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Control logic for a novel HVAC system providing room-based indoor climate control in residential buildings

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ABSTRACT

One of the major changes regarding HVAC systems in new or renovated residential buildings is the incorporation of heating in the ventilation system, providing an air heating system. In buildings equipped with air heating systems, indoor climate is commonly controlled based on a reference zone, which leads to unsatisfactory thermal condition in the other zones. In addition, this strategy is not energy-efficient for buildings due to overheating that can occur in some zones. This study presents control logic for a novel-designed demand-controlled ventilation and air heating system and develops a variable air volume (VAV) control strategy to ensure an energy-efficient operation of the system, while providing a satisfactory indoor climate conditions. The air heating system can change airflow rate and air temperature in each room quickly and accurately enabling the control of indoor climate on a room level within a dwelling. The performance of the air heating system, in terms of its effect on thermal comfort and indoor air quality in each room as well as the stability of the system has been evaluated through a full-scale laboratory experiment. The experimental result indicated the ability of the system to satisfy different temperature set-points ranging from 20 °C to 25 °C in different zones with ±0.3°C deviation from the set-points, while the system was remained stable when temperature set-points were changed in several zones during the experiment. Individual indoor air quality requirements were also satisfied with providing the required supply airflow rate to each zone, while the supply fan was running based on the critical zone reset control strategy to achieve an energy-efficient operation.

1. Introduction

Heating, ventilation and air conditioning (HVAC) systems are used in commercial, residential and industrial buildings to maintain a controlled indoor climate, i.e. indoor thermal condition and indoor air quality. Depending on the heat transfer medium used for heat transfer, HVAC systems are classified into three main categories: All-air, All-water and Air-water systems [1]. Among these categories, All-air system in which heating and/or cooling are provided via the conditioned air becomes an economic solution especially in buildings with low thermal energy demand. In these buildings, thermal comfort satisfaction can be achieved with rather low airflow rate, close to the amount of airflow rate needed for indoor air quality satisfaction [2,3]. This eliminates the

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need for separate heating and/or cooling systems, thereby reducing the investment and operating costs. Particularly, in highly-airtight new or energy retrofitted residential buildings, natural ventilation system may not be able to satisfy the indoor air quality requirements enforced in building regulations [4,5]. The application of mechanical ventilation systems seems inevitable in these residential buildings. Hence, there is a great potential in incorporating heating in the ventilation system providing a combined ventilation and heating system as an energy efficient HVAC solution. An example is Passive House buildings, where the reduction of heating demand allows the integration of heating in the ventilation system, called air heating system [6,7].

There are two main control strategies for air temperature regulation in air heating systems leading to two types of system, Constant Air Volume (CAV) systems and Variable Air Volume (VAV) systems. CAV systems supply air with variable temperature at constant airflow rate while VAV systems supply constant air temperature with variable airflow rate in order to regulate the room temperature [8]. Multi-zone CAV systems are commonly controlled based on temperature measurement in a reference zone or the main extract duct with central regulation of supply air temperature. This strategy is simple to implement and economic compared to VAV systems, but cannot satisfy different room temperature requirements in all zones [9]. On the other hand, the need for different room air temperature within a residential building, e.g. a colder bedroom, is rising. For instance, the research studies done by Berge et al. [10,11] indicate the desire for lower temperature in bedrooms in comparison to other living areas in high-performance residential building with high insulation and mechanical ventilation system. In [12], Berge et al. used IDA ICE to simulate four cases with different reference zone temperatures for controlling a CAV ventilation system. Three of the cases are one-zone with a reference temperature either in the supply air temperature or in the bedroom and one of the case is two-zone with reference temperature both in the bedroom and the living room. They conclude the clear potential of the two-zone case in satisfying thermal comfort requirement in the bedrooms and reduction of heating demand, while the one-zone solutions have clear limitation in this regard. The two-zone case is modeled with two air handling units in IDA ICE. In another study in [13], decentralized ventilation with two separate air handling units has identified as the best solution to enable thermal zoning with two isothermal zones in a detached single family house.

An alternative solution to achieve accurate temperature control in a multi-zone CAV system is the use of local reheat coils installed in each zone downstream of a single central CAV system. The local reheat coils can be either hot water coil or electric coil. The central CAV system delivers air at a sufficiently low temperature to satisfy the requirement of the zone with the lowest heating load. Then, the reheat coils add the extra heating loads required in the other zones locally [14]. This solution can satisfy local temperature requirements, however, at the same time, it can lead to high energy use. The solution may need simultaneous central cooling and local heating of a constant amount of air that is the waste of energy [15]. Moreover, this solution can have rather high maintenance and installation costs. For instance, in case of having hot water reheat coils, the system requires circulation of both air and water to each zone which makes the system more expensive than a conventional CAV or VAV system [16].

Polak et al. have recently designed and developed a new combined ventilation and air heating system, so-called Heat Valve Ventilation (HVV) system, which provides the possibility of temperature control in individual rooms in a residential building [17]. Instead of local reheat coils, which can have rather high maintenance and installation costs, the air heating system comprises a central heating box with a built-in heating coil and temperature dampers for individual room temperature regulation. Numerical and experimental results, both in a laboratory and a real-life building indicate the ability of HVV system to achieve air temperature control on room level without a significant thermal stratification and draught rate in the rooms. HVV system can be a useful solution particularly in highly-insulated residential buildings with low heating demand, where balanced mechanical ventilation systems are frequently used. The focus of this work is however on the mechanical development of the system, where several experiments were done to illustrate e.g. the variation of supply air temperature to each zone, the pressure drop in the system and the effectiveness of the coil heat transfer. The system was only operating at constant airflow rate similar to a CAV system and the study lacks applying and evaluation of different control strategies.

In VAV systems, supply airflow rate can be regulated locally via terminal dampers or motor flaps to satisfy each zone specific temperature requirement. In principle, compared to CAV systems, VAV systems are more efficient for multi-zone buildings due to more precise temperature regulation and less energy use. However, VAV systems require more sophisticated control strategies in order to operate at their best performance and optimum efficiency [18,19], hence being more expensive solution. On the other hand, VAV systems are expected to be more cost-competitive in the future, with the recent advancements of low-cost IoT sensors, devices and data communication technologies. In addition to energy efficiency, VAV system can enhance the energy flexibility of a building with providing an optimal and an adaptive control of the HVAC system. Buildings can offer energy flexibility services to energy system to mitigate the impact of unpredictable renewable resources on the stability of power grid and in return, they can benefit in several ways [20]. Due to fast response of variable speed fans compared to coil temperature to maintain the set-points, VAV system can provide ancillary services such as fast response frequency regulation [21]. The energy flexibility of buildings with VAV HVAC systems has been mostly studied for commercial buildings [22,23], whereas the research works are limited for residential buildings. In these studies, a single indoor temperature has been considered, despite of growing interest for having different temperature zones within a residential building [24].

1.1. Research gap and contribution

The application of multi-zone VAV systems in residential buildings is becoming accessible economically, while there can be a great potential for improving energy efficiency and energy flexibility of residential buildings and at the same time addressing the need for thermal zoning with such systems. However, the application of VAV systems in residential buildings has not been very common in practice [25] and these systems have been rarely investigated and improved in academia. In the current research study,

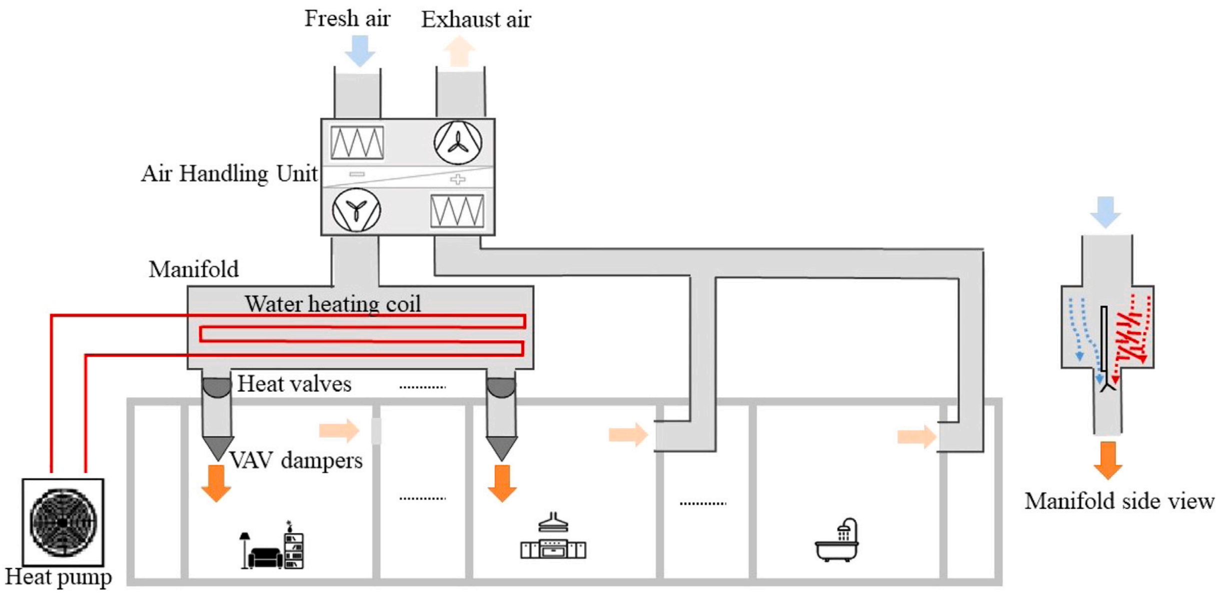

Fig. 1. Novel-designed air heating and ventilation system, so-called HVV system: Overall design (left) and the manifold side view (right).

we develop a novel-designed multi-zone VAV system for both ventilation and heating of single family houses. Our base is the novel HVV system introduced and successfully tested as a CAV system in [17]. In the current research, control logic for operating the system as a VAV system are provided and the results from experimental evaluation of the logic are presented. Unlike conventional VAV systems, our proposed VAV system provides the possibility for both airflow rate and air temperature regulation in each room within a dwelling. This possibility can further enhance the flexibility of the system as studied in our previous works [26,27].

Fig. 1 shows a sketch of the novel HVV system design. In this design, there can be rooms with only supply duct and rooms with only extract duct. The latter can be bathrooms or utility rooms, where, exhaust ventilation is a suitable solution and heating is usually provided via a supplementary heating system, e.g. a floor heating. There can also be rooms with both supply and extract ducts. An example of such rooms is kitchen. The system is a balanced mechanical ventilation system with an Air Handling Unit (AHU) with standard components i.e. supply and exhaust fans, filters and a heat exchanger. The novel component is the manifold with a built-in heating coil and so-called Heat Valves (HVs) for each room with supply duct. The manifold provides individual room temperature control via regulation of associated Heat Valve [28]. The Heat Valve regulates supply air temperature to each room by adjusting the amount of air passing or bypassing the heating coils in the manifold, see Fig. 1, manifold side view. Supply air temperature at the dwelling level can be regulated via adjustment of supply water temperature to the manifold. Similar to conventional VAV ventilation systems, supply airflow rate can be regulated at the dwelling and the room level via the adjustment of supply fan speed and VAV damper opening degree respectively.

In the current research, not only control of indoor air temperature and ventilation rate, but also control of indoor relative humidity is considered in the control logic. However, this part of the logic has not been evaluated experimentally due to limitation of the experimental setup and the evaluation is postponed to our future field experiment. The rest of the paper is organized as follows: The VAV control logic is provided in Section 2. The laboratory experimental setup employed to test the control logic is presented in Section 3. Section 4 presents and discusses the experimental results. Concluding remarks as well as suggestions for future work are provided on Section 5.

2. VAV control system

2.1. Control objective

The VAV control system ensures the operation of the HVV system as a balanced VAV ventilation system such that the requirements for indoor climate are satisfied and the system operates energy-efficient. Indoor climate requirements are considered as:

$$T_{i\text{-set}} - dT_i \le T_i \le T_{i\text{-set}} + dT_i \quad \text{for} \quad i = 1, \dots, n_s \tag{1}$$

$$Q_i \ge Q_{i\text{-min}} \quad \text{for} \quad i = 1, \dots, n_s \tag{2}$$

$$RH_i \le RH_{i\text{-max}} \quad \text{for} \quad i = 1, \dots, n_e \tag{3}$$

where, T_i, $T_{i\text{-set}}$ and dT_i are the temperature [°C], desired temperature set-point [°C] and allowed temperature difference around the temperature set-point [°C] in supply zone i respectively. The latter is considered to avoid frequent on/off operation of the motors, hence reducing wear and tear. Q_i [L/s] and $Q_{i\text{-min}}$ [L/s] are the supply airflow rate and minimum requirement for the supply airflow rate to supply zone i respectively. RH_i [%] and $RH_{i\text{-max}}$ [%] denote the relative humidity and the maximum allowed relative humidity in exhaust zone i respectively. The number of supply zones and the number of exhaust zones are denoted with n_s and n_e respectively. Eq. (1) specifies the requirement to satisfy thermal comfort, whereas Eq. (2) specifies the requirement to satisfy indoor air quality. Eq. (3) is considered to limit the maximum relative humidity in the exhaust zones, e.g. in bathrooms with occasional high relative humidity to prevent mold and building damage.

To achieve energy-efficient operation, the HVV system should be controlled according to the actual demand. This is implemented in two ways: (1) by setting $Q_{i\text{-min}}$ to a lower value when the zone is unoccupied (2) by regulating the supply fan speed according to the most air demanding zone. The latter is called critical zone reset strategy. The supply fan should operate with a lower capacity when demand decreases. Practically, maximum energy saving can be obtained by regulating the supply fan speed according to the demand of the most air demanding zone, so-called critical zone. This ensures that enough airflow is available in all zones, but simultaneously ensures that the supply fan does not provide more pressure than necessary to meet the demand. In this strategy, the supply fan is regulated such that at least the VAV damper at the critical zone is kept wide open. In practice, for stable system operation, the set-point for VAV damper opening degree at the critical zone is often set to 85% instead of 100%, allowing an opening range of 85% to 100% at the critical zone. The critical zone may change during the system operation. One way for detecting the critical zone is to monitor the opening degree of terminal devices. The zone with the most open terminal device is the critical zone [29].

2.2. Control logic

The HVV system provides the possibility to control supply airflow rate and supply air temperature of each individual zone within a dwelling. Hence, when heating demand changes in a specific zone, there are two ways to change the supply heating, either by changing first the supply airflow rate, then the supply air temperature to that zone or vice versa. Control logic of the HVV system is based on keeping supply airflow rate as low as possible during the system operation. This reduces the noise of the system and the risk of having dry air. Therefore, any increase in the heating demand of a specific zone, first activates the HV associated to that zone to open in order to raise the supply air temperature. When the HV is fully open and the heating demand is not still met, the associated VAV damper is activated to raise the supply airflow rate. On the other hand, any decrease in the heating demand first activates the VAV damper to close in order to lower the supply airflow rate, then the HV to close in order to lower the supply air temperature.

Algorithm 1 Control logic for HVV system

while HVV system is in operation **do**
  for $i = 1$ to $i = n_s$ **do**
    Call HV logic in supply zone i
    Call VAV logic in supply zone i
  end for
  Call Supply fan logic
  Call Exhaust fan logic
  Wait τ
end while

Considering the above-mentioned principle, control logic for HVV system operation is outlined in Algorithm 1. The algorithm consists of four subroutines which are executed in parallel every τ seconds while the system is in operation. The subroutines are for Heat Valve (HV) operation, VAV damper operation, supply fan operation and exhaust fan operation shown in flowcharts in Figs. 2–5 respectively. Relevant measured signals are read in the beginning of each flowchart. Table 1 lists all the signals/parameters transferred throughout the plant, i.e. the HVV system, the controller and the building in four categories. The first category are the input signals to the controller that are measured from the HVV system and the building. Temperature and CO2 concentration of each supply zone and relative humidity of each exhaust zone are the required signals measured from the building. From the HVV system, HV and VAV opening degree in each supply zone, supply and exhaust fan power in percentage and air pressure in the manifold are measured. The second category of Table 1 shows the control signals as output of the controller to the HVV system. These are basically the output of the four flowcharts, i.e. set-points for HV and VAV opening degree as well as set-points for supply and exhaust fan power in percentage. The last two categories of the table are the system parameters that can be either tunable or fixed during system operation. Tunable parameters include maximum relative humidity preferred in each exhaust zone, temperature set-point in each supply zone and the allowed temperature variation around the set-point in occupied and unoccupied situation. These parameters can be adjusted by the building user or a third-party e.g. an Smart Grid aggregator. There are also fixed parameters shown in Table 1. These parameters are either related to the system characteristics, e.g. the number of supply/exhaust zones or defined during system commissioning e.g. minimum/maximum air pressure in the manifold.

Table 1
Data framework.

Symbol	Description
Input measured signals	
T_i	Temperature in supply zone i [°C]
RH_i	Relative humidity in exhaust zone i [%]
C_i	CO2 concentration in supply zone i [ppm]
HV_i	HV opening degree of supply zone i [%]
VAV_i	VAV damper opening degree of supply zone i [%]
f_s	Supply fan power [%]
f_e	Exhaust fan power [%]
p	Air pressure in the manifold [Pa]
Output control signals	
$HV_{i\text{-set}}$	Set-point for HV opening degree of supply zone i [%]
$VAV_{i\text{-set}}$	Set-point for VAV damper opening degree of supply zone i [%]
$f_{s\text{-set}}$	Set-point for supply fan power [%]
$f_{e\text{-set}}$	Set-point for exhaust fan power [%]
Tunable parameters	
$T_{i\text{-set}}$	Set-point for temperature in supply zone i [°C]
$dT_{i\text{-occ}}$	Allowed temperature variation around the set-point in supply zone i, when it is occupied [°C]
$dT_{i\text{-unocc}}$	Allowed temperature variation around the set-point in supply zone i, when it is unoccupied [°C]
$RH_{i\text{-max}}$	Maximum relative humidity in exhaust zone i [%]
Fixed parameters	
$p_{\min}$	Minimum air pressure in the manifold [Pa]
$p_{\max}$	Maximum air pressure in the manifold [Pa]
$Q_{i\text{-occ}}$	Minimum supply airflow rate to supply zone i, when it is occupied [L/s]
$Q_{i\text{-unocc}}$	Minimum supply airflow rate to supply zone i, when it is unoccupied [L/s]
$Q_{i\text{-max}}$	Maximum supply airflow rate to supply zone i [L/s]
n_s	Number of supply zones
n_e	Number of exhaust zones
a, b	Constants for exhaust fan logic
K_l, K_g	Constant correcting factors
T_d	Constant time delay

After reading necessary signals, there are parameters should be calculated every cycle in HV logic and VAV logic, namely minimum and maximum limit of temperature in supply zones, supply airflow rate and minimum limit of supply airflow rate to supply zones. Limits of temperature are calculated as follows:

$$\begin{aligned} T_{i\text{-min}}(t) =& M_i(t)(T_{i\text{-set}}(t) - dT_{i\text{-occ}}(t)) \\ &+ (1 - M_i(t))(T_{i\text{-set}}(t) - dT_{i\text{-unocc}}(t)) \end{aligned} \tag{4}$$

$$\begin{aligned} T_{i\text{-max}}(t) =& M_i(t)(T_{i\text{-set}}(t) + dT_{i\text{-occ}}(t)) \\ &+ (1 - M_i(t))(T_{i\text{-set}}(t) + dT_{i\text{-unocc}}(t)) \end{aligned} \tag{5}$$

where, $M_i(t)$ is the occupancy in supply zone i, which returns a binary value with 0 as unoccupied and 1 as occupied. Minimum temperature limit is the temperature set-point minus the allowed temperature variation around the set-point, whereas maximum temperature limit is the temperature set-point plus the allowed temperature variation around the set-point. The allowed variation can be different depending on the occupancy condition. Occupancy in supply zones can be estimated based on indoor climate data, mainly CO2 concentration [30]. However, occupancy estimation is not in the scope of current paper and has been addressed in our research in [31].

Supply airflow rate to each supply zone should be available for the control logic. In the current study, airflow rate supplied to each zone is estimated based on one single measurement of air pressure in the manifold and the opening degree of terminal VAV damper at that zone (See Appendix A for more information about airflow rate estimation in HVV system). Minimum supply airflow rate required for each supply zone is varied between two levels according to occupancy as calculated in Eq. (6):

$$Q_{i\text{-min}}(t) = M_i(t)Q_{i\text{-occ}}(t) + (1 - M_i(t))Q_{i\text{-unocc}}(t) \tag{6}$$

where, $Q_{i\text{-occ}}$ [L/s] and $Q_{i\text{-unocc}}$ [L/s] are the minimum supply airflow rate required in occupied and unoccupied condition respectively. Minimum airflow rates per square meter are defined according to building regulations to ensure acceptable indoor air quality. Considering the designed capacity of the system, supply airflow rate to each supply zone i should also be below a maximum limit. Having the heat loss of a zone which occurs in the coldest day of a year, the maximum supply airflow rate for that zone can be calculated from the following equation:

$$Q_{i\text{-max}} = 1000 \times \frac{P_i}{\rho c_p (T_{\text{supply air}} - T_{\text{supply zone}})} \tag{7}$$

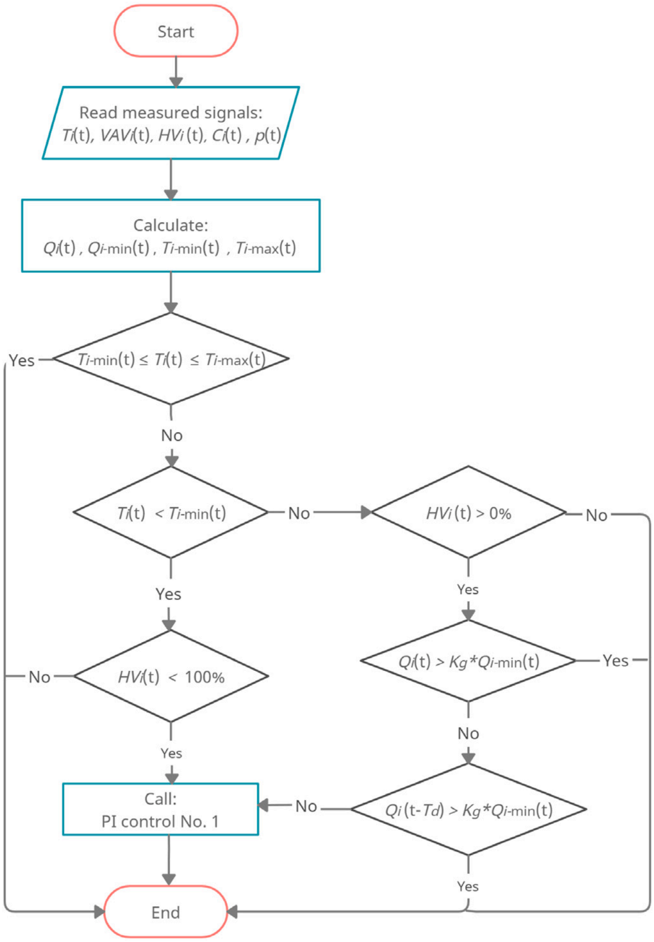

Fig. 2. Heat Valve (HV) logic in supply zone i.

Table 2
PI controllers used in the logic.

No.	$e(t)$	$u(t)$
1	$T_{i\text{-set}} - T_i$	$\Delta HV_{i\text{-set}}$
2	$T_{i\text{-set}} - T_i$	$\Delta VAV_{i\text{-set}}$
3	$Q_{\text{set}} - Q_i$	$\Delta VAV_{i\text{-set}}$
4	$VAV_{\max} - VAV_{\text{set}}$	$\Delta f_{s\text{-set}}$
5	$p_{\text{set}} - p$	$\Delta f_{s\text{-set}}$

where, $Q_{i\text{-max}}$[L/s] is the maximum supply airflow rate to supply zone i. P_i[W] is the heating power delivered to supply zone i and should be equal to the heat loss of the zone in the coldest day of a year. $T_{\text{supply air}}$[°C] and $T_{\text{supply zone}}$[°C] are the maximum supply air temperature and the desired supply zone temperature designed for the coldest day of a year. ρ[kg/m^3] and c_p[J/kg°C] are the density and specific heat capacity of air. Heat loss in the coldest day can be calculated with having information regarding U-values[W/m^2 °C] of building surfaces e.g. walls, windows, doors and the building dimensions.

The logic consist of five proportional–integral (PI) controller, which regulate the opening degree of HV and VAV dampers as well as supply and exhaust fan power in percentage in different operating conditions. Eq. (8) shows the overall control function, where $e(t)$ is the input to the controller defined as the deviation of the measured process variable from the desired set-point and $u(t)$ is the output of the controller. K_p and τ_i known as proportional gain and integral time are the tuning parameters. $e(t)$ and $u(t)$ for each of the PI control is given in Table 2.

$$u(t) = K_p e(t) + \frac{K_p}{\tau_i}\int_0^{\tau} e(\tau)\, d\tau \tag{8}$$

Following are all the regulations enforced by the control logic:

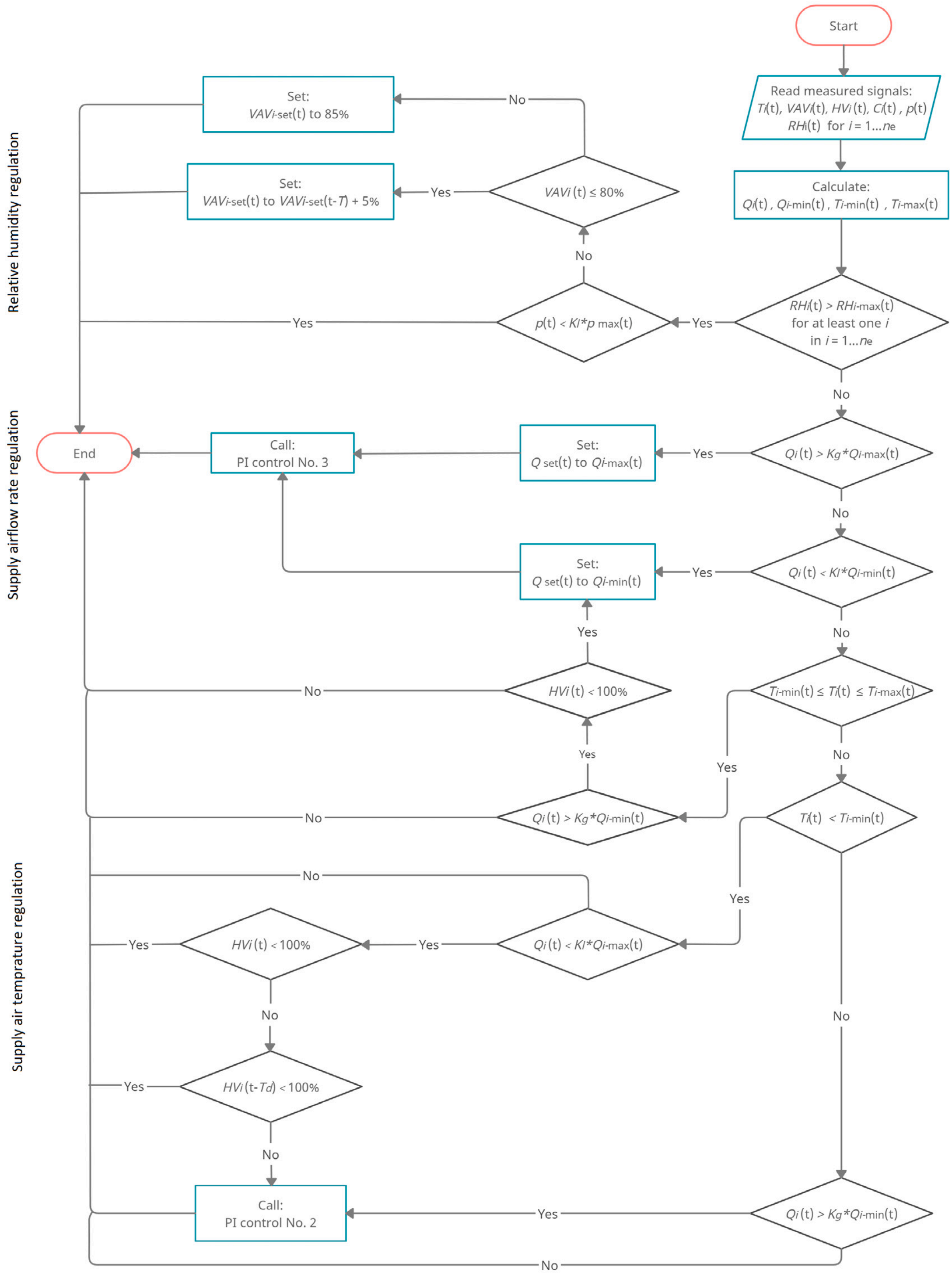

Fig. 3. VAV damper logic in supply zone i, in which the top, middle and bottom part of the flowchart represent the relative humidity, the supply airflow rate and the supply air temperature regulation of the logic respectively.

Temperature regulation in supply zones: When temperature is below the minimum limit in a supply zone, the associated HV, if it is not 100%, is regulated to open more via PI control No. 1 (see Fig. 2). When the HV is 100% for more than T_d seconds and the temperature is still below the minimum limit, the associated VAV damper, if the supply airflow rate is

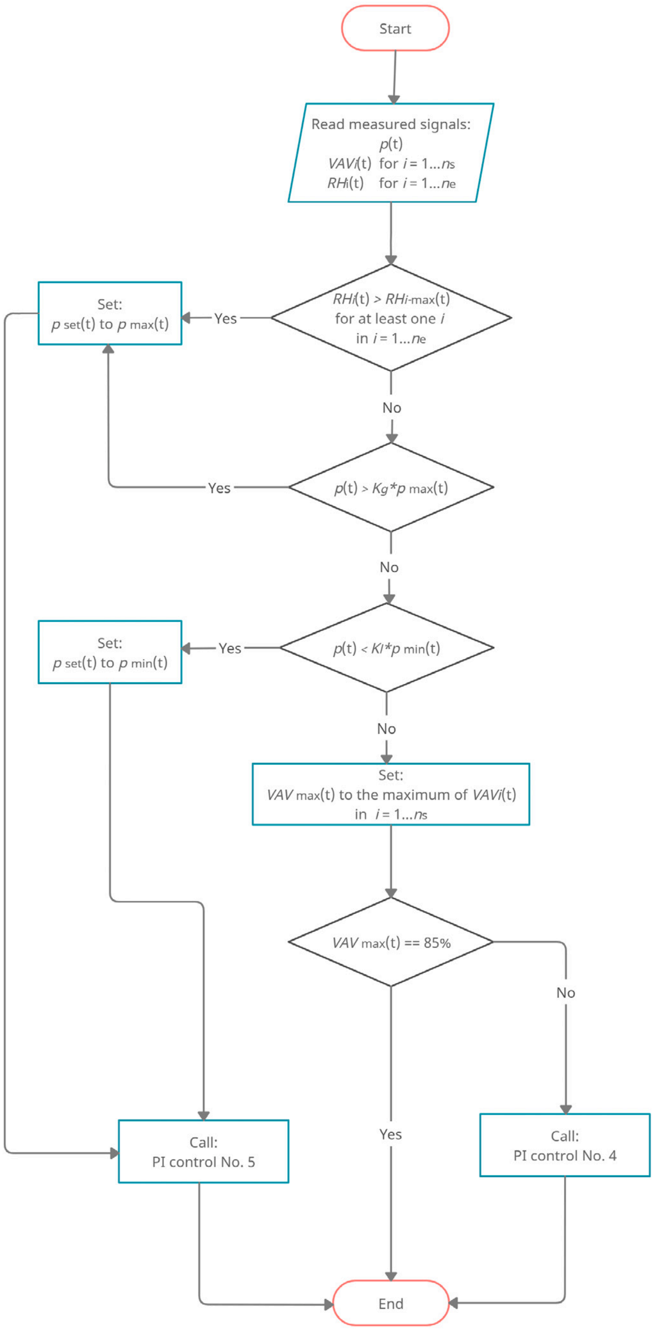

Fig. 4. Supply fan logic.

below its maximum limit, is regulated to open more via PI control No. 2 in order to provide extra heating (see Fig. 3). When temperature is above the maximum limit in a supply zone, the associated VAV damper, if the supply airflow rate is above its minimum limit, is regulated to close more via PI control No. 2 (see Fig. 3). When the supply airflow is at its minimum limit for more than T_d seconds and the temperature is still above the maximum limit, the associated HV, if it is not 100% closed, is regulated to close more via PI control No. 1 to further reduce the heating (see Fig. 2).

Supply airflow rate regulation in supply zones: Supply airflow rate to each supply zone should be always above a minimum and below a maximum limit during the system operation. As explained above, the minimum limit is due to requirement for indoor air quality defined in building regulations and the maximum limit is due the capacity of the system defined for the

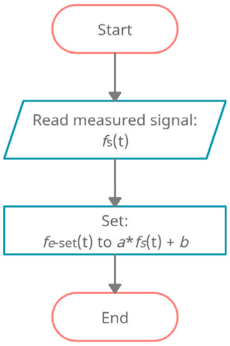

Fig. 5. Exhaust fan logic.

coldest day of the year. When these limits are violated, the associated VAV damper is regulated to open/close more via PI control No. 3 (see Fig. 3).

Manifold pressure regulation: Air pressure in the manifold should be always above a minimum and below a maximum limit during the system operation, where these limits are due to requirement for indoor air quality (at the building level) and the capacity of the system defined in the commissioning period. When these limits are violated, supply fan speed is regulated to increase/reduce the air pressure via PI control No. 5 (see Fig. 4).

Fan optimizer regulation: When the manifold pressure is within the limits, supply fan speed is regulated to keep the most open VAV damper at least 85% open during the system operation via PI control No. 4, i.e. applying the critical zone strategy for optimizing the fan energy use (see Fig. 4).

Relative humidity regulation in exhaust zones: When relative humidity exceeds the maximum limit in any of the exhaust zones, supply fan speed is regulated to raise the manifold pressure to the maximum pressure limit via PI control No. 5 (see Fig. 4). When the manifold pressure reaches the maximum limit and the relative humidity is still above the maximum limit in any of the exhaust zones, VAV dampers in all supply zones open for 5% per cycling time (τ). This will continue until either the acceptable relative humidity level is reached in all exhaust zones or the VAV dampers are 85% open in all supply zones (see Fig. 3).

Exhaust fan regulation: Exhaust fan speed is regulated based on supply fan speed (see Fig. 5), such that total exhaust airflow rate is always 5% more than the total supply airflow rate. This creates a negative pressure in the building. Although the system is a balanced ventilation system, a slight negative pressure in the building is desirable to prevent exfiltration. a and b are constant values in the logic that are defined during the system commissioning.

The constants, K_l and K_g are correcting factors which set to 0.95 and 1.05 respectively. They are added to soften the inequality conditions, since there can be an insignificant steady-state error in following the minimum and the maximum limits in practice. The constants prevent the logic to be stuck at the inequality conditions that can occur due to this steady-state error.

3. Experimental setup

The performance of the proposed control logic was evaluated with a prototype of the HVV system constructed in a controlled laboratory environment. Fig. 6 shows an overview of the laboratory setup (a) and the manifold system in the laboratory (b). The mechanical part of the prototype consisted of an Air Handling Unit (AHU), a manifold with six outlets connected to six supply zones and six built-in HVs, six VAV dampers, supply air ducts connecting the AHU, manifold and supply zones, extract air ducts removing the air from three exhaust zones and an electric water heater as a heat source. The supply and exhaust zones were simulated with nine white wood cabinets placed in two rows, where eight of them had the dimension of 0.6 m width, 1.0 m length and 2.36 m height and one had a bigger dimension as 0.6 m width, 2.0 m length and 2.36 m height. To simulate the heat loss in the cabinets, a separate cooling system was connected to the prototype, where the cooled air was distributed in the ductworks circulated in the cabinets, see Fig. 6(c). In addition, bottles of water were placed in the cabinets during the experiments to add further thermal capacity to the zones.

The HVV system prototype was equipped with a number of measuring instruments providing the necessary signals for the control logic. The manifold pressure was measured with ±1 Pa measuring error. Air temperature was measured with $0.3 + 0.005\ |T|$°C measuring error. Airflow rate was measured with the measuring error of ±5% or ±1 L/s depending on which was the highest value. Furthermore, a number of measuring instruments were installed for the monitoring purpose. The measured data were transferred to local data loggers either as 4–20 mA and 0–10 V analog signals or Modbus signals and then via a gateway unit with a secure MQTT connection to a cloud system. The control logic was implemented in the cloud system, which provided a web-based API as well as a Matlab and a Python toolbox [32] for reading time-series measurements and sending set-points to the HVV prototype. The separate cooling system had its own control logic also implemented in the cloud system. The cooling control logic received cooling

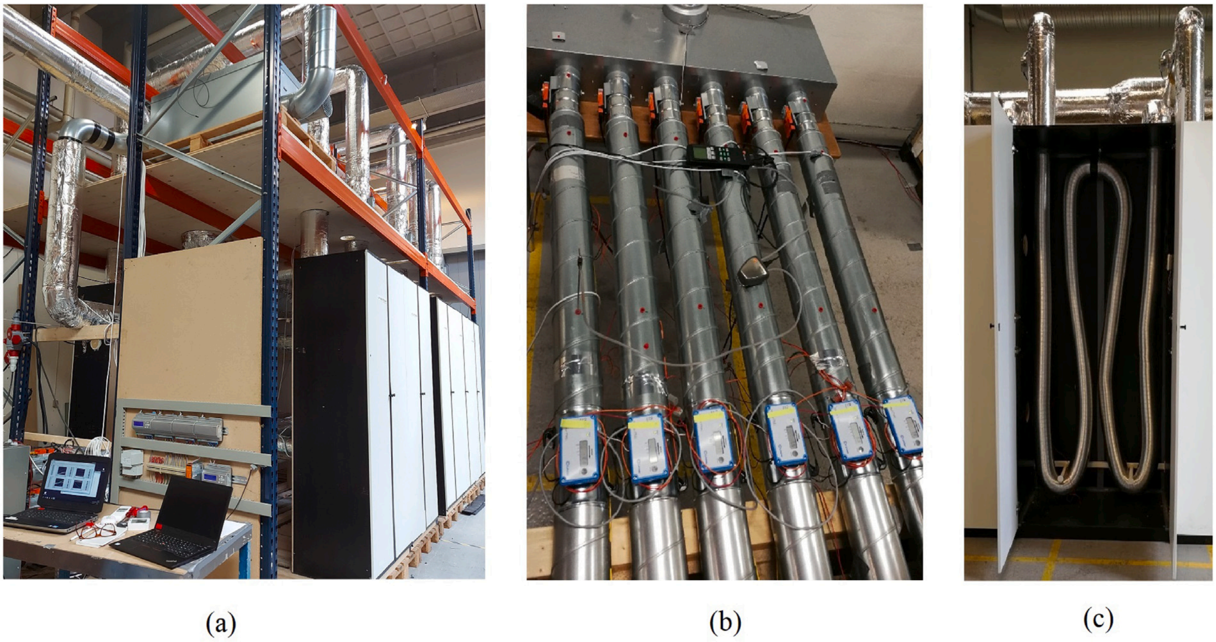

Fig. 6. (a): Overview of the laboratory experimental setup (b): Manifold (c): Cooling system for simulation of heat loss.

Table 3
Floor area of the rooms and heating demands in the rooms for the coldest day in a year.

Rooms	Floor area	Heating demand
Bedroom 1	11.6 m^2	187 W
Office	12.2 m^2	161 W
Living room	46 m^2	505 W
Kitchen	30.9 m^2	516 W
Big bedroom	17 m^2	336 W
Bedroom 2	11.6 m^2	187 W

Table 4
Minimum and maximum supply airflow rate to supply rooms commissioned for the laboratory prototype.

Rooms	$Q_{i\text{-min}}$	$Q_{i\text{-max}}$
Bedroom 1	4.5 L/s	7.3 L/s
Office	4.2 L/s	6.7 L/s
Living room	12.4 L/s	18.6 L/s
Kitchen	12.4 L/s	18.9 L/s
Big bedroom	9.2 L/s	13.7 L/s
Bedroom 2	5.4 L/s	8.1 L/s

load set-point, measured supply and return cooling air temperature as input and output a set-point for cooling airflow rate to the HVV prototype.

The laboratory setup consisted of six supply zones further referred as Bedroom 1, Office, Living room, Kitchen, Big bedroom and Bedroom 2. Although all the rooms had the same actual size (except Kitchen which was twice the size of the others), by the means of the cooling system, these rooms were commissioned as rooms of a real-size building with different floor areas and heating demands, as listed in Table 3. Different heating demands were generated in the rooms by assigning different cooling load set-points to the rooms in the cooling system. The minimum supply airflow rate per square meter was considered 0.3 L/sm^2 in occupied condition, according to Building Regulations (BR18) in Denmark [4]. Hence, the minimum supply airflow rate to each room was calculated by multiplying the minimum supply airflow rate per square meter and the floor area given in Table 3. Likewise, the maximum supply airflow rate to each room was calculated based on heating demands given in Table 3 applying Eq. (7). The commissioned values for the minimum and maximum supply airflow rate were a bit different from the calculated values thought, given in Table 4. The minimum and the maximum air pressure limits in the manifold were defined as $p_{\min}$ = 18 Pa and $p_{\max}$ = 34 Pa after the system commissioning (See Appendix B for more information about the HVV system commissioning).

Table 5
Tuning parameters for the PI controllers of the laboratory prototype.

No.	K_p	τ_i
PI control 1	1.5 $\frac{\%}{°C}$	300 s
PI control 2	2 $\frac{\%}{°C}$	300 s
PI control 3	2 $\frac{\%}{L/s}$	10 s
PI control 4	0.1 $\frac{\%}{\%}$	120 s
PI control 5	2 $\frac{\%}{Pa}$	300 s

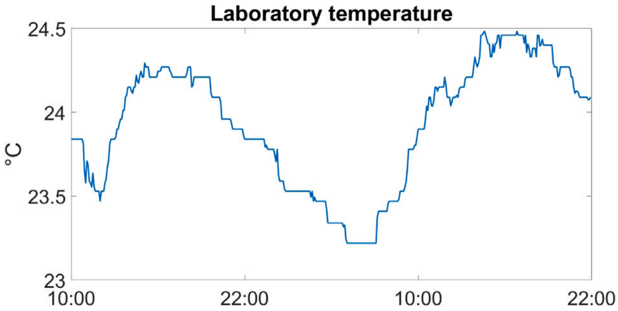

Fig. 7. Laboratory temperature during the experiment.

Limitations: With the laboratory prototype, it was not possible to simulate actual occupancy and daily activities that can generate humidity inside a building. There was no measurement of CO2 concentration and relative humidity. Hence, the parts of the logic related to occupancy estimation and relative humidity control could not be tested during the laboratory experiment. To proceed with the test of the control logic, virtual signals were generated in the cloud system representing relative humidity in the exhaust zones as well as occupancy in the supply zones as either 0 or 1. The occupancy signals were set to 1 in all supply zones and the relative humidity signals were set to values below the maximum limit of relative humidity in all exhaust zones during the experiment.

4. Experimental results and discussion

This section presents the results of an experiment performed for one day and a half on April 25th and 26th 2022 to evaluate the performance of the proposed control logic. Before this experiment, several experiments were done to tune the PI controllers. The final optimal parameters are listed in Table 5. Fig. 7 shows the laboratory temperature variations during the experiment. The outdoor temperature was varying between 2 °C to 16 °C with an average around 10 °C. To evaluate the performance of the system in providing different temperature in different supply zones, six different temperature set-points were initially given to the control logic, i.e. 20 °C, 21 °C, 22 °C, 23 °C, 24 °C, 25 °C. Later, the temperature set-points were changed in several zones, namely in Living room, Kitchen and Big bedroom, to evaluate the dynamic response of the system to temperature set-point change. Fig. 8 shows the temperature in the six supply zones during the experiment. An allowed temperature variation of 0.3 °C around the set-points was considered. The supply zones were subject to various heating demands during the experiment, partly due to variations in the ambient temperature and partly due to variations in the cooling load generated by the cooling system in each zone, as the cooling system operation was on/off. As shown in Fig. 8, the HVV system could handle all of these variations and maintain different temperature set-points in different zones. Except in few occasions, the zone temperature was within the allowable limits in all supply zones. Fig. 9 shows how supply air temperature to each zone changed during the experiment to satisfy the heating demands.

To better understand the HVV system operation, Figs. 10 and 11 show the HV and VAV damper reactions in the six supply zones during the experiment, where the top graphs show the HVs opening degree (right-side plots) together with zone temperature variations and the temperature limits (left-side plots), whereas, the bottom graphs show the VAV dampers opening degree (right-side plots) together with supply airflow rate variations and the airflow rate limits (left-side plots). As expected, the HVs were opened further as soon as the zone temperature dropped below the minimum temperature limit, either due to zone temperature variations in Bedroom 1, Office and Bedroom 2 or due to temperature set-point change in Living room, Kitchen and Big bedroom. There were moments in Office, Kitchen and Big bedroom, when the relevant HV was 100% open and the heating demand was not still satisfied, namely the second day of the experiment at 5:50 in Office, the first day of the experiment at 18:50 in Kitchen and the second day of the experiment at 13:30 in Big bedroom. In these situations, as shown, the relevant VAV damper was opened further, after T_d seconds which was set to 300 s in the experiment, to provide further heating.

On the other hand, as expected, when the supply airflow rate was above the minimum limit in a supply zone, the relevant VAV damper was closed further, as soon as the zone temperature exceeded the maximum limit, e.g. in Big bedroom. However, when the supply airflow rate was at the minimum limit, the relevant HV was closed further to reduce the heating, as soon as the zone temperature exceeded the maximum limit, e.g. in the rest of the supply zones. In relation to supply airflow rate regulation, the HVV

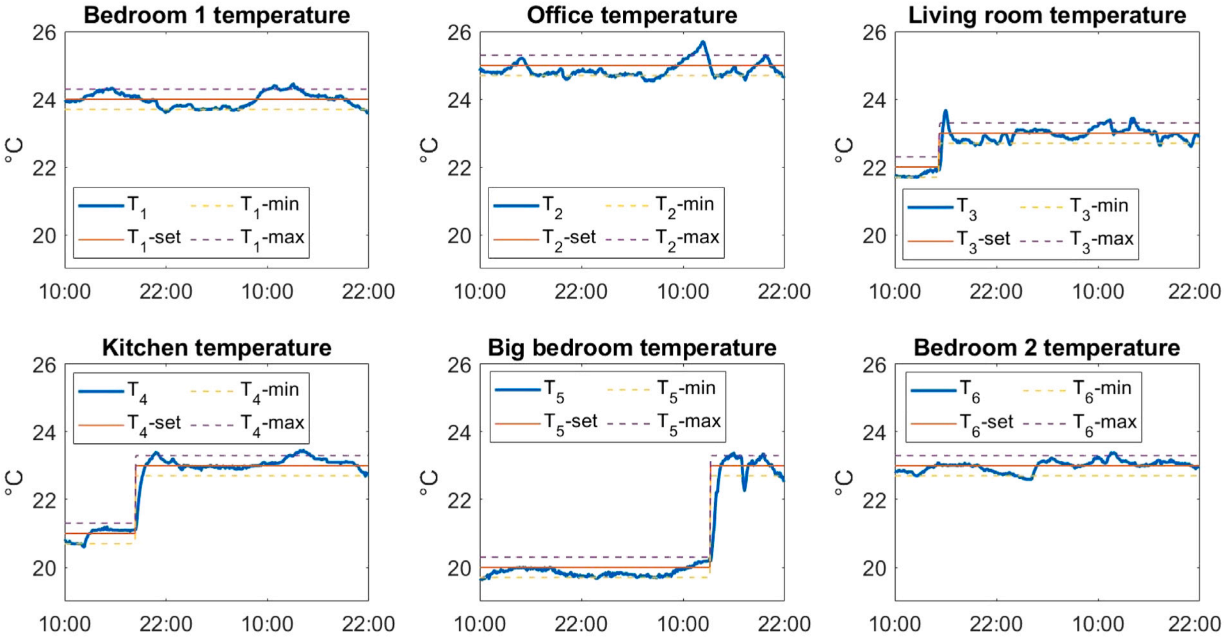

Fig. 8. HVV prototype — Supply zones temperature together with the temperature set-point, the minimum and the maximum temperature limits.

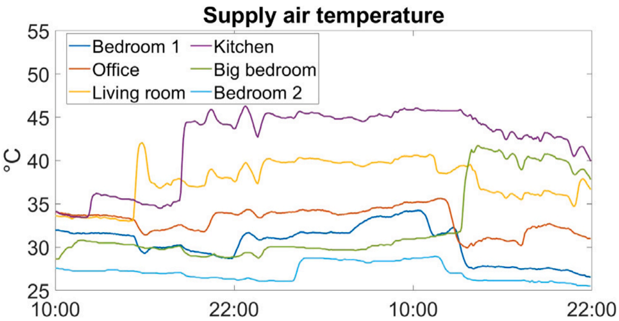

Fig. 9. HVV prototype — Air temperature supplied to supply zones.

system maintained the minimum supply airflow rate, i.e. no greater than $K_l Q_{i\text{-min}}$ (K_l was set to 0.95%) in all zones, as far as the heating demand was satisfied with the relevant HV regulation. When further heating was required and the HV was wide open at 100%, the relevant VAV damper was opened further, e.g. in Big bedroom, increasing the supply airflow rate above the minimum level. The supply airflow rate in all zones was below the maximum limit during the whole experiment, though. As shown, variations in airflow rate were greater in the end of the experiment, namely the second day of the experiment after 13:30. This was due to variations in the supply fan speed, as a result of increase in the VAV damper opening in the critical zone, i.e. Big bedroom, see Fig. 12 . In all zones, except Big bedroom, the VAV dampers closed further in this situation to keep the airflow rate as close to the minimum level as possible.

To understand the supply and the exhaust fan regulation, Fig. 12 shows the VAV damper opening in the critical zone, i.e. VAV-max (the left-side plot) together with the supply and the exhaust fan speed during the experiment (right-side plots). In a similar graph, in Fig. 13, the supply and the exhaust fan speed (right-side plots) are shown together with the manifold pressure and the minimum/maximum pressure limits (left-side plots). Looking at the two graphs, as expected, the VAV damper opening in the critical zone was maintained at least 85% open, while the manifold pressure was between the pressure limits. In the beginning of the experiment, namely until the second day of the experiment at 13:30, $VAV_{\max}$ was below 85%, however. In this period, the manifold pressure was at the minimum level, hence, the control logic did not allow further decrease in the manifold pressure that could make the critical VAV damper more open. During the experiment, the manifold pressure did not violate the pressure limits though. As defined by the logic, the exhaust fan speed was regulated based on the supply fan speed, following the same trend with just a small deviation.

Overall, the results of the laboratory experiment indicate the ability of the HVV system prototype in satisfying both thermal comfort and indoor air quality requirements of each individual zone. With respect to individual thermal comfort requirement,

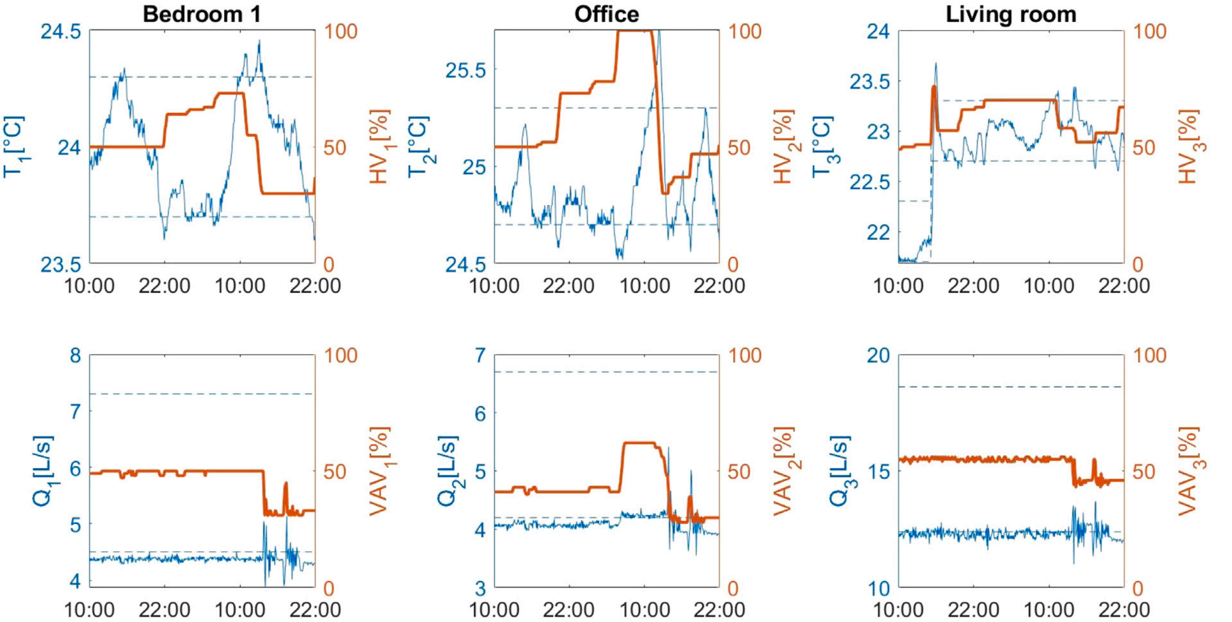

Fig. 10. HVs opening degree (top graphs, right-side plots) together with zone temperature variations and the temperature limits (top graphs, left-side plots), VAV dampers opening degree (bottom graphs, right-side plots) together with supply airflow rate variations and the airflow rate limits (bottom graphs, left-side plots) in Bedroom 1 (left), Office (middle) and Living room (right).

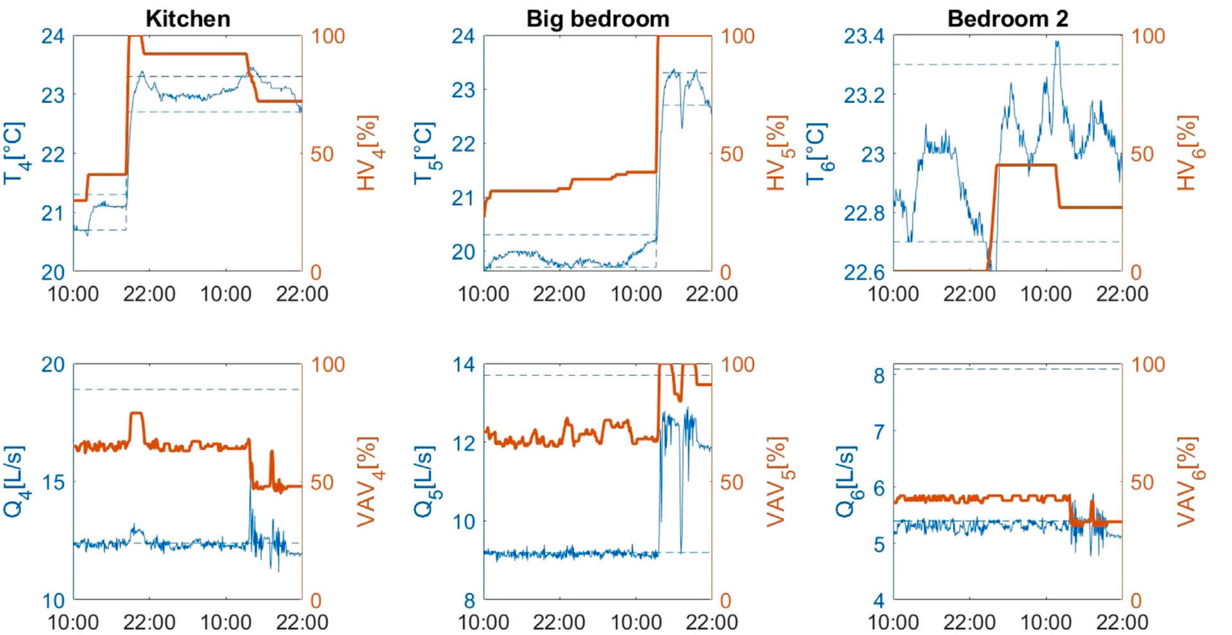

Fig. 11. HVs opening degree (top graphs, right-side plots) together with zone temperature variations and the temperature limits (top graphs, left-side plots), VAV dampers opening degree (bottom graphs, right-side plots) together with supply airflow rate variations and the airflow rate limits (bottom graphs, left-side plots) in Kitchen (left), Big bedroom (middle) and Bedroom 2 (right).

different temperature set-points ranging 20 °C to 25 °C were satisfied with only ± 0.3 °C deviation around the set-points. When temperature set-points were changed, new set-points were followed quickly with an acceptable overshoot. However, more investigation is required regarding the response time of the system to change in the temperature set-points, since the size of the rooms in the experimental setup was rather small compared to a real-life case. In addition, only one indicator of thermal comfort, i.e. air temperature has been discussed in the experimental results. In particular, the impact of air velocity on the thermal comfort,

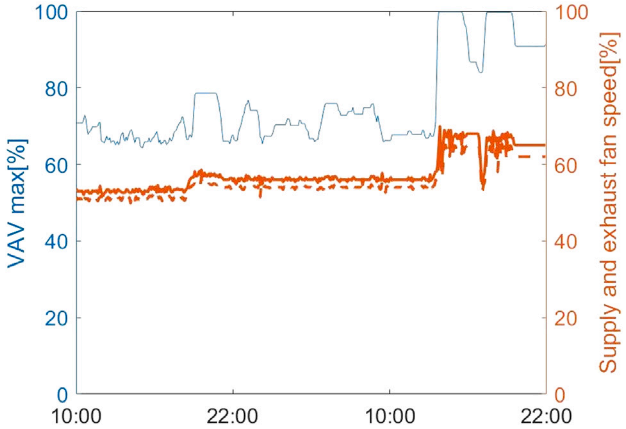

Fig. 12. VAV damper opening in the critical zone, VAV-max (the left-side plot) together with the supply and the exhaust fan speed (right-side plots, where the supply fan speed is shown with the solid line and the exhaust fan speed is show with the dashed line).

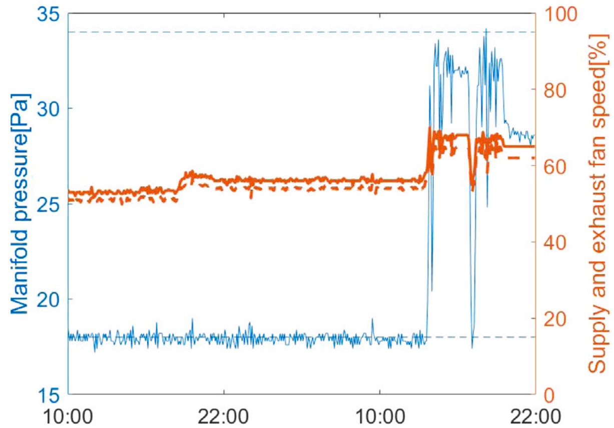

Fig. 13. The manifold pressure and the minimum and the maximum pressure limits (left-side plots) together with the supply and the exhaust fan speed (right-side plots, where the supply fan speed is shown with the solid line and the exhaust fan speed is show with the dashed line).

e.g. with measuring the draft rate should be evaluated for an air heating system. In this regard, the location of the supply openings in the supply zones can be an important matter. Another factor which has not been considered and should be addressed is the relative humidity in the supply zones, since there can be a high risk of dry air with an air heating system during winter time. Since the proposed logic is based on keeping supply airflow rate as low as possible, there is not much concern regarding high draft rate and having dry air though.

Likewise, the system could satisfy the individual zone requirement for supply airflow rate to provide an acceptable indoor air quality. However, the supply airflow rate variations were rather high in the end of the experiment as a result of variations in the supply fan speed in this period. The integral time (τ_i) of PI control No. 4 can be increased to prevent these high variations occurring when there is a significant change in the heating demand. This can lower the speed of the system in general though, i.e. also when there is no significant change in the heating demand. Alternatively, a self-tuning PI controller can be employed which adapts the PI parameters according to the system dynamic changes [33]. Due to the lack of occupancy simulation in the laboratory prototype, the response of the system to change in airflow rate requirements in occupied and unoccupied conditions was not tested and needs to be investigated.

5. Conclusion

This study presented a novel-designed HVAC system, namely an air heating and ventilation system comprises a manifold with temperature dampers, and developed control logic for variable air volume regulation of the system. The proposed control logic enables energy-efficient and room-based control of indoor climate, i.e. room temperature and room air quality as well as the maximum level of relative humidity in rooms with rather high relative humidity, e.g. bathroom and laundry room. Except the relative humidity control, the performance of the control logic was tested in a laboratory environment with a full-scale prototype of the novel HVAC system. The experimental results indicated the ability of the system in providing and maintaining individual room temperature which varied in a wide range between 20 °C to 25 °C, while maintaining the required supply airflow rate needed for having an acceptable indoor air quality. As defined in the logic, the supply airflow rates were kept as low as possible during the experiment and only increased further when more heating was needed and the relevant temperature damper was fully-open. The system could quickly adapt to changes in the temperature set-points and satisfied the new desirable temperature without being unstable.

Although the experiment was performed with a full-scale prototype of the HVAC system, the cabinets that simulated the thermal zones were still far from a real-life residential building. In addition, the experimental setup did not provide the possibility to simulate occupancy and daily indoor activities such as cooking and taking a bath, hence the possibility to evaluate the performance of the logic in controlling indoor relative humidity. Furthermore, to evaluate the energy-saving potential of the proposed control logic in comparison with the reference zone control, a longer period of experiment during the heating season is required. To fully evaluate the performance of the HVAC system with respect to aforementioned aspects, the novel HVAC system has been installed in a single-family house in north of Copenhagen, Denmark, where the proposed logic will be implemented and tested in our future work.

CRediT authorship contribution statement

Samira Rahnama: Conceptualization, Methodology, Software, Validation, Investigation, Visualization, Writing – original draft, Review & editing, Funding acquisition. **Göran Hultmark:** Conceptualization, Methodology. **Klemen Rupnik:** Conceptualization, Methodology. **Pierre Vogler-Finck:** Conceptualization, Software, Investigation, Review & editing. **Alireza Afshari:** Conceptualization, Methodology, Review & editing, Funding acquisition, Project administration.

Declaration of competing interest

The authors declare the following financial interests/personal relationships which may be considered as potential competing interests: Alireza Afshari reports financial support was provided by The Energy Technology Development and Demonstration Programme (EUDP).

Data availability

Data will be made available on request.

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Appendix A. Airflow rate estimation

Supply airflow rate to each supply zone should be available for the control logic. To avoid excessive cost of direct airflow rate measurement in each supply zone, in the current study, airflow rate supplied to each zone is estimated based on one single measurement of air pressure in the manifold and the opening degree of terminal VAV damper at that zone. The latter is already available from the HVV system. The basis for this calculation is the relationship between the fully developed turbulent airflow and the pressure drop in a duct system, where the airflow rate is proportional to the square root of the pressure drop. This leads to Eq. (A.1) for calculation of the airflow rate:

$$Q_i(t) = K_i[VAV_i(t)]\sqrt{p(t)} \tag{A.1}$$

where, K_i is a function of VAV damper opening degree in supply zone i, Q_i [L/s] is the supply airflow rate to supply zone i and p[Pa] is the air pressure in the manifold. K_i functions are defined with interpolation techniques based on data collected during the commissioning. Hence, direct airflow measurement is only required during the system commissioning.

Fig. A.14 shows the measured and the calculated supply airflow rate to the six supply zones during the experiment. For the zones with a lower supply airflow rate, e.g. Bedroom 1, Office and Bedroom 2, the difference between the measured and the calculated airflow rate was within the uncertainty range of the airflow rate measuring instrument (±1 L/s). However, in the zones with a higher supply airflow rate, the difference was beyond the uncertainty range and could reach up to 47% at one point in Living room. The highest average difference between the measured and the calculated supply airflow rate also belonged to Living room as 33%. Apart from Living room, the average difference was below 18%. With respect to airflow rate estimation based on one single pressure measurement in the manifold, the accuracy of the method can be acceptable considering the cost of airflow rate measuring devices. However, more field investigation on the accuracy of the proposed method is required, since the performance of the system particularly in relative humidity control and the energy use of the system are dependent on the amount of supply airflow rate.

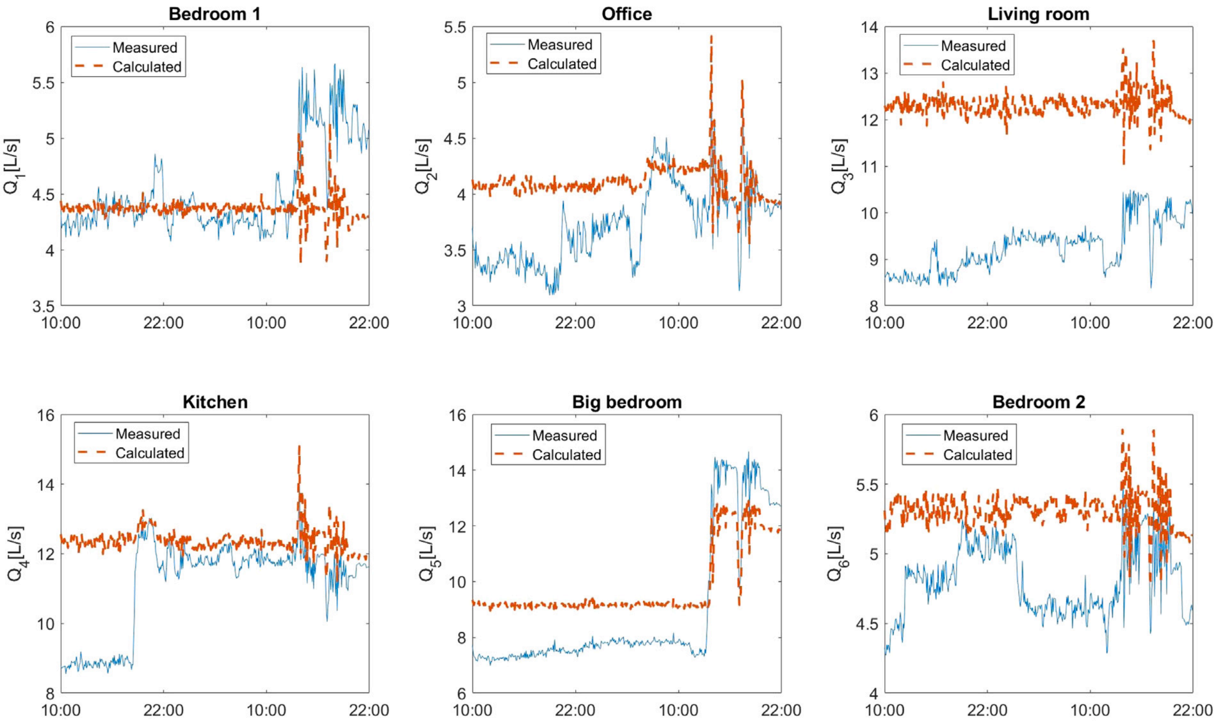

Fig. A.14. The measured supply airflow rate (solid line) and the calculated supply airflow rate based on the K_i functions (dashed line).

Appendix B. Commissioning

Other than terminal VAV dampers which dynamically regulate the supply airflow rate to each zone, additional control valves are also required for initial adjustment of supply airflow rate before the system operation in the commissioning process. The purpose of the commissioning is to find the position of the control valves based on the maximum designed capacity of each zone. The position of control valves will then be fixed during the system operation. Furthermore, the system parameters, namely $p_{\min}$, $p_{\max}$, a, b (constants for exhaust fan logic) and K_i functions are defined in the system commissioning. Here are the steps we follow for the system commissioning:

1. In all supply zones, the control valves are set to fully-open (100%), so as the terminal VAV dampers (85%). The supply fan is set to its maximum capacity.
2. After step 1, two situations can occur, Case 1: all supply zones have airflow rate above the maximum airflow rate calculated for the coldest day (see Eq. (7)), and Case 2: one or several supply zones has/have supply airflow rate below the maximum airflow rate calculated for the coldest day. In Case 1, we lower the supply fan speed, until we find a zone with the maximum supply airflow rate, while the supply airflow rate in the other zones is above the maximum limit. In Case 2, we close the control valve in the zone with the biggest deviation of supply airflow rate above the maximum airflow rate, such that the zone with the biggest deviation below the maximum airflow rate reaches to the maximum airflow rate, i.e. zero deviation.
3. After step 2, the air pressure in the manifold is recorded as $p_{\max}$.
4. The position of control valves in the rest of the supply zones are adjusted to reach the maximum supply airflow rates, while we keep the manifold pressure level at $p_{\max}$ by reducing the supply fan speed.
5. After step 4, the position of all control valves are defined. We lower the supply fan speed until we reach the minimum occupied airflow rate at the building level.
6. After step 5, the air pressure in the manifold is recorded as $p_{\min}$.

To define the K_i functions, a number of experiments are conducted, in which the VAV dampers opening degree in all supply zones are changing in steps from 0% to 100%, e.g. with 10% intervals and the supply airflow rates to each zone are recorded. During these experiments, the manifold air pressure should be kept constant. Then the k value at each opening degree of the VAV damper is calculated with Eq. (A.1). With using interpolation techniques, a polynomial curve is fitted to the k values calculated for supply zone i denoted as K_i function. Similarly, to define a, b constants, a number of experiments are conducted in the system commissioning, in which the supply and exhaust fan speeds are changing in steps and recorded, such that the total exhaust airflow rate is kept 5% more than the total supply airflow rate. Then, a, b are calculated with fitting a linear curve to the recorded data.

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