in Buildings and Communities (EBC) Programme of the International Energy Agency (IEA).

2.1 Ventilative Cooling Principles

Ventilative Cooling (VC) can be defined as the application of the cooling capacity of the outdoor air flow by ventilation to reduce or even eliminate the cooling loads and/or the energy use by mechanical cooling in buildings, while guaranteeing a comfortable thermal environment.

Ventilative Cooling utilizes the cooling and thermal perception potential of cool outdoor air and the air driving force can be either natural, mechanical or a combination of the two. The most common technique is the use of increased daytime ventilation airflow rates and/or night-time ventilation.

There is a wide range of ventilative cooling principles, and their application depends on climate and microclimate, building type, ventilation approach and user expectations. Ventilative cooling can be combined with other natural cooling solutions utilizing other natural heat sinks in the environment or with mechanical cooling solutions under unfavourable weather conditions.

Ventilative cooling principles for different outdoor climatic conditions and building ventilation systems are summarized in Table 2.1.

P. Heiselberg (✉)
Department of the Built Environment, Aalborg University, Aalborg, Denmark
e-mail: pkh@build.aau.dk

G. Chiesa et al. (eds.), *Innovations in Ventilative Cooling*, PoliTO Springer Series,
https://doi.org/10.1007/978-3-030-72385-9_2

Table 2.1 Overview of typical ventilative cooling strategies applied depending on outdoor climatic conditions and type of ventilation system [1]

Temperature difference[a]	Ventilative cooling	Supplementary cooling options
Cold (ΔT more than 10 °C)	Minimize air flow rate—draught free air supply	–
Temperate (2–10 °C lower than comfort zone)	Increasing air flow rate from minimum to maximum	Strategies for enhancement of natural driving forces to increase air flow rates Natural cooling strategies like evaporative cooling, earth to air heat exchange to reduce air intake temperature during daytime
Hot and dry (ΔT between −2 and +2 °C)	Minimum air flow rate during daytime Maximum air flow rate during nighttime	Natural cooling strategies like evaporative cooling, earth to air heat exchange, thermal mass and PCM storage to reduce air intake temperature during daytime Mechanical cooling strategies like ground source heat pump, mechanical cooling
Hot and humid	Natural or mechanical ventilation should provide minimum outdoor air supply	Mechanical cooling/dehumidification

[a]Temperature difference between indoor comfort temperature and mean outdoor air temperature

2.1.1 Ventilative Cooling During Cold Outdoor Conditions

In wintertime when the outdoor air temperature can be very cold, the main challenge is to introduce outdoor air to the space without creating a high risk of draught and with a minimum use of electricity use for air transport.

For ventilation systems driven by natural forces, another challenge is the balance between required air flow rate to ensure an acceptable indoor air quality and to remove the excess heat load. If the heat load in the building is relatively small, the required air flow rate for indoor air quality might re-move more heat than needed. This will increase the heating system energy use, as effective heat recovery is difficult to be applied to naturally driven systems. Accurate control of the air flow rate is important to minimize the energy use for heating. The system should only be implemented, if the additional energy use for heating in winter associated with natural ventilation is compensated by larger energy savings in the rest of the year. Spaces in buildings with internal heat loads of more than 30 W/m^2 will typically benefit from natural ventilation.

For mechanically driven ventilation systems, the main challenge in exploiting outdoor air for cooling is to minimize the energy use for air transport. Typical mechanical systems cannot provide cold outdoor air to the building without increasing the risk of draught of the occupants. Supply air temperature is therefore increased by efficient heat recovery. This reduction in cooling capacity is compensated by an increased air flow rate up to 4–5 times the required for indoor air quality purposes. Increased pressure loss for heat recovery and in the air distribution system, increases the energy use for air transport considerably and in some cases outweighs the benefit of the "free cooling capacity" of the outdoor air. Solutions that can provide low temperature air supply without creating a draught risk for the occupants are therefore essential for mechanical ventilation system, especially in winter.

2.1.2 Ventilative Cooling During Temperate Outdoor Conditions

Under temperate conditions, outdoor air can be provided to the building and the occupied zone without creating a risk of draught. The air flow rate should be controlled according to the temperature and will typically be higher than required to ensure an acceptable indoor air quality. As in naturally driven systems there is no energy use for heating, cooling or air transport, the control requirements for the air flow rate are not very strict and technically relatively simple systems (like manual or automatic window opening in the façade) can handle the ventilative cooling appropriately. However, in periods with small temperature differences between indoor and outdoor air, where the naturally driving buoyancy forces are limited, it might be necessary to enhance them by implementing additional technical solutions to the building. In windy climates, solutions that can enhance wind forces are typically suitable (wind catchers, high positioned roof openings, etc.), while in sunny climates enhancement of buoyancy forces by solar chimneys might be useful.

For mechanically driven system, the cooling capacity can be kept constant at increasing outdoor air temperature by reducing the heat recovery efficiency. Not until outdoor air temperatures is above 18–19 °C, the cooling capacity will drop as increase in air flow rates is not possible or only to a very limited extend.

To enhance the ventilative cooling capacity, it is important to position the air intakes in a cool environment (shaded side of the building). It might also be necessary to further reduce the outdoor air intake temperature by supplementary natural cooling solutions like ground cooling (earth to air heat exchange) or evaporative cooling.

2.1.3 Ventilative Cooling During Hot Outdoor Conditions

In summer, in dry climates with high outdoor air temperatures during day-time, the air flow rates should be controlled to a minimum to ensure an acceptable indoor air quality and minimum additional heat load on the building. Effective night-time ventilation should be applied to remove the absorbed heat during daytime by cooling the building thermal mass. If the night-time cooling capacity is high enough and the building is well-designed with well-balanced glass area in the facades, efficient solar shading and exposed thermal mass, the next day's indoor temperature profile will be lower than outdoor temperature. Otherwise, supplementary natural cooling solutions and/or mechanical cooling will be required to reduce daytime outdoor air in-take temperatures in the warmest periods.

In hot and humid climates, naturally driven ventilative cooling will not be useful in the warm period. Mechanical ventilation systems are required to be supplemented by mechanical cooling to ensure a constant high cooling capacity regardless of the outdoor temperature and humidity.

2.1.4 Application of Hybrid Solutions

As aforementioned, adopting naturally or mechanically driven ventilation systems for ventilative cooling presents different challenges.

Naturally driven ventilation systems are most effective in buildings with high heat loads in winter, in buildings with low heat loads in summer and in periods of the year where the outdoor temperatures are temperate, while mechanical systems are more suitable in buildings with relative low heat loads in winter, in buildings with high heat loads in summer and in periods of the year where the outdoor temperature is either very cold (utilization of heat recovery to decrease energy use) or very warm (mechanical cooling can be applied to ensure thermal comfort).

In many cases it can be beneficial from both an energy and a thermal comfort point of view to combine the two different types of ventilation systems to exploit their different strengths and avoid their weaknesses. The most appropriate strategy for the combination of systems will depend on the outdoor temperature (climate) as well as the building type and the overall cooling demand.

In cold climates, the typical combination is the use of mechanically driven ventilation in the winter season and naturally driven ventilation during intermediate and summer seasons. In temperate climates, naturally driven ventilation can be used during the whole year. In warm climates, naturally driven ventilation is used in the winter period, while mechanically driven ventilation is preferable in the rest of the year.

Different systems can also be used at different times of the day. Generally, mechanically driven ventilation is used during occupancy hours and naturally driven ventilation is activated at night-time to increase the cooling capacity at night.

2.2 Climatic Potential for Cooling During Nighttime

The climatic potential for the ventilative cooling of buildings by night-time ventilation in Europe is evaluated in [2]. A method was developed which is basically suitable for all building types, regardless of building-specific parameters. This was achieved by basing the approach solely on a building temperature variable within a temperature band given by summertime thermal comfort.

2.2.1 *Definition of CCP*

Degree-days or degree-hours methods are often used to characterise a climate's impact on the thermal behaviour of a building. The daily climatic cooling potential, CCP_d, was defined as degree-hours for the difference between building temperature, T_b and external air temperature, T_e (Fig. 2.1):

$$CCP_d = \sum_{t=t_i}^{t_f} m_{d,t}\left(T_{b(d,t)} - T_{e(d,t)}\right) \quad \begin{cases} m = 1h \quad if \quad T_b - T_e \geq \Delta T_{crit} \\ m = 0 \quad if \quad T_b - T_e < \Delta T_{crit} \end{cases} \tag{2.1}$$

where t stands for the time of day, with $t \in \{0,\ldots, 24\ h\}$; t_i and t_f denote the initial and the final time of night-time ventilation, and ΔT_{crit} is the threshold value of the temperature difference, when night-time ventilation is applied. In the numerical analysis, it was assumed that night-time ventilation starts at $t_i = 19$ h and ends at $t_f = 7$ h. As a certain temperature difference is needed for effective convection, night ventilation is only applied if the difference between building temperature and external temperature is >3 K.

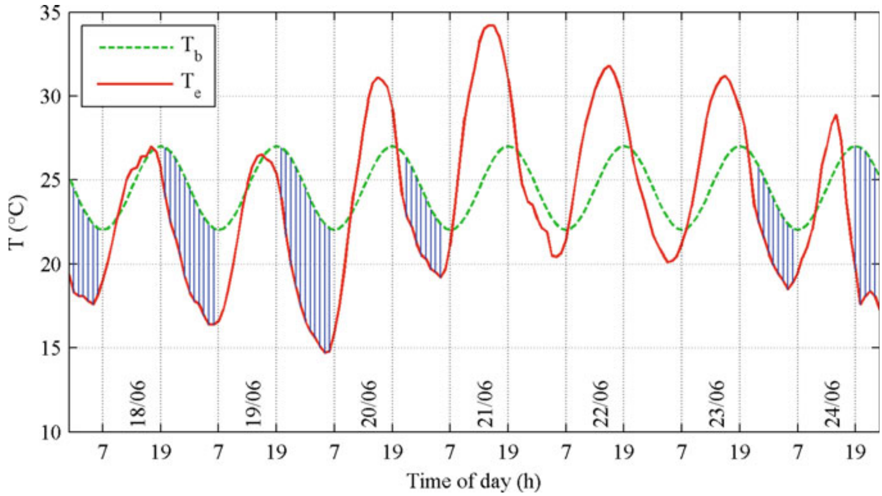

Fig. 2.1 Building temperature, T_b and external air temperature, T_e during one week in summer 2003 for Zurich SMA (ANETZ data). Shaded areas illustrate graphically the climatic cooling potential, CCP [2]

As heat gains and night-time ventilation are not simultaneous, energy storage is an integral part of the concept. In the case of sensible energy storage, this is associated with a variable temperature of the building structure. This aspect is included in the model by defining the building temperature as a harmonic oscillation around 24.5 °C with amplitude of 2.5 K:

$$T_{b(t)} = 24.5 + 2.5cos\left(2\pi\frac{t - t_i}{24}\right) \tag{2.2}$$

The maximum building temperature occurs at the starting time of night ventilation, and given a ventilation time of 12 h, the minimum building temperature occurs at the end time (Fig. 2.1). The temperature range $T_b = 24.5$ °C ± 2.5 °C corresponds to that recommended for thermal comfort in offices [3].

2.2.2 Practical Significance of CCP

To discuss the practical significance of the calculated degree-hours, an example shall be given. It is assumed that the thermal capacity of the building mass is sufficiently high and therefore does not limit the heat storage process. If the building is in the same state after each 24 h cycle, the daily heat gains Qd(Wh) stored to the thermal mass, equal the heat which is discharged by night ventilation:

$$Q_d = \dot{m} \cdot c_p \cdot CCP_d \tag{2.3}$$

The effective mass flow rate is written as $\dot{m} = A_{Floor} * H * \eta * ACR * \rho$, where A_{Floor} is the floor area [m^2] and H the height of the room [m], ACR the air change rate [h^{-1}] and η a temperature efficiency, which is defined as $\eta = (T_{out} - T_e)/(T_b - T_e)$ and takes into account the fact that the temperature of the outflowing air T_{out} is lower than the building temperature T_b. The density and the specific heat of the air are taken as $\rho = 1.2$ kg/m^3 and $c_p = 1000$ J/(kg K). Assuming a room height of H = 2.5 m and a constant effective air change rate of $\eta * ACR = 6$ h^{-1} yields:

$$\begin{aligned}\frac{Q_d}{A_{floor}} &= H \cdot \eta \cdot ACR \cdot \rho \cdot c_p \cdot CCP_d \\ &= \frac{2.5\ \text{m} \cdot 6\ \text{h}^{-1} \cdot 1.2\ \text{kg/m}^3 \cdot 1000\ \text{J/kg K}}{3600\ \text{s/h}} CCP_d = 5\frac{W}{m^2K}CCP_d\end{aligned} \tag{2.4}$$

For the climatic cooling potential needed to discharge internal heat gains of 20 W/m^{2}K and solar gains of 30 W/m^{2}K during an occupancy time of 8 h follows:

$$CCP_d = \frac{Q_d}{A_{floor}}/5\frac{W}{m^2K} = \frac{(20 + 30) \cdot 8}{5}\text{kh} = 80\ \text{kh} \tag{2.5}$$

This example should be seen as a rough estimation only, as solar and internal gains of an office room can vary substantially depending on the type of building use, local climate, and the solar energy transmittance and orientation of the façade.

2.2.3 Nighttime Cooling Potential

The degree-hour method was applied for a systematic analysis of the potential for nighttime cooling in different climatic zones of Europe. Semi-synthetic climate data [4] from 259 weather stations was used to map the cumulative frequency distribution of CCP for 20 European locations (Fig. 2.2). These charts show the number of nights per year when CCP exceeds a certain value.

In the whole of Northern Europe (including the British Isles) a very significant climatic cooling potential was found, and therefore passive cooling of buildings by night-time ventilation seems to be applicable in most cases. In Central, Eastern and even in some regions of Southern Europe, the climatic cooling potential is still significant, but due to the inherent stochastic properties of weather patterns, series of warmer nights can occur at some locations, where passive cooling by night-time ventilation might not be sufficient to guarantee thermal comfort. If lower thermal comfort levels are not accepted during short periods of time, additional cooling systems are required. In regions such as southern Spain, Italy and Greece climatic cooling potential is limited and night cooling alone might not be sufficient to provide good thermal comfort during all the year. Nevertheless, night-time ventilation can be used in hybrid cooling systems during spring and fall.

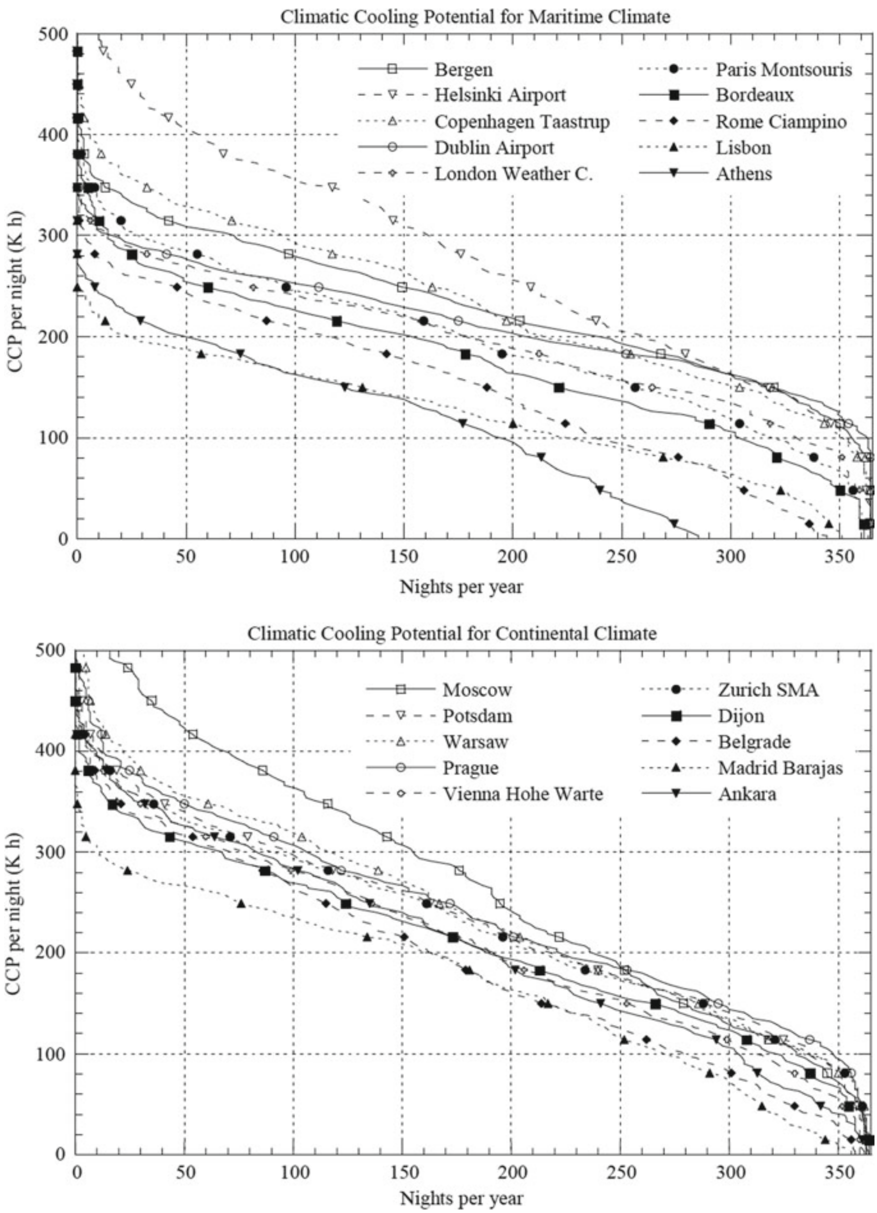

Fig. 2.2 Cumulative frequency distribution of CCP for maritime (top) and continental (bottom) locations [2]

2.3 Simplified Tool for Prediction of Ventilative Cooling Potential

Ventilative cooling is dependent on the availability of suitable external conditions to provide cooling. As buildings with different use patterns, envelope characteristics and internal loads level react differently to the external climate condition, the ventilative cooling potential analysis cannot abstract from building characteristics and use. In an assessment of the potential it is important to limit the evaluation of the cooling potential to the period where cooling is needed. Therefore, it is necessary, to look at the outdoor climate as well as the expected cooling need of the building.

A ventilative cooling potential tool (VC Tool) was developed within the Annex 62 project with the aim to assess the potential effectiveness of ventilative cooling strategies by taking into account building envelope thermal properties, occupancy patterns, internal gains and ventilation needs. It has to be considered only as a preliminary analysis on the assumption that the thermal capacity of the building mass is sufficiently high and therefore does not limit the heat storage process.

The ventilative cooling potential tool refers to the method proposed in [5] and is further developed within the IEA EBC Annex 62 activities [1].

This method derives from the energy balance of a well-mixed single-zone delimited by heat transfer surfaces. It assumes that a heating balance point outdoor air temperature can be determined below which heating must be provided to maintain indoor air temperatures at a defined internal heating set point temperature. Therefore, when outdoor dry bulb temperature exceeds the heating balance point temperature, direct ventilation is considered useful to maintain indoor conditions within the comfort zone. At or below the heating balance point temperature, ventilative cooling is no longer useful but heat recovery ventilation should be used to meet minimum air change rates for indoor air quality control and reduce heat losses.

This relies on the assumption that the accumulation term of the energy balance is negligible. It is a reasonable assumption if either the thermal mass of the zone is negligibly small or the indoor temperature is regulated to be relatively constant. Under these conditions, the energy balance of the zone is steady-state and can provide an approximate measure to characterize the ventilative cooling potential of a climate [5].

The analysis is based on a single-zone thermal model applied to user-input weather data on hourly basis. For each hour of the annual climatic record of the given location, an algorithm splits the total number of hours when the building is occupied into the following groups:

Ventilative Cooling mode [0]: when the outdoor temperature is below the heating balance point temperature, no ventilative cooling is required since heating is needed.

Ventilative Cooling mode [1]: Direct ventilation with airflow rate maintained at the minimum required for indoor air quality can potentially ensure thermal comfort when the outdoor temperature exceeds the balance point temperature, yet it falls below the lower temperature limit of the comfort zone.

Ventilative Cooling mode [2]: Direct ventilative cooling with increased air-flow rate can potentially ensure comfort when the outdoor temperature is within the range of comfort zone temperatures. In this case, the tool calculates the airflow rate required to maintain the indoor air temperature within the comfort zone temperature ranges. Direct ventilative cooling is not considered useful if the temperature difference between indoor and outdoor is below 3 K.

Ventilative Cooling mode [3]: Direct evaporative cooling (DEC) can potentially ensure comfort even if direct ventilation alone is not useful because the outdoor temperature exceeds the upper temperature limit. The evaporative cooling potential is considered when the expected temperature of the treated air is within the upper operative temperature limit minus 3 K. The expected outlet temperature of a DEC system is calculated according to [6, 7]. Moreover, an indirect limitation on DEC potential to prevent too high relative humidity values is also included, fixing a maximum reference for the outdoor wet bulb temperature—see [7] for residential buildings and [8] for offices.

Ventilative Cooling mode [4]: Direct ventilative cooling is not useful when the outdoor temperature exceeds the upper temperature limit of the comfort zone. Furthermore, this limit is also overtaken from the expected DEC outlet temperature.

If direct ventilative cooling is not useful for more than an hour during the occupied time, the night-time climatic cooling potential (NCP) over the following night is evaluated using the method described in [2]. Night-time ventilation is calculated by assuming that the thermal capacity of the building mass is sufficiently high and therefore all the exceeding internal gains can be stored in the building mass. Night-time cooling potential (NCP) over the following night is evaluated as the internal gains that may be offset for a nominal night-time air change rate.

Figure 2.3 shows as an example a prediction with the tool of the ventilative cooling potential for a building divided into the different ventilative cooling modes as well as the required flow rate.

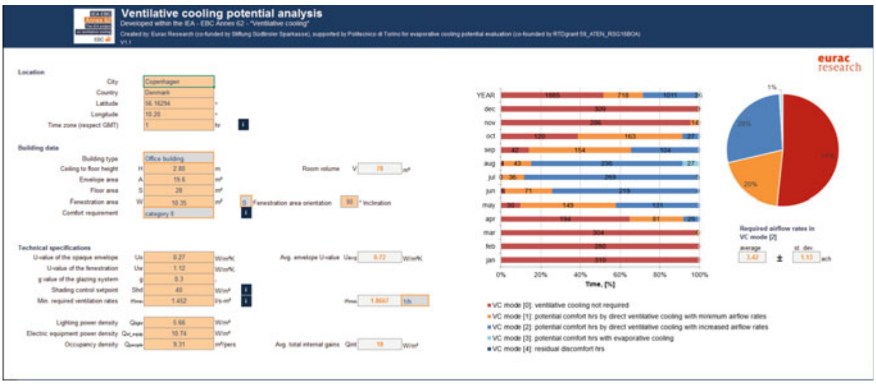

Fig. 2.3 Ventilative cooling potential and required air flow rate for a building predicted by the VC tool [1]

2.4 Simplified Method for Calculation of Opening Areas

The opening area in the building required to deliver the ventilative cooling air flow rate depend on the outdoor conditions and the position of the openings, i.e. the ventilation strategy applied.

2.4.1 Single-Sided Ventilation

In a single-sided ventilation strategy with only one opening used for ventilative cooling the necessary opening area can with reference to EN 16798-7:2017, [9] be calculated as:

$$A_{\mathit{eff},e} = \frac{2 \cdot q_v}{1000\sqrt{maks\left(C_v \cdot v_{ref}^2; C_{t,e} \cdot h_v \cdot abs(t_i - t_u)\right)}} \tag{2.6}$$

where

$A_{\mathit{eff},\,e}$ is the effective opening area for single sided ventilation (m^2/m^2 floor area)

q_v is the air flow rate (l/s m^2 floor area)

C_v is coefficient taking into account wind speed in airing calculations = 0.001 (1/(m/s))

$C_{t,e}$ is coefficient taking into account stack effect in airing calculations = 0.0035 ((m/s)/(mK))

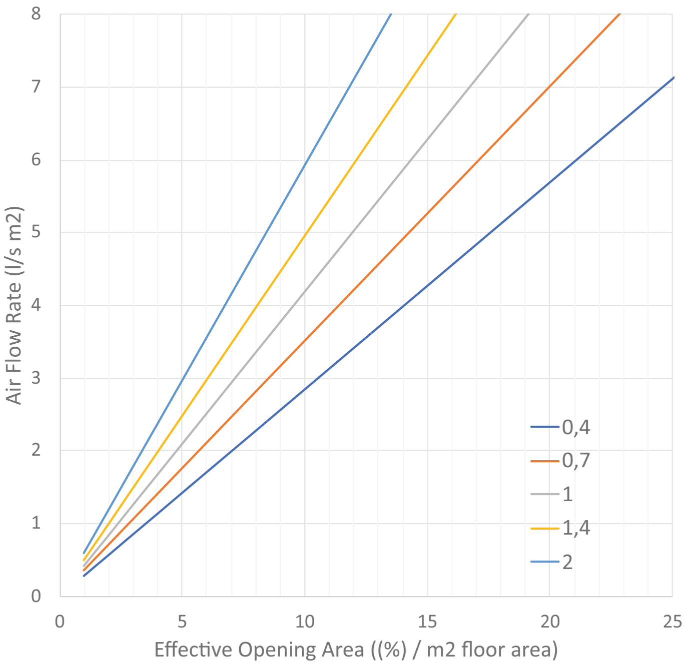

Fig. 2.4 Example of air flow rate per m^2 floor area for single-sided ventilative cooling as a function of opening area for different opening heights under the conditions $\Delta T = 2$ °C and $v_{ref} = 1.8$ m/s

v_{ref} is reference wind speed in 10 m height (m/s)
h_v is opening height (m)
t_i is indoor temperature (K)
t_u is outdoor temperature (K)

Figure 2.4 illustrates an example of its use. Opening height is very important for the necessary opening area.

2.4.2 *Stack Ventilation*

In a stack ventilation strategy with multiple openings positioned in two different heights in the same facade, the necessary opening area for ventilative cooling can with reference to [10] be calculated as:

$$A_{eff,o} = \frac{q_v}{1000\sqrt{C_{t,o} \cdot h_{st} \cdot abs(t_i - t_u)}} \tag{2.7}$$

where

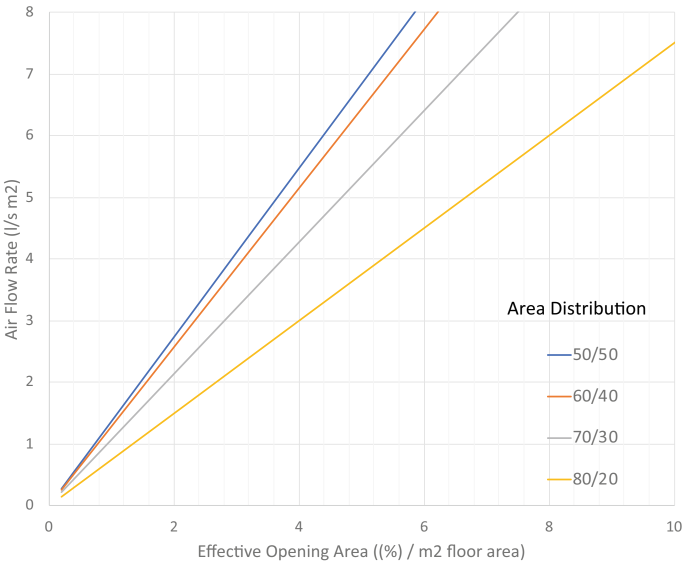

Fig. 2.5 Example of air flow rate per m^2 *floor area for stack ventilation as a function o*f opening area and for different relative distribution of the area between openings under the condition $\Delta T = 2$ °C and $h_{st} = 3.0$ m

$A_{eff,\,o}$ is the effective opening area for stack ventilation (m^2/m^2 floor area)
q_v is the air flow rate (l/s m^2 floor area)
$C_{t,o}$ is coefficient taking into account stack effect in airing calculations = 0.025 ((m/s)/(mK))
h_{st} is the effective height for stack ventilation (m)
t_i is indoor temperature (K)
t_u is outdoor temperature (K)

The effective height for stack ventilation, h_{st}, can be found as the height difference between the middle of the top and bottom opening, respectively. The necessary opening area will depend on the distribution of the area between the openings in the two different heights, see Fig. 2.5.

2.4.3 Cross Ventilation

In a cross-ventilation strategy with several openings in different facades the necessary opening area can with reference to EN 16798-7:2017, [9] be calculated as:

$$A_{eff,v} = \frac{q_v}{1000 \cdot C_D \cdot v_{ref}\sqrt{\Delta C_p}} \qquad (2.8)$$

where

$A_{eff,\,v}$ is the effective opening area for cross ventilation (m^2/m^2 floor area)

q_v is the air flow rate (l/s m^2 floor area)

C_D is a discharge coefficient for air flow through an opening = 0.60 (−)

v_{ref} is reference windspeed in 10 m height (m/s)

ΔC_p is the difference in wind pressure between different opening orientations (−)

The necessary opening area will depend on how the opening area is distributed on the different openings in the different façade orientations. EN16798-7:2017 includes a detailed methodology for calculation of the effective opening area for cross ventilation. A minimum opening area was obtained when the opening area is divided equally between openings (Fig. 2.6).

2.5 Key Performance Indicators

Key Performance Indicators (KPIs) are quantifiable measures used to evaluate design goals and to provide means for the measurement and monitoring of the progress of the design towards those goals. In IEA EBC Annex 62 national experts have discussed and developed KPIs to represent the performance of ventilation cooling [1].

2.5.1 Thermal Comfort

Thermal comfort performance cannot be represented well by a single indicator [11]. A set of indicators is needed. The standard EN 15251:2007 proposes methods for long-term evaluation of general thermal comfort conditions, where the combination of the "Percentage Outside the Range Index" (method A) and the "Degree-hours Criterion" (method B) enable the evaluation of both frequency and severity of overheating and overcooling occurrences. The reference comfort temperature can be derived from the Fanger model or the adaptive comfort model.

The Percentage Outside the Range (POR) index [%] calculates the percentage of occupied hours, when the PMV (Predicted Mean Vote) or the operative temperature is outside a specified range.

$$POR = \frac{\sum_{i=1}^{Oh}(wf_i \cdot h_i)}{\sum_{i=1}^{Oh} h_i} \tag{2.9}$$

where wf is a weighting factor which depends on the comfort 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.

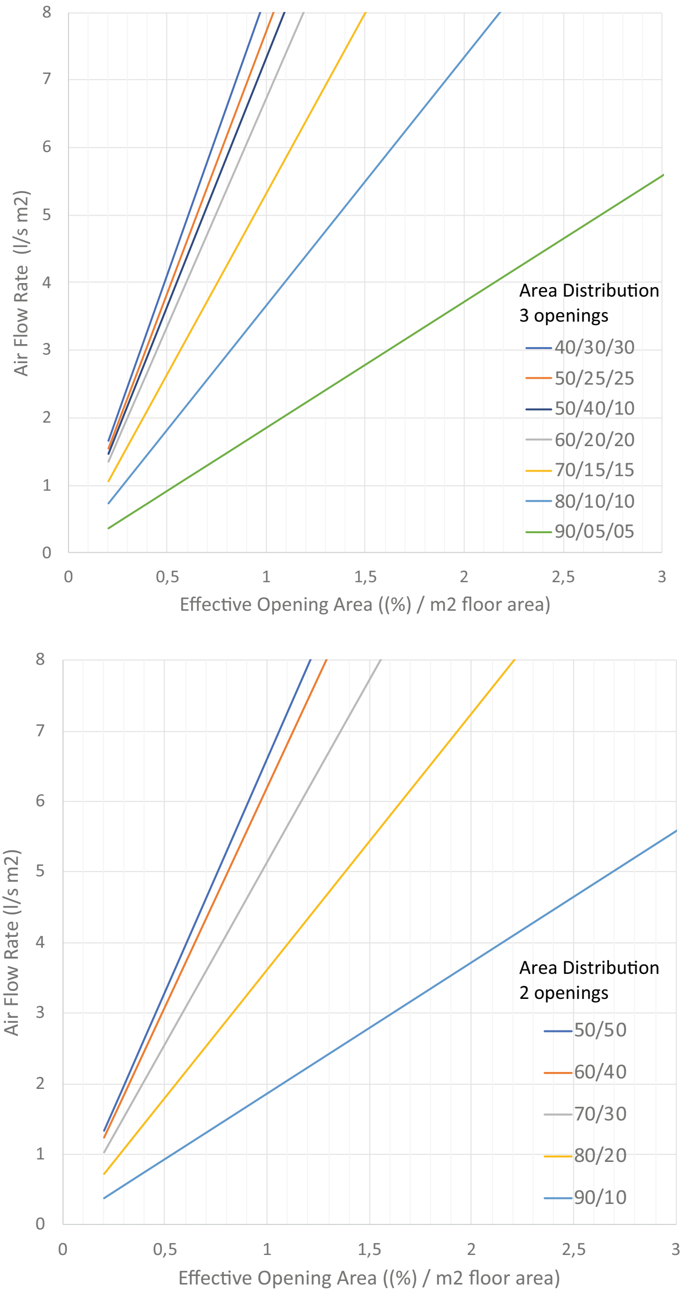

Fig. 2.6 Example of air flow rate per m^2 floor area for cross ventilation as a function of opening area under the condition of $v_{ref} = 1.8$ m/s, $\Delta Cp = 0.75$), when the opening area is distributed on two or three openings, respectively

According to the Degree-hours criterion (DhC) the time during which the actual operative temperature exceeds the specified range during the occupied hours is weighted by a factor which is a function of how many degrees the range has been exceeded.

$$DhC = \sum_{i=1}^{Oh} (wf_i \cdot h_i) \tag{2.10}$$

where weighting factor wf is here calculated as the module of the difference between actual or calculated operative temperature, θ_{op}, at a certain hour, and the lower or upper limit, $\theta_{op,limit}$, of a specified comfort range.

In case the comfort range is expressed in terms of PMV, the comfort operative temperature range has to be estimated by making assumptions on clothing and metabolic activity.

In case of compliance demonstration, it is recommended to use a concise indicator able to summarize the building performance in terms of thermal comfort. Previous studies [12], identified the long-term percentage of dissatisfied (LPD) index [%] as the optimal index to evaluate comfort conditions.

$$LPD(LD) = \frac{\sum_{t=1}^{T} \sum_{z=1}^{Z} \left(p_{z,t} \cdot LD_{z,t} \cdot h_t\right)}{\sum_{t=1}^{T} \sum_{z=1}^{Z} \left(p_{z,t} \cdot h_t\right)} \tag{2.11}$$

where t is the counter for the time step of the calculation period, T is the last progressive time step of the calculation period, z is the counter for the zones of a building, Z is the total number of the zones, $p_{z,t}$ is the zone occupation rate at a certain time step, $LD_{z,t}$ is the *Likelihood of dissatisfied* inside a certain zone at a certain time step and h_t is the duration of a calculation time step (e.g., one hour).

The Likelihood of dissatisfied can be formulated in different ways depending on the reference comfort model [12]. This indicator is concise, symmetric, robust and can be derived from building energy simulation outputs or long-term monitoring and can be used to compare the performance of different buildings as it is expressed in terms of percentage.

2.5.2 Energy

Existing energy indicators suit to all building typologies and evaluate active systems only. Energy indicators only implicitly consider the benefits of passive solutions, as energy need reduction, or the side effects, as the increase of heating need due to cold draughts or higher infiltrations or the increase of auxiliary energy consumption for control and automation. Passive systems are implicitly taken into account in the energy need calculation, but the related energy savings are not explicitly shown.

These calculation methods do not allow fair comparison between passive and active design options and other competitive measures.

Furthermore, most of the existing indicators consider either cooling or ventilation energy use, but not the total energy use for cooling and ventilation. Free cooling is meant to reduce or to avoid active cooling. Energy consumption of the fans is used to reduce or substitute the active cooling energy. The energy use for hygienic ventilation is usually not disaggregated from the overall energy consumption for ventilation.

From these considerations arose the need for an energy indicator or a set of indicators able to tackle the following aspects:

cooling need and/or energy savings related to ventilative cooling;
ventilation need and/or savings related to ventilative cooling only, possibly excluding the energy needed by hygienic ventilation;
possible drawbacks on energy behaviour during heating season, i.e. increase of heating need due to cold draughts or higher infiltrations, auxiliary energy consumption for control and automation;
ventilative cooling effectiveness as the match of cooling need and ventilative cooling potential.

In IEA EBC Annex 62 a new set of energy indicators was developed and tested for the evaluation of ventilative cooling system performances [1]. These are presented in the following.

The first indicator, the Specific Primary Energy Consumption of a ventilative cooling system, is meant to express the primary energy consumed by the ventilative cooling system per heated floor area.

$$Q_{pe,vc} = Q_{pe,v} + Q_{pe,h} + Q_{pe,c} - Q_{pe,v\_hyg} \tag{2.12}$$

where $Q_{pe,v}$ is the annual primary energy consumption of the fan, $Q_{pe,h}$ and $Q_{pe,c}$ are the annual primary energy consumption for space heating and cooling respectively and Q_{pe,v_hyg} is the annual primary energy consumption of the fan when operating for hygienic ventilation.

The second indicator, the Cooling Requirements Reduction (CRR), is meant to express the percentage of reduction of the cooling demand of a scenario in respect to the cooling demand of the reference scenario. It can be easily calculated by post processing outcomes of building energy simulation runs of a reference scenario (e.g. mechanically cooled building) and a ventilative cooling scenario (e.g. natural night cooling and daytime mechanical cooling). Therefore, it is particularly suitable to compare different design scenarios and drive design decisions.

$$CRR = \frac{Q_{t,c}^{ref} - Q_{t,c}^{scen}}{Q_{t,c}^{ref}} \tag{2.13}$$

where $Q_{t,c}^{ref}$ is the cooling demand of the reference scenario and $Q_{t,c}^{scen}$ is the cooling demand of the ventilative cooling scenario.

This indicator can range between -1 and $+1$. If CRR is positive, it means that the ventilative cooling system reduces the cooling need of the building. If CRR is equal to 1, the ventilative cooling scenario has no cooling requirement. If CRR is zero or negative, the ventilative cooling scenario does not reduce the cooling need of the building.

CRR can also be applied on a natural ventilation scenario, calculating the cooling need by means of dynamic energy simulations in ideal loads/unlimited power mode.

In the case of mechanical ventilation systems, it is worth noting that this indicator does not take into account the energy required for air distribution. Therefore, in case of mechanical ventilation, the design decision cannot be taken regardless of the ventilative cooling effectiveness, including fan energy use in the rating.

2.6 Critical Limitations and Barriers to Ventilative Cooling

2.6.1 Impact of Global Warming on Potential for Night-Time Ventilation

Global warming with increasing temperatures will have an impact on the potential for night cooling, [13] presents developed linear regression models to estimate the daily climatic cooling potential (CCP_d) from the minimum daily air temperature, T_{min}. For eight case study locations representing different climatic zones across a North–South transect in Europe, CCP was computed for present conditions (1961–1990) using measured T_{min} data from the European Climate Assessment (ECA) database. Possible future changes in CCP were assessed for the period 2071–2100 under the Intergovernmental Panel on Climate Change (IPCC) "A2" and "B2" scenarios for future emissions of greenhouse gases and aerosols defined in the Special Report on Emission Scenarios [14].

As an example Fig. 2.7 shows for Zurich and Madrid significant changes in the percentage of nights per season when the daily cooling potential, CCP_d exceeds a certain value. For Zurich, under current climate conditions CCP_d is higher than 80 Kh (roughly necessary to discharge heat gains of 50 W/m^2, see Sect. 2.2.2) throughout most of the year, except for about 10% of summer nights. Under the "A2" scenario CCP_d was found to fall below 80 Kh in more than 50% ("B2": 45%) of summer nights.

For the studied locations in Southern Europe CCPd values under present climatic conditions were found to be below 80 Kh throughout almost the entire summer, but a considerable cooling potential was revealed in the transition seasons. For the whole year the percentage of nights when CCP_d exceeds 80 Kh in Madrid was found to decrease from 70% under present conditions to 52% under "A2" conditions, [13].

The decreases found in mean cooling potential have regionally varying implications. In Northern Europe the risk of thermal discomfort for buildings that use

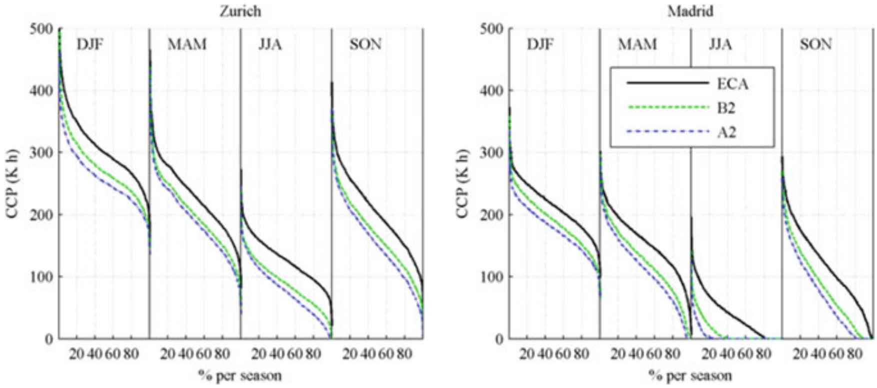

Fig. 2.7 Seasonal cumulative distributions of CCP_d in Zurich (left) and Madrid (right) for current climate (ECA) and averages for forcing scenarios "A2" and "B2" [13]

exclusively ventilative night cooling is expected to steadily increase up to possibly critical levels in the second half of the twenty-first century. In Central Europe extended periods with very low night cooling potential—where thermal comfort cannot be assured based on night-time ventilation only—could already become more frequent in the next few decades, if a strong warming scenario became real. For Southern Europe the potential for ventilative night cooling will sooner or later become negligible during summer and will decrease to critical levels in the transition seasons.

It should be noted that although cooling by night-time ventilation is expected to become increasingly ineffective during summer, it is likely to remain an attractive option in the transition seasons. This will be even more the case, if it is considered that under general warming the cooling season will tend to start earlier in spring and end later in autumn. In fact, the decreasing cooling potential and the simultaneously increasing cooling demand result in a shift of possible applications of night-time ventilation in Europe from South to North and from summer to the transition seasons.

Any assessment of possible changes in future climate is subject to large uncertainties. Nevertheless, the extent and rate of the expected climatic changes and the long service life of buildings imply the need for designing buildings capable of providing comfortable thermal conditions under more extreme climatic conditions.

2.6.2 Impact of Urban Environment (Heat Island and Reduced Natural Driving Forces)

The urban environment will have an impact on the ventilative cooling potential and also impose constraints for the use of natural driving forces. Urban environments have typically lower wind speeds, higher temperatures and higher noise and pollution levels [15].

In many cities the heat island effect with higher temperatures causes a decrease in the potential for ventilative cooling in the urban area compared to surrounding rural areas—where climate data usually originate. The CCP concept was applied to assess the implications of heat islands for night-time ventilative cooling [16]. A reduction in CCP during summer of about 9% was found for London Even larger effects were found for Adelaide, Australia (up to 26%) and Sde Boqer, Israel (up to 61%).

2.6.3 Outdoor Noise Levels

Outdoor noise levels in the urban environment can be a major barrier for application of ventilative cooling by natural driving forces and methods for estimating noise levels in urban canyons is needed to assess the potential as well as to assess the risk that occupants will close windows to keep out noise but also compromise the ventilative cooling strategy. In urban canyons the noise level increases with traffic density and decreases with height above the street at the attenuation increases with distance to the source. The attenuation decreases with increasing street width. Based on these relationships and measurements performed in 9 different urban canyons in Athens [17] developed a simple model calculating the direct as well as the reverberant noise component at a certain height above the street level. Calibration of the model with measurements showed that the noise attenuation was almost entirely a function of the street width and the height above the street. Making the assumption that traffic level is a function of the street width the noise level becomes purely a function of the street geometry.

In [17] a tolerable noise level in European offices was suggested to be around 60 dB. At the same time the noise attenuation at an open window is accepted as 10–15 dB. Thus an outdoor noise level of 70 dB or less is likely to be acceptable. Using special methods and window designs, a further 3–5 dB attenuation is possible. Figure 2.8 shows the expected noise levels in Athens at different street widths and heights above the street and the implications of this for use of the natural ventilative cooling potential at different heights above street level according the above rules of thumb.

2.6.4 Outdoor Air Pollution

Key outdoor pollutions like NO_2, SO_2, CO_2 O_3 and suspended particulate matter PM are usually measured continuously in larger urban environments and are often considered as a major barrier for application of natural ventilative cooling.

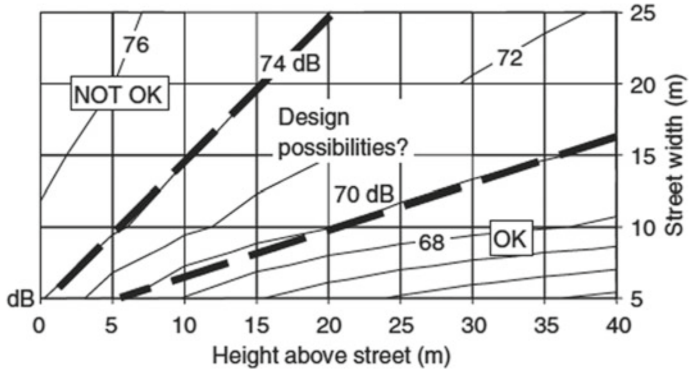

Fig. 2.8 Contours of noise level at different heights above the street and street widths. Configurations in which natural ventilation is possible are indicated (OK), as are those in which it is ruled out (NOT OK). Between these two extremes is a region in which there are possibilities for design solutions, [17]

The mean levels of SO_2 are equal outdoors and indoors, while NO_2 and O_3 reacts with the building materials resulting in a lower concentration indoors than outdoor for an airtight building. The transport of PM depends on the particle size. Estimation of the indoor/outdoor pollution ratio is the key to an assessment of the potential use of natural ventilative cooling in an urban environment.

In [15] the indoor/outdoor pollution ratio was reported in nine school buildings with different facade permeability, see Fig. 2.9.

In the experiments the indoor/outdoor (I/O) pollution ratio were studied for ozone, nitrogen dioxide and 15 sizes of PM. The ratio of indoor/outdoor concentration was

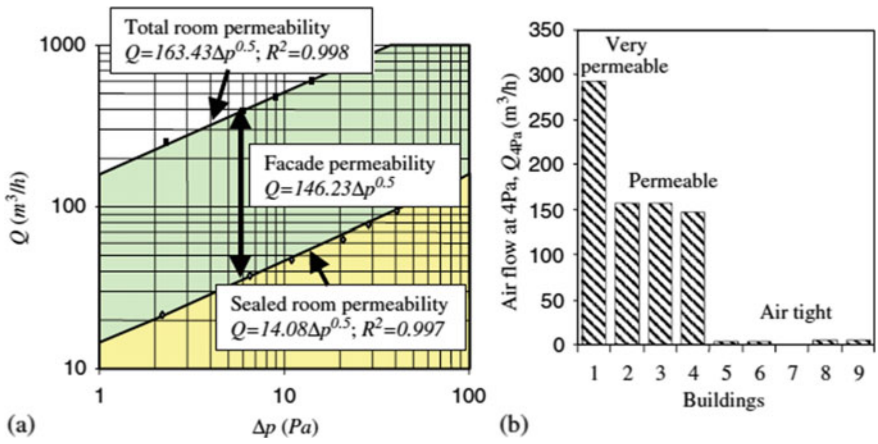

Fig. 2.9 Building permeability for nine school buildings used in experiments on indoor/outdoor pollution ratio ([15], original data from [18])

found to be a function of airflow through the façade (façade airtightness) and of the outdoor concentration. The indoor concentration was smaller inside than outside. Ozone presented the lowest I/O ratio (0.1–0.4), with higher I/O ratios measured for higher outdoor ozone concentration. The I/O ratio for nitrogen dioxide was between approximately 0 and 0.95 with lower values for higher outdoor concentration. The I/O ratio for PM depended on the particle size. The most important variation (0.25–0.70) was measured for particles of small size (0.3–0.4 mm); particles of larger size (0.8–3 mm) represented lower, but comparable, variation of the I/O ratio (0.3–0.7).

Acknowledgments The material presented in this chapter is based on the outcome of the international research project IEA EBC Annex 62 Ventilative Cooling and is the result of an international joint effort conducted in 15 countries. All those who have contributed to the project are gratefully acknowledged.

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