Ing. Ondřej NEHASIL
Ing. Daniel ADAMOVSKÝ, Ph.D.

CTU in Prague, University Centre for Energy Efficient Buildings

Ventilative Cooling Control Strategy for Variable Air Volume Ventilation Systems

Strategie řízení chlazení větracím vzduchem pro systémy s proměnným průtokem vzduchu

Reviewer
Ing. Jan ŠIROKÝ, Ph.D.

One disadvantage of highly insulated buildings is their overheating, with the subsequent necessity of removing excess heat. This is often done via mechanical cooling. However, increased energy consumption related to mechanical cooling is far from compatible with achieving zero-energy buildings.
This paper presents a detailed description of a control mechanism that can be implemented in newly designed or even existing buildings with a VAV (Variable Air Volume) ventilating system, which leads to a significant reduction in the annual energy consumption of mechanical cooling. A control strategy has been developed and validated on the multi-zone model of a school building. The results show energy savings above 40% when using ventilative cooling. The maximum efficiency was found in the range between 0.5 and 0.7 times the nominal volumetric flow rate.
Keywords: *air handlig units, control, cooling, VAV*

Jednou z nevýhod dobře zaizolovaných budov je jejich sklon k přehřívání s následnou nutností odvádění tepelné zátěže. K tomu se obvykle používá strojní chlazení. Spotřeba energie související se strojním chlazením je však v rozporu se snahou stavět budovy s téměř nulovou spotřebou energie.
Tento článek představuje podrobný popis řídícího mechanismu, který lze realizovat v nově navržených nebo dokonce existujících budovách s větracím systémem VAV (Variable Air Volume), což vede k významnému snížení roční spotřeby energie na chlazení. Byla vyvinuta a potvrzena strategie řízení chlazením větracím vzduchem na vícezónovém modelu školní budovy. Výsledky ukazují úsporu energie větší než 40% při použití chlazení větracím vzduchem na případové studii. Maximální účinnost byla zjištěna v rozmezí 0,5 až 0,7 násobku jmenovitého objemového průtoku vzduchu.
Klíčová slova: *vzduchotechnická jednotka, regulace, chlazení, VAV*

INTRODUCTION

Cooling is an important issue for contemporary buildings, and its importance is growing with the increasing quality of a building's envelope, the indoor heat gains due to the increasing amount of electronic equipment, and the tightening of the requirements on indoor environments and climatic change [1]. Cooling devices came into the market in the 1930s as rare luxury devices, which first became affordable in 1947 [1]. In 2015, they represented a rapidly developing industry sector, approaching a total annual turnover of close to 100 billion dollars, and it is still growing.

The cooling of buildings represents a considerable percentage of the total energy consumption in the world. The world annual energy consumption for cooling in 2010 was close to 1.25 PWh. More than 45% of that energy was consumed by commercial buildings. The future average cooling energy demand for commercial buildings is calculated to rise 275% by the year 2050 from the consumption in 2016 [2], despite the efforts to achieve energy efficient buildings [3].

However, unlike heating, cooling can often be provided by natural and economic systems of low-energy cooling techniques, with no compressor-based cooling. Most of those techniques are based on the alternation of day/night temperatures, the temperature differences between the outdoor and indoor air, and high temperature sources of natural coolness, such as the ground or a water mass. All these techniques have their own individual limitations, especially regarding the available power or the total amount of available coolness [4].

Ventilative cooling, a frequently discussed method for low-energy cooling techniques, requires the use of outdoor air at a lower temperature than the indoor air to cool the interior. In residential buildings, this is a convenient way to keep the indoor temperature low enough to be comfortable [5]. The potential for using outdoor air is certainly not negligible, especially in combination with night pre-cooling, at least in European climates [6], in North Africa [7] or in China [8]. Even in southern Europe or central Turkey, there are places where the mean climatic potential is more than 40 Kelvin-hours per night during July [6]. And in a very hot and humid climate, such as in Taiwan, it is still convenient to cool by using outdoor air via hybrid ventilation [9].

But natural night or hybrid ventilation also holds the risk of overcooling, which leads to discomfort for the occupants in the morning. This suggests adding a weather forecast and a building model to the BMS (Building Management System), and it has been proven that this leads to better results [10-12].

Night-time cooling is also very sensitive to the usable thermal capacity. Coolness, which is available during the night-time, is needed during the daytime. The accumulation of the coolness is possible in the building, as well as outside the building in the groundmass [13]. The thermal capacity of a building can be increased by PCM materials, which can be used not only as a plastering material, but can also be implemented directly in air-ducts to store the night-time coolness for daytime utilisation [14].

The idea of utilising the 24-hour continuous one-way operation of a ground-tube with a high accumulation ability has also been found to be particularly successful. Due to the phase shifting of the temperature peaks, this brings the night-time coolness into the interior during the day, but, due to the same phenomenon, hot air during the night as well [14]. To eliminate this disadvantage, it is necessary to add multiple dampers and an additional fan to the pipes, as well as to employ the more sophisticated operation of the ground-tubes or, alternately, to abandon the idea of utilising the groundmass heat capacity.

It has been proven that problems with ventilative cooling, such as too small a cooling capacity during the day or overcooling during the night, can be solved if the ventilative cooling is controlled and if an adaptive model of the thermal comfort is used [15]. Commercial buildings are usually equipped with mechanical ventilation to satisfy the indoor air quality requirements [16]. Under these circumstances, the utilisation of ventilation equipment for cooling would clearly be advantageous [17].

Multiple studies [5, 6, 10, 18] prove that the major issue in ventilative cooling is the fan energy consumption and optimal fan operation. To provide effective ventilative cooling, the supply-air temperature and supply-air flow rate are the primary parameters to be optimised in time in order to reach a high energy efficient VAV system [19].

Most previously published strategies deal with airside economisers in VAV systems, used for fresh air supply and heating/cooling [20-30], with no heat recovery between the fresh and exhaust air. Much less attention has been paid to heat recovery usage in VAV [31-33]. This has led to the necessity of finding a balance between the fresh air rate, the total amount of the supply-air (and its temperature) to simultaneously satisfy the requirements of thermal comfort and Indoor Air Quality (IAQ) [20] [23-26]. Some attention must also be paid to the speed of the response to regulatory intervention and the energy consumption in the transition states of VAV [22, 28, 34].

This paper deals with idea that the most effective VAV ventilation system serves the fresh air supply only, when the thermal comfort of any room is ensured by the additional systems of heating or cooling. The strategy presented develops the idea of synergy between ventilation, heating and cooling, while describing how to operate systems mainly designed for ventilation in order to assist the cooling of the ventilated spaces.

Using heat recovery exchangers and delegating the responsibility for the thermal comfort to the additional heating/cooling systems, VAV systems no longer need an economiser and operate at all times with 100% fresh air. The supply air temperatures can then be modified by altering the amounts of ventilated air by bypassing the heat recovery exchanger. A continuous control strategy for the bypass valve position and supply air volume is described in the paper.

Unlike most previously presented strategies [24, 28, 30, 33], the physical quantity regulated is not the temperature in the intake air-duct, with little regard for the needs of the space, but directly attributed to the temperature of ventilated spaces. The temperature in the inlet duct is only measured so as not to exceed certain safety and hygienic limits.

CONTROL STRATEGY

The method entails a newly developed control strategy, driving both the position of a bypass valve around the heat recovery exchanger and the air flow supply. This strategy was then tested in a case study – a TRNSYS model of an actual building located in the Netherlands.

Boundary conditions for the Ventilative cooling control

The operation is intended for an AHU with heat recovery, which also has a bypass channel around the heat recovery installed (or equivalent ability to stop or slow the rotary exchanger) to stop or slow the rotary exchanger in the VAV systems, with a high level of control – its actual critical route must be ascertained continuously. In the commercial buildings sector, VAV systems are the most energy efficient systems in use today [19].

To increase the power and efficiency of ventilative cooling, an adaptive model of thermal comfort [35] must be utilised, setting a range of comfortable indoor temperatures. The adaptive model adopted must include all contexts, such as adjusting the dress-code or allowing the occupants to change seats, as well as an individual setting of the local heating or cooling elements.

This strategy is appropriate for office buildings and public buildings such as schools, where similar temperature requirements pertain to multiple rooms.

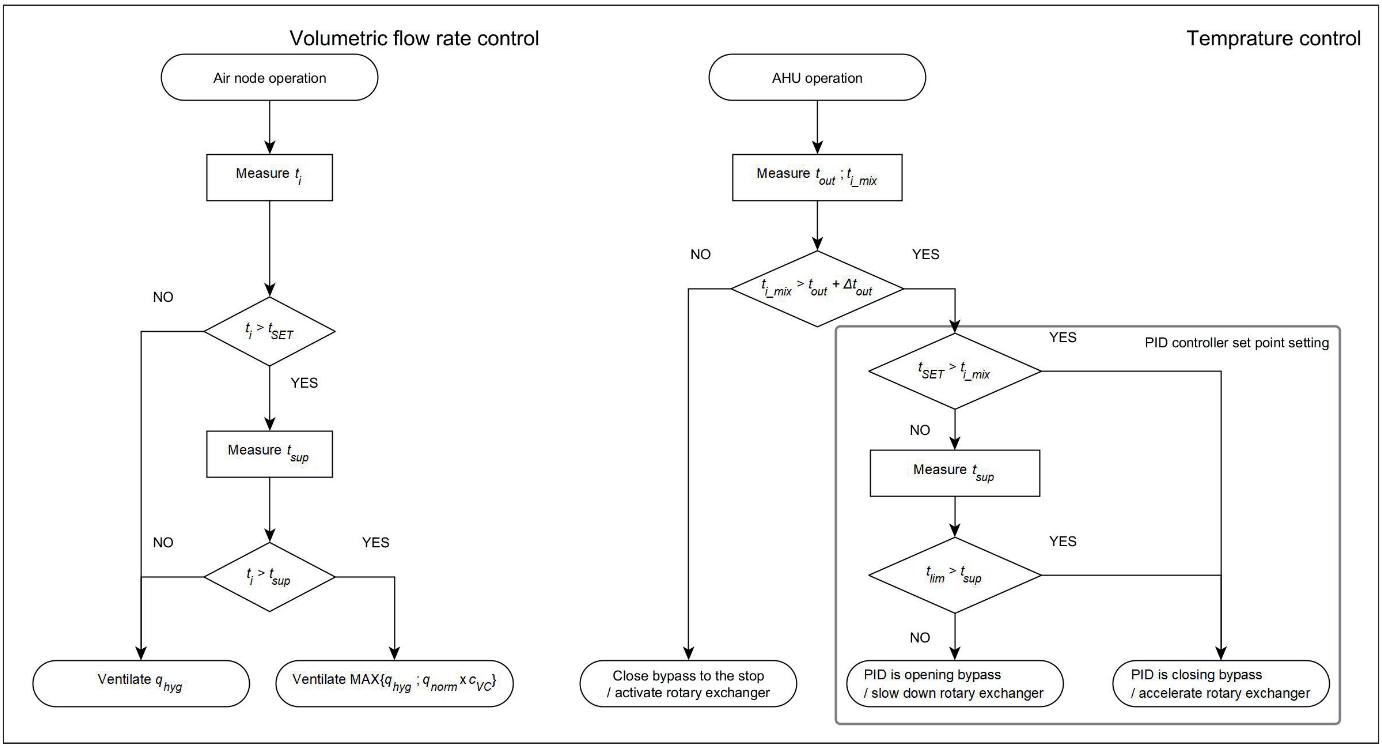

Fig. 1 Ventilative cooling operation flowchart

Operation of the Ventilative cooling control

The operation strategy is based on the concept that the temperatures and volumetric flow rates of the fresh air will be controlled by several criteria. The first of these is hygienic ventilation. Our study does not deal with the methods for setting the ideal hygienic flow rates of the fresh air supply for the occupants. This field is deeply discussed in [21, 29, 36-39].

The second criterion is the temperature in the ventilated spaces, which is connected with the information whether cooling is required.

The control mechanism is then divided in such a way that the VAV box uses a single air-node determining the volume of the ventilated air, and a single AHU part is used to regulate the temperature of the air intake, as shown in Figure 1. For most of the year in the Netherlands, it is possible to get fresh air at a lower temperature than the indoor temperatures without mechanical cooling.

Operation – AHU part

The AHU must supply fresh air to the air-ducts in order to satisfy the requirements of the VAV boxes. The AHU can regulate the temperature of the air by setting the position of the bypass valve around the heat-recovery exchanger, as shown in Figure 1, on the right side. The proportional-integral-derivative (PID) regulation must be implemented where the regulated quantity is the air temperature in the AHU's inlet duct from the ventilated spaces, where the mean temperature of all the ventilated spaces is measured. Similarly, the temperature of the ventilated zones obtained by a weighted average from local temperature sensors can be used. The set-point for this temperature is usually t_{com}, previously calculated as the set-point for the BMS. A rise in temperature in the air-duct above this set-point shows that the interior is becoming overheated, so the AHU must open the bypass to decrease the temperature at the fresh air inlet.

For the part of the year when the outdoor temperature is lower than the indoor temperature, this rule is appropriate. It can be suspended in the case where:

- ❑ the outdoor temperature t_{out} rises above the temperature measured in the AHU's inlet from the ventilated spaces, or
- ❑ the temperature at the fresh air inlet decreases to the comfort limit t_{lim}.

The temperature of the air supply to the ventilated spaces is then controlled in concert for all those spaces on the same AHU.

Operation – VAV box part

Simultaneously, each VAV box is also controlled to satisfy the individual requirements of a given space. The primary purpose of a VAV box is ventilation, so fresh air for hygienic ventilation must always be supplied. But when the temperature of the ventilated space rises above t_{com}, it is advantageous to supply a higher volume of cold air so as to extract the heat load. The ideal flow-rate for the ventilative cooling is usually not the maximal flow-rate to a given space each and every time. This is due to the fan energy consumption and the pressure drop in the air-ducts. So, when the room is occupied, the fresh air requirement may be higher than the cooling requirement. However, during a lunch break or just after working hours, the fresh air requirement drops, while it still may be convenient to ventilate because of the accumulated heat dissipation.

Also, since the temperature is set for all the spaces collectively, and since no room can remain totally unventilated, overheated rooms may cause the slight ventilative overcooling in others, as is shown in Figure 2. This phenomenon can be mitigated by the design of the ventilation system – – a small number of similar rooms connected to the same AHU behave better than a large number of variably sized spaces.

Fan speed control

The energy for the fans is almost the only energy required in ventilative cooling. The consumption of the controllers, motor dampers and sensors are negligible [41] (available in EN as [42]). Due to the low temperature differences between the fresh air and ventilated space, and also due to the low thermal capacity of the air, a high flow rate is required to ensure sufficient cooling power. But the transport of the air is highly energy intensive, and it is necessary to focus on the fans in ventilative cooling applications. Fan energy consumption is given as a multiplication of the volumetric flow rate and the pressure drop. For economical operation, both the pressure drop and the volumetric flow rate must be decreased in order to reduce the fan energy consumption under partial load to a minimum.

Ventilative cooling uses the Static Pressure Reset (SPR) principle based on [43], as is described below. Each VAV box must be connected to the BMS and must measure the air flow through itself (e.g., by the pressure drop on the distribution element), adjusting the damper position to reach the required flow rate, as is shown in Figure 3. Each air-node indicates its state by two logical outputs:

- ❑ OK, to indicate that the required flow rate has been reached
- ❑ FULLY to indicate that the damper has been totally opened

These signals are collected by the controller of the AHU, which adjusts the fan speed by means of the inverter. When a damper is fully open, but does not signal OK, it means that the static pressure is not high

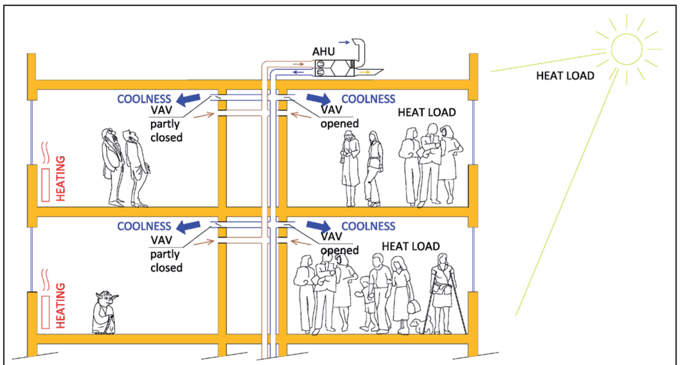

Fig. 2 Cross section of a group of rooms cooled by the ventilation [40].

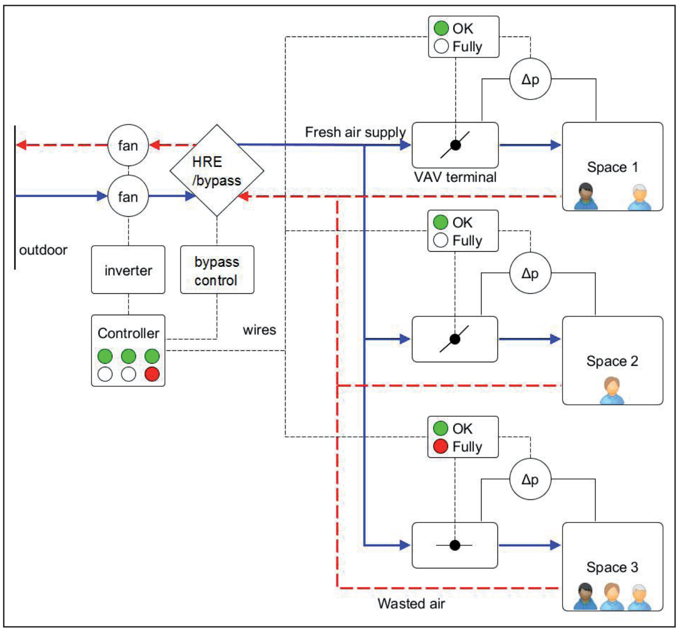

Fig. 3 SPR control, every time, at least one damper is fully opened.

enough to ventilate the requested air flow rate and, therefore, the static pressure set-point will be increased by a certain value, e.g., 5 Pa. When none of the terminals are fully opened, the static pressure set-point will be decreased by the same value, pushing the VAV terminals to open their valves.

This ensures that, at all times, at least one damper is fully opened, so that the dampers on the actual critical route will be opened. The number of opened dampers or dissatisfied terminals must be set higher than 1, for the sake of faults or improper settings in the VAV terminals [44]. Such faults or improper settings appear quite often, and a site survey indicated that 20.1% of the VAV terminals were found to be ineffective [45]. It is highly recommended that the entire system's operation should be assessed continuously, e.g., by AHU's performance assessment rules (APAR) [46].

From the perspective of practice, it is recommended to design the SFP in accordance with the recommendation of P.G. Schild and M. Mysen [47] (available in EN as [42]) or even lower.

Coefficients and thresholds

To set up all the control mechanisms, several constants must be set.

Ventilative cooling is practical only if the outdoor temperature is lower than the indoor temperature. However, the temperature of the air rises slightly when it passes through the fan. A threshold is necessary to switch the ventilative cooling on while condition (1) is met:

$$\pounds_{i\_mix} \aleph_{out} \quad {}_{out} \tag{1}$$

The difference Δt_{out} must be set between 0 K and 3 K, with respect to the efficiency of the ventilation system. A crucial aspect for ventilative cooling is the volumetric flow rate, which is represented by the coefficient c_{VC}. At the VAV terminal, if there is a cooling request, set the higher of the two volumes:

$$q_V = MAX \begin{Bmatrix} q_{hyg} \\ q_{norm} \cdot c_{VC} \end{Bmatrix} \tag{2}$$

It was found to be ineffective to provide ventilative cooling at the maximal design speed ($c_{VC} = 1$), because of the rise in the air temperature at the fan and the higher energy consumption, which decrease the overall system efficiency. Reducing the volume of the ventilated air increases the system efficiency while, at the same time, it decreases the total cooling power. Ventilative cooling can then be very useful after the working day, for example, to pre-cool the offices for the next day. The best value for c_{VC} depends on the nominal *SFP*, the control mechanism, the nominal *ACH* of the space, as well as on the *EER* of the additional cooling or thermal capacity of the building. To find the best value, a set of annual building energy simulations can be undertaken, analysing function (3) to seek its global minimum.

$$f\left(c_{VC}\right) = Q_{C\_an} + FEC_{an} \cdot EER \tag{3}$$

In addition, the set-point temperature t_{set} for the operation must be specified. The comfort temperature must be determined as in (4) [48]:

$$t_{com} = 0.09 \cdot t_{rm} + 22.6 \tag{4}$$

where t_{rm}, depending on the outdoor temperature history, is expressed iteratively as in (5) [48]:

$$t_{rm} = 0.2 \cdot t_{out_{yesterday}} + 0.8 \cdot t_{rm\_yesterday} \tag{5}$$

Consider that the occupants can adapt to higher temperatures if the weather remains hot for a longer period [35, 48]. The range t_{com} ±2 K is considered the comfort range, and the interior temperature should be kept within this range. It is recommended that the ventilation set-point be specified as t_{com} (6) for the majority of the time. Then the cooling set-point can be t_{com} + 2 K (6), while the difference of 2 K is the area where ventilative cooling can be used.

Unfortunately, two Kelvins is a very small difference, which does not allow higher pre-cooling of the interior. But the comfort range is also specified under the comfort temperature, down to t_{com} – 2 K. During the cooling season, when a temperature rise over a day is expected, the ventilation set-point can be modified to t_{com} – 2 K at a time when the building is not used (6), so as to prepare it for the following day. The space will then begin in the morning with a boundary comfort temperature t_{com} – 2 K, which will then rise continuously during the entire

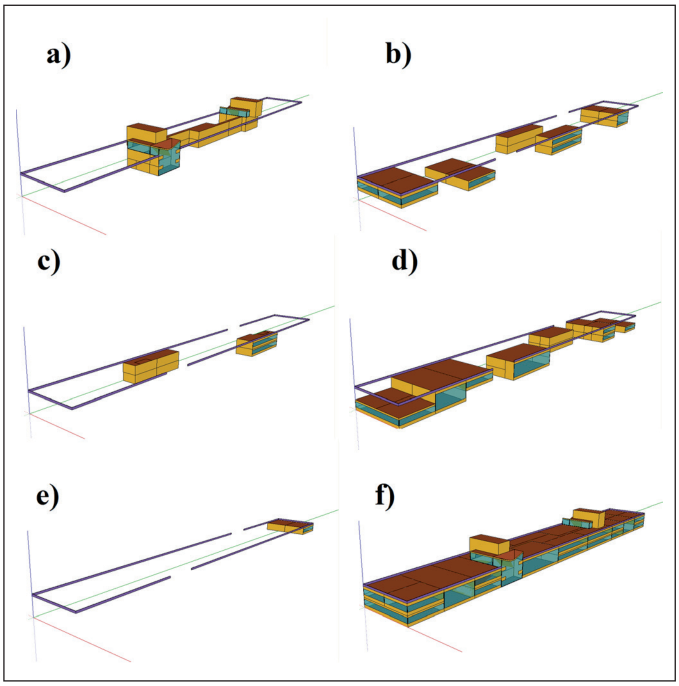

Fig. 4 Different types of air-nodes – a) corridors; b) classrooms; c) offices; d) open spaces; e) labs; f) whole model

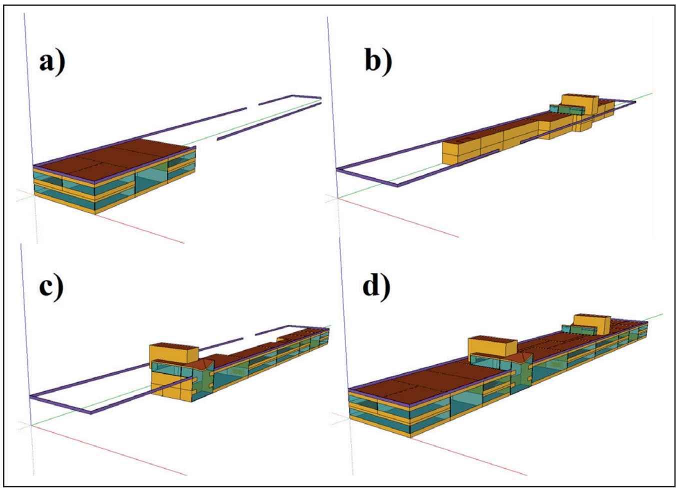

Fig. 5 Ventilated zones. a) South zone; b) West zone; c) East zone; d) whole model

day, up to the threshold for the mechanical cooling t_{com} + 2 K, which ameliorates the rise in this temperature.

$t_{set} = t_{com}$ Normally, during winter and the workday
$t_{set} = t_{com} - 2K$ In the cooling season, outside working hours
$t_{cool} = t_{com} + 2K$ Mechanical cooling set-point, valid for the entire year. (6)

To avoid discomfort for the occupants, a maximal allowed temperature difference between the air intake and room temperature should be defined. This is done by the variable t_{lim}, as a lower limit of the supply temperature t_{sup}. The recommended temperature difference between the room temperature and air supply temperature Δt_{i_s} is between 3 K and 10 K, depending on the method of air distribution. More mixed air distribution, such as swirl diffusers, allows a lower temperature at the air inlet. This limit is not justified after the working hours when the occupants are not present. The temperature limit of the supply air can be set as:

$t_{lim} = t_{i\_mix} - \Delta t_{i\_s}$ during the working hours
$t_{lim} = 5\ °C$ outside the working hours (7)

The limit can be additionally corrected if there is the risk of condensation on the air-ducts.

CASE STUDY

To verify the designed strategy and find the appropriate values for the coefficients and thresholds, a set of building simulations was performed. The ventilation strategy was implemented on a building, based on an existing school facility, as modelled in the TRNSYS simulation studio [49]. The building model was adopted from an evaporative cooling study made in 2014 [40].

Building model description

The building model represents the last two floors of the main building of the TU/e Eindhoven and is situated in the moderate maritime climate of the Netherlands. It contains 35 air-nodes on two floors, with different purposes and use profiles (see Figure 4), different volumes and heights, and different orientations. The air-nodes are grouped into three AHU's, which ventilate the group of neighbouring air-nodes, as is shown in Figure 5.

The building is considered modern, after refurbishment to Dutch standard qualities. This means that the lightweight building envelope has a *U-value* of 0.22 W/m²K of opaque walls, and 0.75 W/m²K of triple glazed windows with a *g-value* of 0.613. The roof has a *U-value* 0.106 W/m²K. The building is constructed from a concrete skeleton, where the concrete slab is on every second floor, while the remainder of the construction is lightweight, which easily allows the creation of spaces with a height of two floors. Therefore, the thermal accumulation ability of the building is limited.

Air handling units were simulated in detail. The SFP in the design speed is calculated according to [47] as 1 kW/(m³/s), which fulfils the requirements of [50] for 2018 and later. The fan motor works in the ventilated air flow, and all energy inputs are considered to transfer into the air, thereby increasing its temperature.

The design volumetric flow rate for each air-node is specified by the *ACH* setting, to avoid misguided values related to the occupancy or heat gains. The number of maximal air changes per hour is set as 0.5 h^{-1} for the stairs and corridors, 1.5 h^{-1} for the offices and labs, and 2.5 h^{-1} for the classrooms. When a room is not occupied, the air change decreases to 0.1 h^{-1}, but never entirely stops.

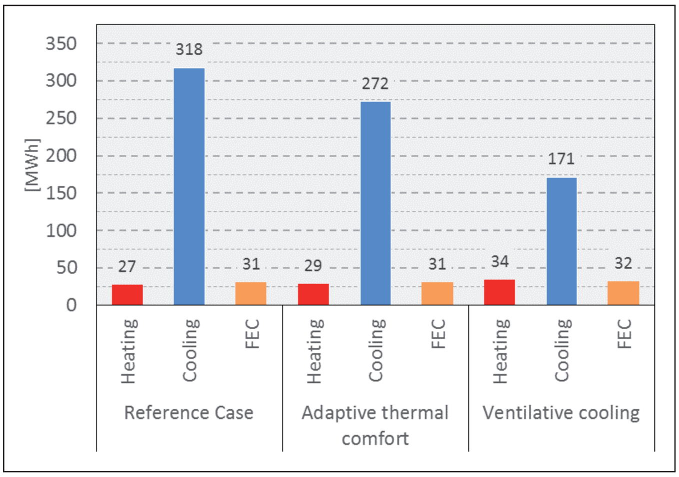

Fig. 6 Annual energy demand for three cases

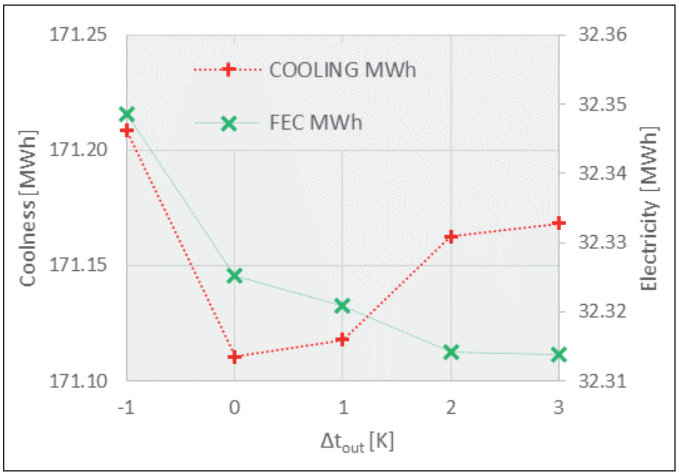

Fig. 7 Influence of Δt_{out} on the fan energy consumption and cooling demand

The heat recovery exchanger changes its efficiency in relation to the various temperature differences and volumetric flow rates of the air. The Number of Transfer Units (NTU) method is used for the heat recovery calculation [51]. For the bypass damper position regulation, a PID Type-23 is used. Comparative decisions are made by a Type-2, where hysteresis or thresholds are also implemented.

All simulations were on an annual basis and made with a 5-minute time-step, so as to capture the influence of the controllers, which switches the fans or dampers at the moment when the set-point or threshold on continuously changing (interpolated) temperatures was reached.

The ventilation control cases followed

As a **reference case**, no special approach was used. The ventilation serves as the fresh air supply only, with heat recovery during the winter and through the bypass during the cooling season. The heating set-point is 20 °C and the cooling set-point is 24 °C – which is normal in Dutch office buildings. All cooling loads are discharged by additional cooling equipment.

An adaptive thermal comfort case was considered so as to separate the influence of the adaptive model of the thermal comfort from the influence of ventilative cooling. In this case, if other conditions for this model are met (see [41]), the cooling set-point may be a floating variable that is dependent on the outdoor temperature (4). This can lead to cooling energy savings by itself, simply because of the lower temperature differences between the indoors and outdoors.

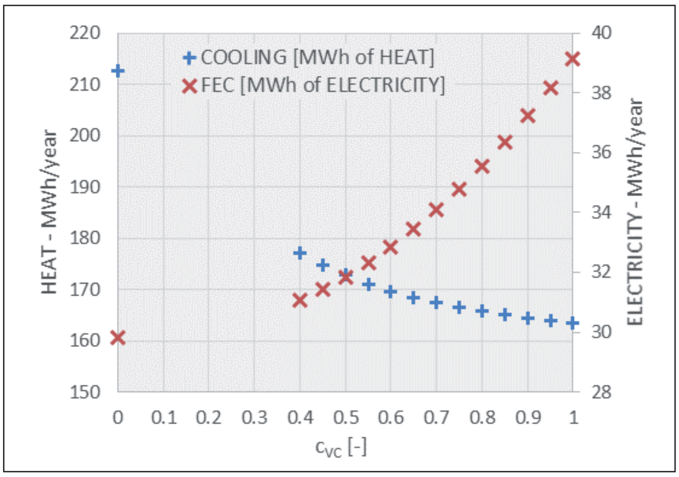

Fig. 8 Influence of the cVC to the fan energy consumption and cooling demand

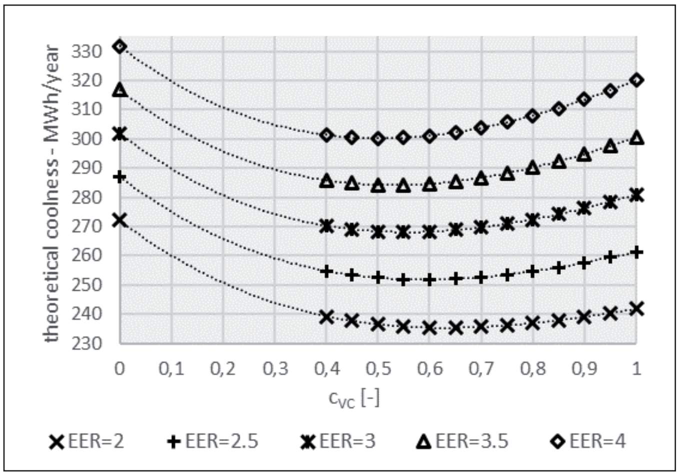

Fig. 9 The theoretical coolness, calculated accordingly (3).

The ventilative cooling case is the last one, and here all the findings and principles mentioned in the „Building Model Description" section above are included.

RESULTS

The results of the annual simulation, after the coefficients and thresholds have been found, are given in Figure 6. In the reference case, the simulation shows a significant imbalance between the heating and cooling, which was expected. The annual cooling demand is 10 times higher than the annual heating demand. Using the adaptive model of thermal comfort, 14% of the cooling demand can be saved. Ventilative cooling can save another 32% of the initial cooling demand, so the overall savings are 46% of the coolness, which means 147 MWh in the case study.

The use of ventilative cooling also brings a slight increase in the heating demand, because of the unavoidable overcooling in some spaces, as is shown in Figure 6. This additional heating demand is 25% in total, while the initial heating demand in absolute numbers was very low: the rise accounts for 7 MWh of heat. The building is heated mostly by the fans, the equipment, the occupants and the sun. Of this 7 MWh additional heating demand, 2 MWh is caused by an increase in the indoor temperature by the adaptive model of the thermal comfort and the remaining 5 MWh by compensation of the overcooling related to the cooling via the ventilation. This fraction can be reduced if the building was divided into more independently ventilated zones.

Another set of simulations has been provided to prove the sensitivity of the results to a change in the coefficients Δt_{out} and c_{VC}. This is shown in Figure 7. The fan energy consumption increases with a lower Δt_{out}, due to the additional operating hours. Simultaneously, the cooling demand decreases until Δt_{out} reaches zero. When Δt_{out} drops under zero, the ventilation brings an extra thermal load, and the cooling demand increases. In any event, the influence of the total amount of energy or coolness required is very small and only causes the fans to switch on or off a few minutes earlier or later. The sensitivity results of Δt_{out} was found to be negligible.

A marked effect with the same parameters was found in the c_{VC} coefficient, which has a major influence on the fan speed, as shown in the graph in Figure 8. With an increasing c_{VC}, the cooling demand decreases, while the fan energy consumption increases. When coefficient c_{VC} is set to zero, no additional air is ventilated other than that from the fresh air for the occupants. The cooling demand in this case is only 212 MWh whereas, without the ventilative cooling in the Adaptive Thermal Comfort case, it was calculated to be 272 MWh.

The difference relates to the bypass damper operation in the AHU. Increasing the c_{VC} leads to even more positive results. The energy demands in Figure 8 cannot be compared, because of the incomparability of the electricity and coolness. To find the optimum coefficient of the ventilative cooling, energies must be converted to a common base. This can be expressed as a theoretical coolness, expressed above as $f(c_{vc})$ (3) as the sum of the coolness actually consumed and the coolness theoretically generated by the extra electricity used for the fans under the actual conditions.

In Figure 9, the theoretical coolness is shown in relation to the *EER* of the additional cooling. The higher the *EER*, the lower the appropriate c_{VC}. If a low-efficiency cooler is used, with an $EER = 2$, then it is appropriate to ventilate more air and set the c_{VC} to a higher value of around *0.7*, as the function of theoretical coolness has a global minimum there. If a high-efficiency cooler is used, with an $EER = 4$, the ideal c_{VC} is around 0.5, ventilating less air. For all the usual ranges for machinery cooling, the preferred values of the c_{VC} are between 0.5 and 0.7.

It is important to note that this dependency is affected by the design *SFP* (equal to 1 kW/(m^3/s) in our case) and the design air change rates in each space. The lower the design *SFP*, the higher the optimal c_{VC}.

DISCUSSION

Cooling by outdoor air has great potential for energy savings, especially for highly insulated buildings in cold or mild climates. The simulations indicate that the energy savings potential, measured by the saved coolness, extra heat and extra electricity for fans, is still around 40% of the cooling energy demand when the presented operation strategy is used. No special equipment is necessary to provide ventilative cooling, other than an ordinary ventilation system with VAV, which is generally used to ensure hygienic air change.

Its entire potential is hidden in the control, and in the regulation of the bypass dampers around the heat recovery exchanger. Even better results might well be obtained using a PID-MPC, but these Model Predictive Controllers are far harder to implement [52]. Setting t_{set} within a range of $t_{com} \pm 2K$ is a future task of weather-predictive regulation which has, in the past, proven to be a powerful tool for energy savings [10-12].

The air change rate has a major effect on the performance of the entire system. The coefficient of ventilative cooling, which determines the power of the ventilative cooling, deserves special attention. Due to the fan energy consumption, it is not appropriate to ventilate with the maximum fan speed. Decreasing the volume of the ventilated air to 50% – 70% of the nominal leads to a significant increase in the overall system efficiency.

This phenomenon should be investigated more closely in the future, so that the relationship between the building mass, the specific fan power, the *EER* of the auxiliary cooling and the coefficient of ventilative cooling can be determined.

On the other hand, the results are not fully susceptible to the threshold of the temperature difference between the outdoor and indoor temperatures. Outdoor temperatures change continuously, and the threshold setting determines whether the fan switches on a few minutes earlier or later. But the overall effect of this on the entire question is negligible.

CONCLUSION

We found that ventilative cooling is an excellent way to reduce the cooling loads and energy consumption related to the cooling of buildings equipped with VAV ventilation systems. Compared to a building cooled only by a mechanical chiller, savings of around 40% of cooling energy consumption can be reached.

Current trends leading to increased indoor climate requirements in a growing number of buildings result in an overall increase of building energy consumption. Ventilative cooling, using the presented operation strategy, can mitigate the impact of higher environmental requirements, and therefore save energy. At the same time, fan-driven controlled ventilative cooling has no adverse effects on the quality of the indoor environment, as is often seen in night-cooling via natural ventilation.

The strategy presented appears to be a strong tool that can be helpful in achieving the goal of nearly zero-energy buildings.

Contact: ondrej.nehasil@fsv.cvut.cz

ACKNOWLEDGEMENT. This work has been supported by the Ministry of Education, Youth and Sports within National Sustainability Programme I (NPU I), project No. LO1605 - University Centre for Energy Efficient Buildings – Sustainability Phase.

Literature

[1] LESTER, P. History of Air Conditioning. 2015 [online]. Accessible from: https://energy.gov/articles/history-air-conditioning
[2] SANTAMOURIS, M. Cooling the buildings – past, present and future. *Energy and Buildings.* 2016, vol. 128, pp. 617-638.
[3] DIRKS, J. A., et al. Impacts of climate change on energy consumption and peak demand in buildings: A detailed regional approach. *Energy.* 2015, vol. 79, (C), pp. 20-32.
[4] SANTAMOURIS, M., ASIMAKOPOULOS, D. Passive Cooling of Buildings. (1st pub. ed.) London: James & James, 1996.
[5] FOLDJERG, P., THORBJØRN, A., HOLZER, P. Ventilative Cooling of Residential Buildings – Strategies, Measurement Results and Lessons Learned from Three Active Houses in Austria, Germany and Denmark. *International Journal of Ventilation.* 2014. Vol. 13 No. 2.
[6] ARTMANN, N., MANZ, H., HEISELBERG, P. Climatic potential for passive cooling of buildings by night-time ventilation in Europe. *Appl. Energy.* 2007. Vol. 84, no. 2, pp. 187–201.
[7] CHIESA, G., GROSSO, M. Geo-climatic applicability of natural ventilative cooling in the Mediterranean area. *Energy Build.* 2015. Vol. 107, pp. 376–391.
[8] X. ZHU, Y. LIU and J. LIU. Night ventilation research of office buildings: Part 2 Cooling potential of continental climate zone in China. In: 2011, DOI: 10.4028//www.scientific.net/AMR.250-253.1066.
[9] K. T. HUANG, R. L. HWANG. Parametric study on energy and thermal performance of school buildings with natural ventilation, hybrid ventilation and air conditioning. *Indoor and built environment.* 2016. Vol. 25 (7) , pp. 1148-1162.
[10] MEDVED, S., BABNIK, M., VIDRIH, B., ARKAR, C. Parametric study on the advantages of weather-predicted control algorithm of free cooling ventilation system. *Energy.* 2014, Vol. 73, pp. 80–87.
[11] ŠIROKÝ, J., OLDEWURTEL, F., CIGLER, J., PRÍVARA, S. Experimental analysis of model predictive control for an energy efficient building heating system. *Appl. Energy.* 2011, Vol. 88, no. 9, pp. 3079–3087.
[12] HUANG, G. Model predictive control of VAV zone thermal systems concerning bi-linearity and gain nonlinearity. *Control Engineering Practice.* 2011, Vol. 19, (7), pp. 700-710.
[13] CAMPANICO, H., HOLLMULLER, P., SOARES, P. M. Assessing energy savings in cooling demand of buildings using passive cooling systems based on ventilation. *Appl. Energy.* 2014, Vol. 134, pp. 426–438.
[14] ALIZADEH, M., SADRAMELI, S. M. Development of free cooling based ventilation technology for buildings: Thermal energy storage (TES) unit, performance enhancement techniques and design considerations – A review. *Renew. Sustain. Energy Rev.* 2016, Vol. 58, pp. 619–645.
[15] ENGELMANN, P., KALZ, D., SALVALAI, G. Cooling concepts for non-residential buildings: A comparison of cooling concepts in different climate zones. *Energy Build.* 2014, Vol. 82, pp. 447–456.
[16] THAM, K. Indoor air quality and its effects on humans-A review of challenges and developments in the last 30 years. *Energy and Buildings.* 2016, Vol. 130, pp. 637-650.
[17] CAO, X., DAI, X., LIU, J. Building energy-consumption status worldwide and the state-of-the-art technologies for zero-energy buildings during the past decade. *Energy and Buildings.* 2016, Vol. 128, pp. 198-213.
[18] ALONSO, M. J., MATHISEN, H. M., COLLINS, R. Ventilative cooling as a solution for highly insulated buildings in cold climate. *Energy Procedia.* 2015, Vol. 78, pp. 3013–3018.
[19] OKOCHI G., YAO, Y. A review of recent developments and technological advancements of variable-air-volume (VAV) air-conditioning systems. *Renewable & Sustainable Energy Reviews.* 2016, Vol. 59, pp. 784-817.
[20] ZHANG, B. et al. Demand control ventilation: Influence of terminal box minimum airflow setting on system energy use. *Energy and Buildings.* 2014, vol. 79, pp. 173-183.
[21] LIN X., LAU, J. Demand controlled ventilation for multiple zone HVAC systems: CO_2-based dynamic reset (RP 1547). *HVAC R Res.* 2014, Vol. 20, no. 8, pp. 875–888.
[22] MORADI, H., SAFFAR-AVVAL, M., BAKHTIARI-NEJAD, F. Nonlinear multivariable control and performance analysis of an air-handling unit. *Energy Build.* 2011, Vol. 43, no. 4, pp. 805–813.
[23] YANG, X. et al. Evaluation of four control strategies for building VAV air-conditioning systems. *Energy and Buildings.* 2011, Vol. 43, (2), pp. 414-422.
[24] MOSOLLY, M., GHALI, K., GHADDAR, N. Optimal control strategy for a multi-zone air conditioning system using a genetic algorithm. *Energy.* 2009, Vol. 34, (1), pp. 58-66.
[25] XU, X., WANG, S., SUN, Z., XIAO, F. A model-based optimal ventilation control strategy of multi-zone VAV air-conditioning systems. *Appl. Therm. Eng.* 2009, Vol. 29, no. 1, pp. 91–104.
[26] YUAN, S., PEREZ, R. Multiple-zone ventilation and temperature control of a single-duct VAV system using model predictive strategy. *Energy Build.* 2006, Vol. 38, no. 10, pp. 1248–1261.
[27] WANG, S., QIN, J. Sensor fault detection and validation of VAV terminals in air conditioning systems. *Energy Convers. Manag.* 2005, Vol. 46, no. 15–16, pp. 2482–2500.
[28] WANG, S., XU, X. A robust control strategy for combining DCV control with economizer control. *Energy Convers. Manag.* 2002, Vol. 43, no. 18, pp. 2569–2588.
[29] WANG, S., JIN., X. CO_2 –Based Occupancy Detection for On-Line Outdoor Air Flow Control. *Indoor Built Environment.* 1998, Vol. 7, pp. 165–181.
[30] KE, Y. P., MUMMA, S. A., Optimized supply-air temperature (SAT) in variable-air-volume (VAV) systems. *Energy.* 1997, Vol. 22, no. 6, pp. 601–614.
[31] SCOFIELD, M., PERIANNAN, V. *VAV System Heat Recovery Economizer to Furnish Free Humidification and Exceed Standard 62.1 Ventilation Requirements in Winter.* ASHRAE Transactions, vol. 121, pp. 1lll, 2015.

[32] WANG, L., CURCIJA, D., BRESHEARS, J. The energy saving potentials of zone-level membrane-based enthalpy recovery ventilators for VAV systems in commercial buildings. *Energy Build.* 2015, Vol. 109, pp. 47–52.

[33] ENGDAHL, F., JOHANSSON, D. Optimal supply air temperature with respect to energy use in a variable air volume system. *Energy Build.* 2004, Vol. 36, no. 3, pp. 205–218.

[34] HUANG, G., JORDÁN, F. Model-based robust temperature control for VAV air-conditioning system. *HVAC R Res.* 2012, vol. 18, no. 3, pp. 432–445.

[35] McCARTNEY, K. J., FERGUS, N. J. Developing an adaptive control algorithm for Europe. *Energy Build.* 2002, Vol. 34, pp. 623–635.

[36] ZHUAN, R. N., LI, X., TU, J. An energy saving ventilation strategy for short-term occupied rooms based on the time-dependent concentration of CO_2. Int. *Journal of Ventilation.* 2015, vol. 14, no. 1, pp. 39–52.

[37] OLDEWURTEL, F., STURZENEGGER, D., MORARI, M. Importance of occupancy information for building climate control. *Appl. Energy.* 2013, Vol. 101, pp. 521–532.

[38] WANG, Y., SHAO, Y., KARGEL, C. Demand controlled ventilation strategies for high indoor air quality and low heating energy demand. In: *2012 IEEE Int. Instrum. Meas. Technol. Conf., no. Dcv*. pp. 870–875, 2012.

[39] FISK, W. J, De ALMEIDA, A. T. Sensor-based demand-controlled ventilation: a review. *Energy & Buildings.* 1998, Vol. 29, (1), pp. 35-45.

[40] NEHASIL, O., ADAMOVSKÝ, D. Case study of indirect evaporative cooling in the main building of TU/e Eindhoven. In: *Simulace budov a techniky prostředí 2014*. Praha, 2014.

[41] SCHILD, P.G., MYSEN, M. Recommendations on specific fan power and fan system efficiency – Part I [online]. 2014. Dostupné z: http://vetrani.tzb-info.cz/uspory-energie-vetrani-klimatizace/11079-doporuceni-pro-merny-prikon-ventilatoru-sfp-a-ucinnost-vzduchotechnickych-systemu-i

[42] SCHILD, P.G., MYSEN, M. *Recommendations on specific fan power and fan system efficiency.* AIVC Technical Note 65. 42 pp, 2009.

[43] WARREN, M., NORFOLD, L.K. Integrating VAV zone requirements with supply fan operation. *ASHRAE Journal.* 1993, Vol. 35(4), pp. 43–46.

[44] SHAN, K. et al. In-situ validation of a fault tolerant control strategy for VAV systems. *Applied Thermal Engineering.* 2015, Vol. 87, pp. 362-370.

[45] QIN, J., WANG, S. A fault detection and diagnosis strategy of VAV air-conditioning systems for improved energy and control performances. Energy Build. Vol. 37, no. 10, pp. 1035–1048, 2005.

[46] SCHEIN, J. et al, A rule-based fault detection method for air handling units. *Energy & Buildings.* 2006, Vol. 38, (12), pp. 1485-1492.

[47] SCHILD, P.G., MYSEN, M. Recommendations on specific fan power and fan system efficiency – Part II [online]. 2014. Dostupné z: http://vetrani.tzb-info.cz/vzduchotechnicka-zarizeni/11102-doporuceni-pro-merny-prikon-ventilatoru-sfp-a-ucinnost-vzduchotechnickych-systemu-ii

[48] LAIN, M. *Nízkoenergetické Chlazení Budov.* Praha, 2007 Disertační Práce. České vysoké učení technické v Praze.

[49] Thermal Energy System Specialists, LLC. TRNSYS Simulation Studio 17.01.0016 [software]. NOV 2009. Available: www.trnsys.de. System requirements: Windows 2000 / ME / Windows XP / Vista or later, 200 MB disc space, 128 MB RAM

[50] The European Commission, Commission Regulation (EU) No. 1253/2014 of 7 July 2014.

[51] KAYS, W. M., LONDON, A. L. *Compact Heat Exchangers.* (Repr. 1998 with corrections. ed.) Malabar: Krieger, 1998.

[52] WANG, Y., KUCKELKORN, J., LIU, Y. A state of art review on methodologies for control strategies in low energy buildings in the period from 2006 to 2016. *Energy Build.* 2017, Vol. 147, pp. 27–40.

Nomenclature:

t_{com} comfort temperature [°C]
t_{out} outdoor temperature [°C]
t_{rm} outdoor running mean temperature [°C]
t_i indoor temperature of an air-node (room) [°C]
t_{i_mix} mean indoor temperature, measured in waste-air duct before the AHU (air handling unit) [°C]
t_{sup} temperature of the supply air [°C]
t_{lim} lower temperature limit of the supply air [°C]
t_{set} set-point room temperature for the ventilation [°C]
t_{cool} set-point room temperature for the mechanical cooling [°C]
Δt_{out} threshold temperature difference between the outdoor and indoor temperatures [K]
Δt_{i_s} temperature difference between the indoor and air supply temperature [K]
c_{VC} coefficient of the ventilative cooling [-]
q_V air-node volumetric flow rate [m^3/h]
q_{norm} air-node design volumetric flow rate [m^3/h]
q_{hyg} hygienic fresh air supply [m^3/h]
Q_{C_an} annual cooling demand of a selected building, non-ventilative [kWh]
FEC_{an} annual energy consumption of the fans [kWh]
SFP specific fan power [kW/(m^3/s)]
EER energy efficiency ratio of the cooling device [-]
ACH air change per-hour ratio [-]