Energy efficiency in state institutions using decentralized ventilation in duct systems

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Preface

The present report describes the main findings of the research project *Energy efficiency in state institutions using decentralized ventilation in duct systems (Energieffektivisering i statslige institutioner ved anvendelse af decentrale ventilatorer i kanalsystemet).* The project has been carried out at the Danish Building Research Institute (SBi) in collaboration with two companies, Lindab Comfort A/S, Denmark and AB Regin, Sweden. The project has been financially supported by ELFORSK, a research and development program administrated by the Danish Energy Association under sagsnr. s2014-431 and projekt nr. 347-027. In addition, Bygningsstyrelsen (Danish Building and Property Agency) supported this work financially. The project was launched in September 2015 and completed in September 2017.

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Abstract

Nærværende forskningsprojekt har studeret energibesparelsespotentialet for et nyt mekanisk ventilationssystem, hvor indreguleringsspjæld er erstattet med decentrale ventilatorer. Indledningsvis er et konventionelt VAV-system (VAV=variable air volume=variable luftstrøm) med spjæld studeret med henblik på at indgå som reference for sammenligning med det nye ventilationssystem. Nærværende undersøgelse fremlægger en ny metode til implementering i praksis af styringsstrategien "static pressure reset" (SPR). Almindeligvis styres indblæsningsventilatorens ydelse i et VAV-system på baggrund af en trykføler anbragt i indblæsningskanalen. Ventilatorens omdrejningstal styres, så der opretholdes et konstant statisk tryk, der hvor føleren er placeret. Yderligere reduktion af energibehovet kan opnås ved at sænke setpunktet for trykket i kanalen når ventilationssystemet kun er belastet delvis. Strategien er i litteraturen kendt som "pressure reset control" eller "critical zone reset strategy". Den her fremlagte metode er evalueret gennem eksperimentelle studier i laboratoriet. Studierne viser, at effektbehovet reduceres med mellem omkring 10 % og 50 **%** ved anvendelse af "pressure reset control" i forhold til konstant statisk tryk.

Den fremlagte "static pressure reset"-strategi er i det følgende tilpasset et VAV-system med decentrale ventilatorer. Nærværende undersøgelse fremlægger en metode til at styre såvel hovedventilatoren som de decentrale ventilatorer så systemet er balanceret uanset belastningsgrad. Metoden til styring omfatter to niveauer – zoneniveau og systemniveau – og den er baseret på måling af det statiske tryk i kanalsystemet svarende til den energieffektive styring for ventilationssystemer med spjæld. På baggrund af to identiske forsøgsopstillinger i laboratoriet, én med spjæld og én med decentrale ventilatorer, er der foretaget sammenligninger af effektbehovet for de to systemer. Resultaterne viser den fremlagte metodes evne til at styre ventilationssystemet med decentrale ventilatorer. For de valgte volumenstrømme var de målte effektbehov i de to systemer i hovedsagen ens med undtagelse af meget lave volumenstrømme. Beregninger afslører et energibesparelsespotentiale på omkring 30 % afhængig af hovedventilatorens effektivitet.

Den fremlagte styringsmetode er blevet implementeret i et fuldskala VAV-system, som betjener tre undervisningsrum på Syddansk Universitet, SDU ved Odense. Tre decentrale ventilatorer er installeret i de tre undervisningsrum og de eksisterende spjæld er gjort inaktive i åben tilstand. Der foreligger data fra et års målinger på det oprindelige system og målinger på det nye system pågår. For at kunne indregne indvirkning af varierende udeklima er det nødvendigt, at det nye system med decentrale ventilatorer er i drift adskillige måneder endnu før at det er muligt at sammenligne de to systemer.

Background

Buildings use a large share of total energy use around 35–40% in many countries (Danish Energy Agency, 2013). For instance, in Denmark, buildings account for 40% of the Danish energy use (Dal, Rusbjerg, & Zarnaghi, 2012). In the USA, 41% of the energy is used in the buildings (Westphalen & Koszalinski, 1999). Therefore, it is motivated to investigate the energy saving potential in the building sector. Mechanical ventilation system is one of the most energy-demanding systems in the buildings which operate on a 12-months basis. Many studies have been focused on designing energy efficient ventilation systems and implementing energy efficient control strategies (Chenari, Dias Carrilho, & Gameiro da Silva, 2016) (Choa, Song, Hwangc, & Yunb, 2015) (Fahlen, Capacity Control of Air Coils in Systems for Heating and Cooling, 2007) (Goyal & Barooah, 2013) (Gunner, Afshari, Bergsøe, Vorre, & Hultmark, 2016). In a ventilation system, the dominant energy use is the fan. It pressurizes the air stream in order to overcome resistance in the duct system. Thus, the air can move in the duct system and reach the terminal dampers. The dampers are then used to regulate the airflow rate in response to the zone requirements. In general, there are two types of mechanical ventilation systems, so-called CAV (constant air volume) and VAV (variable air volume). In CAV systems, the supply fan provides a constant airflow rate at all conditions while in VAV systems, the airflow rate continuously varies during the system operation. The fan power demand depends on the provided airflow rate and the fan pressure.

$$P = \frac{\Delta p . q}{\eta} \qquad (1)$$

Where, P [W] is the fan power demand, q [m^3/s] is the required airflow rate, Δp [Pa] is the fan pressure and η is the fan efficiency. Thus, the fan power demand, and consequently the total energy demand, can be reduced either by reducing the total pressure drop in the duct system or the airflow rate at the same efficiency. A previous study at SBi (Gunner, Hultmark , Vorre, Afshari, & Bergsøe, 2014) has investigated a novel ventilation system to reduce the energy demand by reducing the pressure drop in the duct system. The idea was to replace terminal dampers with decentralized fans and thus eliminate the pressure drop across the dampers. This idea was first introduced in a report by Fahlen (Fahlen, Capacity Control of Air Coils in Systems for Heating and Cooling, 2007). The main focus is on capacity control of liquid-to-air heat exchangers. The report proposes to replace the central pump, balancing valves and control valves of a traditional hydronic system by distributed pumps on each radiator aiming at saving electricity. The author then expands the idea to air-based systems and proposes the replacement of VAV-boxes with terminal dampers by decentralized fans. There is however no detailed discussion on the control of such a ventilation system and the amount of energy saving in (Fahlen, Capacity Control of Air Coils in Systems for Heating and Cooling, 2007). Energy saving potential in the ventilation system with decentralized fans has been investigated by Gunner et al. in (Gunner, Hultmark , Vorre, Afshari, & Bergsøe, 2014). However, this study considers a CAV ventilation system with equal airflow rate for all zones. The current research study investigates the potential for energy saving in VAV ventilation systems with decentralized fans as well as the control of such a ventilation system through laboratory experiments and experiments in a real building in Odense, Denmark.

Objective

In this project, the following research questions have been addressed:

1 Is it possible to make balance in a VAV ventilation system with decentralized fans, i.e. to control the speed of main fan and the decentralized fans in relation to each other such that the ventilation demands are satisfied in all zones?

2 What is the potential for energy saving in a VAV system with decentralized fans in comparison with a VAV ventilation system with terminal dampers?

This project has been conducted based on experiments on a mock-up of a ventilation system in a laboratory environment. The VAV ventilation system with terminal dampers has been studied and tested as a reference for comparison with the novel ventilation system with decentralized fans in the initial step. The following objectives have been defined in order to answer the above mentioned questions:

- Development of an energy efficient control method for the VAV ventilation system with terminal dampers
- Development of an energy efficient control method for the VAV ventilation system with decentralized fans
- Experimental verification of the proposed control methods at the laboratory and a real building.
- Analyzing the experimental results

Bygningsstyrelsen provided the possibility to test the proposed methods in a real building, namely on a ventilation plant serving three classrooms at Southern University of Denmark (SDU) in Odense.

Dissemination

The outcome of this project has been disseminated through the following journal and conference papers as well as an indoor climate, comfort and energy event:

1 “Experimental study of the pressure reset control strategy for energy-efficient fan operation – Part 1: Variable air volume ventilation system with dampers”
Samira Rahnama, Alireza Afshari, Niels Christian Bergsøe, Sasan Sadrizadeh
Published in Energy and Buildings, 139 (2017) 72-77

2 “Experimental study of the pressure reset control strategy for energy-efficient fan operation – Part 2: Variable air volume ventilation system with decentralized fans”
Samira Rahnama, Alireza Afshari, Niels Christian Bergsøe, Sasan Sadrizadeh, Göran Hultmark
Submitted to Energy and Buildings, under review

3 “Full-scale experimental study of the static pressure reset control strategy for energy-efficient fan operation in a real building”
Samira Rahnama, Alireza Afshari, Niels Christian Bergsøe
Presented at Healthy Buildings 2017 Europe conference, July 2-5, 2017, Lublin, Poland

4 “Control of VAV ventilation system with decentralized fans”
Samira Rahnama, Amalie Gunner, Alireza Afshari, Niels Christian Bergsøe
Presented at DANVAK DAGEN 2016, Copenhagen

Experiment on the full-scale ventilation plant at SDU is still on-going. The above mentioned conference paper (No. 3) presents the initial results obtained from SDU measurements. At least one more conference paper is planned to be submitted to Indoor Air 2018 conference in which the long term measurement results from SDU will be presented.

The current report has been written based on the above mentioned papers.

Methodology

This section first reviews the existing methods in the literature for energy efficient control of the VAV ventilation system with dampers. The current research project proposes a new method to control the VAV ventilation system with dampers in an energy- efficient manner. The proposed method has been adapted to the VAV ventilation system with decentralized fans. In the following, the proposed control methods for both ventilation systems are explained.
For the sake of simplicity, damper system and fan system are used in the rest of present report denoting the VAV ventilation system with terminal dampers and the VAV ventilation system with decentralized fans respectively.

Energy-efficient control of a damper system

An energy-efficient control of the damper system can be accomplished at two levels, zone-level and system-level control (Maripuu, 2009). Figure 1 shows a simplified damper system with one supply duct connected to a supply fan and four zones.

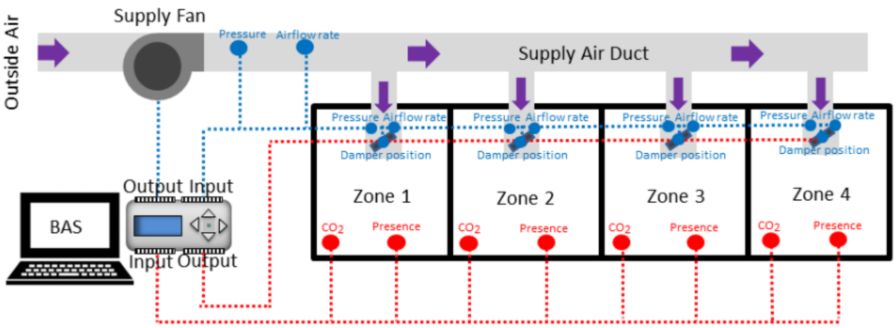


Figure 1: Simplified VAV ventilation system with the terminal dampers (damper system). The automatic control is accomplished at zone-level (red line) and system-level (blue line) through the building automation system (BAS). Red and blue dots indicate the possible measurements in zone-level and system-level control respectively.

Zone-level control

At zone-level, demand controlled ventilation (DCV) can optimize the energy use by customizing the airflow rate to the actual demand, rather than running the system based on a constant occupancy schedule. There are different definitions in the literature for DCV system (IEA, 1992) (EN-13779:2007, 2007) (Sørensen, 2002). A general definition, proposed by Fahlen, is as follows: DCV system is referred to the ventilation system in which the airflow rate is controlled according to the measured demand (Fahlen, Heating, Ventilation and Air Conditioning (HVAC) Systems Engineering, 2008). Different indicators can be used to specify the demand in the zone areas (Maripuu, 2009). For instance, in non-residential buildings, CO_2 concentration can indicate the occupancy. CO_2-based DCV is suitable for zones with highly variable population. For zones with lower variation, occupancy sensor can be used instead.

System-level control

At system-level, the objective is to control the supply fan speed in order to provide sufficient airflow and pressure in the duct system at different demand requirement. A common strategy to control the supply fan speed in a VAV system is to install a pressure sensor in the supply duct system. The idea is to control the fan speed such that the static pressure at the sensor location is held constant, at a specific setpoint. Suppose one of the terminal damper closes to reduce the ventilation demand in one zone. This increases the static pressure at the sensor location. The controller in the building automation system receives the difference between the pressure setpoint and the measured pressure. Then, the controller commands to lower the fan speed to reduce the pressure to the pressure setpoint.

Further energy saving can be obtained by resetting the pressure setpoint to a lower value at partial load conditions. This strategy is known as static pressure reset (SPR) control or critical zone reset strategy in the literature (TRANE, 1991) (PNNL-SA-84187) (Housholder, 2011). In SPR control, static pressure in the duct is controlled such that at least one zone is kept wide-open at any load conditions during operation. A small change in dampers position at partial load conditions will introduce a pressure drop across the dampers. The idea behind SPR control is to keep the dampers as open as possible when the load decreases. This can be accomplished by resetting the static pressure setpoint to a lower value. Practically, the maximum energy saving that can be achieved is when the damper at the critical zone, the zone with the highest pressure drop, is kept wide open at any load condition. Although, the basic idea behind the SPR control is straightforward, implementing the method in practice can be problematic due to interaction between the pressure drop and the damper position.

Several research projects study SPR control in a VAV system. For instance, a study (Shim, Song, & Wang, 2014) categorizes SPR control into two groups, SPR by air handling unit (AHU) feedback and SPR by zone level feedback. In the first group, total airflow rate in an AHU is used for SPR, whereas in the second group, a zone with the highest pressure drop determines the reset strategy. Experimental results show higher energy saving in the first approach compared to the second one, however with a risk of having starved zones. A study (Zhang, et al., 2015) reviews SPR strategies in which SPR based on outside temperature is also introduced other than the two above-mentioned methods. In this method, the static pressure setpoint is simply assumed as a linear function of outside temperature. SPR based on outside temperature is less energy efficient compared to SPR based on airflow rate or SPR based on static pressure and can be applied in combination with them. A comprehensive literature review is provided in (Kimla, 2009). References are classified based on available information on the ventilation system. The study mentions earlier works regarding SPR, for example where there is no feedback from the system or terminal boxes are not digitally controlled. Modern ventilation systems, however, are usually controlled through the building automation system in an automatic manner.

A study (Taylor, 2007) proposes two approaches to implement SPR control which are PID (proportional–integral-derivative) loop on VAV damper position and trim and respond strategy. For the first approach, a PID controller is used to adjust the damper position at 90% open at the critical zone. This requires the knowledge of damper position. The drawback of this method is that a PID loop can decrease or increase the pressure setpoint with a same rate. Decreasing the setpoint should be slow enough to avoid instability, whereas increasing the setpoint should be fast to avoid starving zones. In trim and respond method, pressure setpoint is trimmed regularly until a zone or several zones send a request for more static pressure. In this situation,

the controller responds by increasing the setpoint. This method still requires the knowledge of damper position for sending pressure request. However, analog damper position is not required unlike the PID loop. Moreover, increasing and decreasing of pressure setpoint can be done with different rates.
The current research study proposes a new method to implement SPR control in practice. The method is taken directly from critical zone definition and is based on zone level feedback. However, rather than measuring damper position to identify the critical zone, as suggested in (Taylor, 2007) and (Shim, Song, & Wang, 2014), static pressure can be measured directly at each terminal dampers. Then, unlike the trim and respond method, required decrease in pressure setpoint can be obtained directly as soon as airflow decreases. On the other hand, similar to trim and respond method, increasing the setpoint will be done quickly to avoid starving zones.

SPR control-damper system

In general, the pressure drop in a duct system can be obtained from the following equation:

$$\Delta p = r q^{n} \tag{2}$$

Where Δp [Pa], r and q [m3/s] are the pressure drop, specific resistance and the airflow rate in the duct system, respectively. n is the velocity exponent and it is equal to 2 in a fully developed turbulent airflow. The specific resistance is determined by the dimension and the layout of the duct system, as well as the position of the dampers. Thus, to reduce the energy use, the dampers should be kept as open as possible during system operation. Practically, it is not possible to keep all dampers wide open at any load condition. In SPR control, the damper at the zone with the highest pressure drop is kept wide open to minimize the energy use. This zone is known as the critical zone. In this research study, the static pressure at the terminal damper is measured directly to identify the critical zone.
The basic sequence of the proposed SPR control for a damper system is shown in Figure 2. Assume a damper system with k zones, where p_i (i = 1… k) denote the static pressure at each terminal damper. The reset process is initiated by the change in the total airflow rate. $q(t)$ and $q(t-\tau)$ represent the total airflow rate at the current time and the previous sampling time, respectively. τ is the sample time at which the static pressure and airflow rate are measured. In case of decrease in the airflow rate, i.e. when $q(t) < q(t-\tau)$, the pressure setpoint should be decreased. The decrease in the airflow rate occurs when the demand in one or several zones decreases. This will lead to a change in the relevant damper position to maintain the airflow rate setpoint at the zone level. As a result, the main duct static pressure increases which leads to decrease in the fan speed. Therefore, the change in one damper position will affect the other dampers and it takes some time until the system reaches balance again. Thus, the algorithm waits to allow the zone-level control to adjust the airflow rate at the terminal dampers. After that, the measured static pressures at the terminal dampers are collected to identify the critical zone. The minimum of the measured static pressures, p_{min}, and the relevant damper is detected. The pressure setpoint, p_{set} is reset to $p_{set} - p_{min}$ and the associated damper is set wide open. The algorithm waits again for reaching balance before making decision to repeat the sequence. When $q(t) < q(t-\tau)$ does not hold, the following conditions can happen:

1 $q(t) = q(t-\tau)$ and there is no flow alarm
2 $q(t) > q(t-\tau)$ and there is no flow alarm

3 $q(t) = q(t\text{-}\tau)$ and there is a flow alarm
4 $q(t) > q(t\text{-}\tau)$ and there is a flow alarm

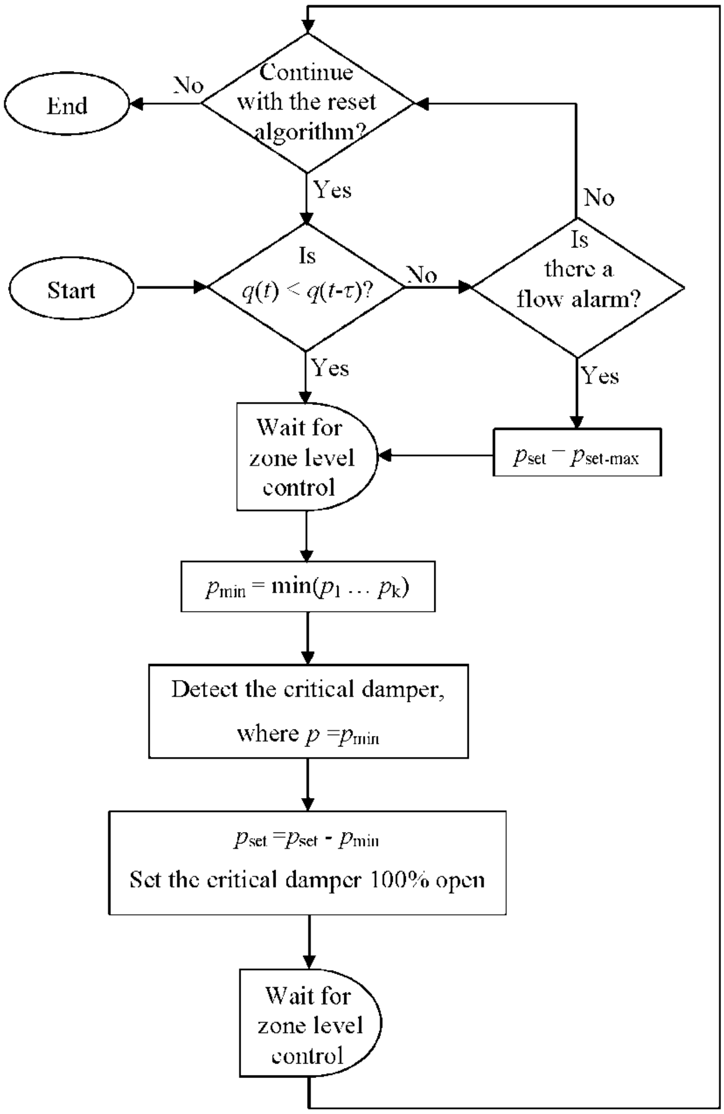


Figure 2: Basic sequence of the static pressure reset (SPR) control based on static pressure measurement for a damper system. The ventilation system is assumed to have k zones and thus k dampers. $p_1 \ldots p_k$ denote the static pressure in the duct before the damper and p_{min} is the minimum of them. $q(t)$ and $q(t\text{-}\tau)$ represent the total airflow rate at the current time and the previous sampling time, respectively. τ is the sample time at which the static pressure and airflow rate are measured. p_{set} denotes the static pressure setpoint at the main fan outlet.

Flow alarm is generated when the airflow rate cannot meet the airflow setpoint within a specified time at the terminal dampers. The first condition is when the airflow at the terminal dampers are same as before and they meet the airflow setpoint. The second condition describes a situation when the total airflow increases however, the current pressure setpoint is still enough to satisfy the demand. The total airflow rate increases when one or several dampers start to open due to further airflow request. This decreases the static pressure in the duct system and thus increases the fan speed. Changing the damper position may not always satisfy the ventilation requirement. For example, when a wide open damper sends request for air, the pressure setpoint should be increased in order to increase the fan speed. In this case, a flow alarm is generated at zone level. The third condition happens when a wide open damper sends request for air from the beginning. In this situation, the total airflow rate is fixed, while there is a flow alarm in the system. The fourth condition is when a partially open damper turns to a wide open damper first and then sends request for air. In this case, the total airflow rate increases, while there is a flow alarm in the system. For the two first conditions, no change is required in the pressure setpoint. For the last two conditions, the setpoint is reset immediately to $p_{set-max}$, being the maximum pressure setpoint at which the maximum demand is satisfied without having a starved zone. The same steps, as described above, are repeated after that. In this way, satisfying the comfort level in the zones has the first priority.

Critical zone

In SPR control of a damper system, the main fan pressurizes the air to ove come the highest pressure drop in the duct system. Static pressure in the duct is controlled such that the terminal damper at the zone with the highest pressure drop, defined as the critical zone, is kept wide-open at any load conditions during the system operation. Other dampers may however be partly closed to regulate the airflow rate in different zones. In the fan system, dampers are replaced with decentralized fans. In this system, the main fan only compensates the pressure of the zone with the lowest pressure drop. The decentralized fans provide the necessary local pressure in the other zones within the duct system. In other words, rather than over pressurizing the air initially to the maximum level and reduce the pressure with the dampers later, the pressure required in each zone is provided with the fan system design. Similar to the damper system design, a zone is defined as the critical zone for the fan system in the present paper. But unlike the damper system design, the critical zone in the fan system is the zone with the lowest pressure drop.

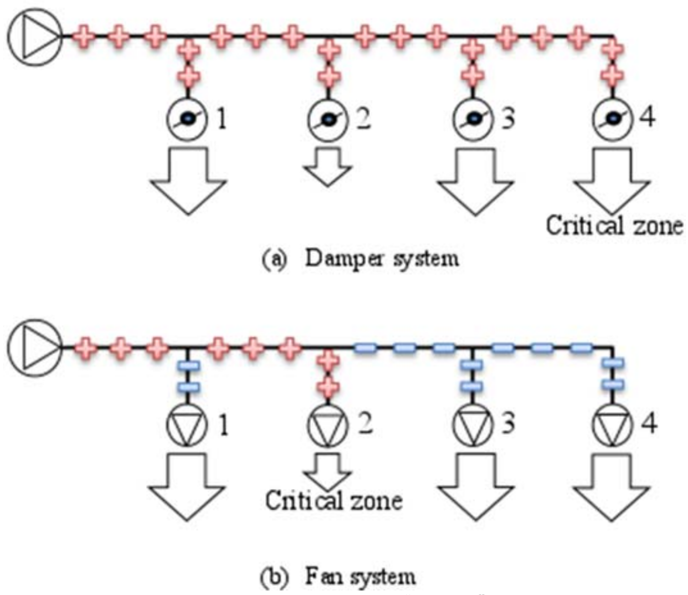


Figure 3: The zone with the highest pressure drop ("zone 4") in the damper system and the zone with the lowest pressure drop in the fan system (zone "2") are considered as the critical zones. In this example, the airflow rate is equal in all zones except zone "2" with the lower airflow rate. The plus and minus signs indicate the places in the duct where there is positive and negative static pressure respectively.

Figure 3 illustrates the definition of critical zone for the two ventilation systems. A simple sketch of the two systems with a main fan and four terminal dampers/decentralized fans is shown in the figure. There is a positive pressure in all ducts in the damper system. In the fan system, there is a negative pressure before the decentralized fans except the fan in the critical zone. In case of equal airflow rates in all zones, zone "4" in the damper system and zone "1" in the fan system are the critical zones in the example shown in Figure 3. However, zone "2" requests a lower airflow rate compared to the other zones in this example. For this operating point, zone "2" may have the lowest pressure drop and become the critical zone rather than zone "1" in the fan system.

SPR control – fan system

Similar to the damper system, an energy-efficient control of a fan system can be accomplished at zone-level and system-level. Zone level control regulates the speed of decentralized fans to customize the airflow rate based on demand controlled ventilation (DCV). In the present research, it is assumed that airflow rate setpoints are given at the laboratory. Thus, the speed of decentralized fans changes to follow the airflow rate setpoints. In a real ventilation system, CO_2 concentration in each zone, occupancy sensors or other desired parameters can control the speed of decentralized fans. System level control, however, regulates the speed of main fan according to total airflow requirement. The SPR control can be applied at the system level and the main fan speed regulated to provide a static pressure setpoint at the fan outlet. The static pressure setpoint is reset based on the sequence shown in Figure 4.

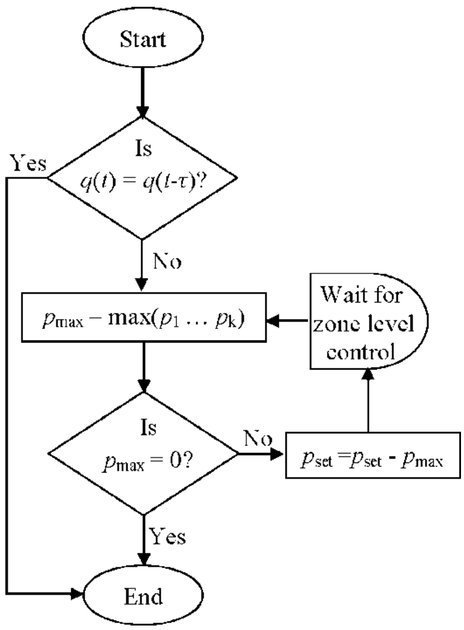


Figure 4: Basic sequence of the static pressure reset (SPR) control based on static pressure measurement for a fan system. The sequence should be repeated every sampling time during the system operation. The ventilation system is assumed to have *k* zones and thus *k* decentralized fans. $p_1 \ldots p_k$ denote the static pressure in the duct before the decentralized fans and p_{max} is the maximum of them. $q(t)$ and $q(t\text{-}\tau)$ represent the total airflow rate at the current time and the previous sampling time, respectively. τ is the sample time at which the static pressure and airflow rate are measured. p_{set} denotes the static pressure setpoint at the main fan outlet.

Fan system versus damper system

The SPR sequence in the fan system in comparison with the SPR sequence in the damper system can be explained as follows:

- The reset process is initiated by the change in the total airflow rate in the same way as the damper system.

- The reset process is based on measuring static pressure at the zone terminal units in both ventilation systems. The pressure setpoint is reset to the current pressure setpoint minus the measured static pressure at the critical zone. However, in the fan system, the maximum of measured static pressure, p_{max} is used according to the definition of critical zone in this ventilation system.

- In both ventilation systems, the main fan pressurizes air to the extent required for the critical zone. This means, no damper or decentralized fan is required at the critical zone. In the damper system, the damper at the critical zone can set wide open which equals to almost no damper situation. In the fan system, however, the decentralized fan should be removed at the critical zone to minimize the energy use. This is not possible in practice since the critical zone is not the same zone during system operation. One can imagine setting the decentralized fan off at the critical zone. But, an off fan can make more pressure drop than an on fan in the duct system. Thus, the speed of decentralized fan at the critical zone is not deliberately set to any value in SPR control of the fan system. Instead, the pressure setpoint reset continues until zero static pressure (or almost zero in practice) provided at the inlet of decentralized fan in the critical zone. In this situation, the decentralized fan works just to eliminate the pressure drop it creates as well as the pressure drop of other components downstream of the decentralized fan in the terminal unit.

- It is possible to make balance in the damper system without applying the SPR control, but the SPR control application can reduce the energy use considerably. In the fan system, applying SPR control is a necessity. Otherwise, there may be a situation in which the system cannot be balanced to satisfy the required demand. In other words, the SPR control of the fan system is a strategy that can be implemented in practice to control the fan system, rather than just a method for reducing the energy use. Thus, the sequence in Figure 4 should be repeated every sampling time during the system operation.

- As shown in Figure 3, there can be both negative and positive pressure at the decentralized fans in each zone. Therefore, the Eq. $p_{set} = p_{set} - p_{max}$ can lead to both decrease and increase in the pressure sepoint, unlike the SPR control of the damper system which considers the increase in setpoint separately (see Figure 2).

Experimental setup

A mock-up of a damper system and a fan system were built at SBi laboratory in Hørsholm, Denmark to evaluate the proposed SPR control and the potential for energy saving. The damper system mock-up was first build and tested at the laboratory. The decentralized fans used at the laboratory were mounted in a steel cubic chamber which made the fan system mock-up different from the first damper system mock-up. In order to have similar setup for comparison, the experiments with dampers were repeated again with having cubic chambers installed in the setup. However, for the experiments with dampers, the fans were removed from the chambers. Thus, two identical mock-ups shown in Figure 5 were tested for the two ventilation systems, except the terminal dampers were replaced with the decentralized fans in the second mock-up.

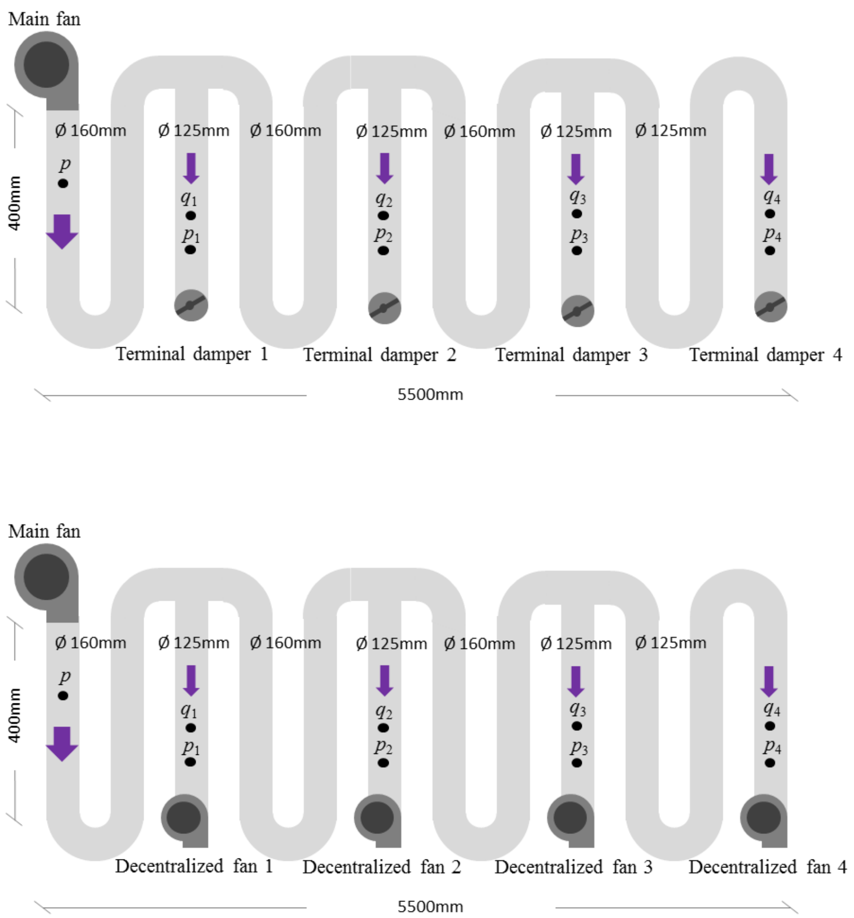


Figure 5: Duct dimensions and the instruments layout of the damper system (top) and the fan system (bottom) mock-ups at Hørsholm laboratory. Black dots are where the static pressure (p, p_1, p_2, p_3, p_4) and the airflow rate (q_1, q_2, q_3, q_4) were measured.

Figure 5 shows the duct dimension as well as the measuring points which are marked with black dots. The measuring points were at minimum five times the hydraulic diameter downstream of the obstacles (Johansson & Svensson, 2007). The static pressure was measured with an advanced silicon based membrane transmitter, FlowGuard 6280 from PSIDAC AB, with a total measurement error of ±0.5 Pa. Measuring range was set at ±125Pa for the branches and ±250Pa for the main duct. The airflow rate was measured with a measuring cross, FRU from Lindab equipped with Belimo VRD3 which provides an output signal (0-10V) proportional to the airflow rate. The meters have an inaccuracy of ±10% within a working area of 0.7-10m/s according to the manufacturer. EC centrifugal fans from Ebm-Papst with backward curved blades with a diameter of 190mm for the main fan and 133mm for the decentralized fans were used. The fan power use was measured with Wireless Electricity Monitor Model efergy e^2 with accuracy greater than 90%. To control the fan speed automatically, the EXOcompact controller from Regin company was used.

Implemented control setup

Figure 6 shows the control setup implemented in Regin EXOcompact controller for the fan system.

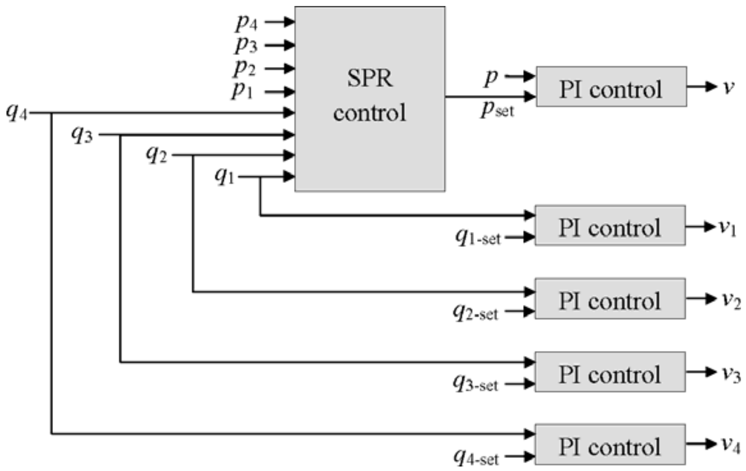


Figure 6: Control setup for the fan system

Four PI (Proportional-Integral) controls were used to control the speed of four decentralized fans. PI controls keep the measured airflow rates (q_i, i = 1…4) at the desired airflow rates ($q_{i\text{-set}}$, i = 1…4). The speed of main fan was also controlled with a PI control to keep the measured static pressure (p) at the static pressure setpoint (p_{set}). Input voltage to the main fan and the decentralized fans are denoted with v and v_i, i = 1…4 respectively. The SPR control updates p_{set} based on the measured static pressure (p_i, i = 1… 4) and the measured airflow rates (providing the total airflow rate q). The terminal dampers were manually adjustable dampers from Lindab with a diameter of 125 mm. Thus, the airflow rates were adjusted manually at the four terminal zones for the damper system. The main fan was controlled with a PI control in the same way as the fan system.

Experimental results

This section presents the measurement results obtained from the damper stem and the fan system tested at the laboratory. The measurement results are provided in three subsections. The first subsection shows the measurements from applying the SPR control on the damper system. These results were obtained from the damper system without having cubic chambers installed in the setup. The second subsection shows the measurements from applying the SPR control on the fan system. This subsection aims to answer the first research question given earlier in Objective section. The last subsection shows the measurements obtained from applying SPR control on both the fan system and the damper system. Here, the aim is to compare the energy use of the two ventilation systems to answer the second research question. The cubic champers were also installed in the damper system while the measurements were recorded for comparison purpose.

Measurement results – SPR control on the damper system

Figure 7 and Figure 8 show the recorded measurements for two separate experiments denoted Experiment 1 and Experiment 2 respectively. The instantaneous measurements are illustrated at three steps for each experiment. The figures show results of measurements of the static pressure and the airflow rate at the terminal dampers, the duct static pressure at the fan outlet and the fan power use. The static pressure setpoints for each step are also shown in the figure. The values on the top (the first step) are the instantaneous values which were recorded while the system was balanced to satisfy a certain demand. One of the zone dampers is wide open in the balanced system which is zone damper "4" in Experiment 1 and zone damper "3" in Experiment 2. The pressure setpoints are p_{set} = 125 Pa and p_{set} = 90 Pa for Experiment 1 and Experiment 2, respectively. As can be seen, there is a small difference, up to 2 Pa, between the pressure setpoint and the measured pressure at the fan outlet. The static pressure at the wide open terminal dampers deviates slightly from zero (p_4 = 5.5 Pa in Experiment 1 and p_3 = 7.6 Pa in Experiment 2). The total airflow has decreased from the first step to the second step (the duct system in the middle) while, the pressure setpoint has been kept constant. The fan power use has decreased by 41% in Experiment 1 and 45% in Experiment 2 from the first step to the second step with constant static pressure at the fan outlet. The total demand has not changed from the second step to the third step, however the static pressure setpoint has reset as follows:

Experiment 1: p_{set} = 126.4 Pa − 53.2 Pa = 73.2 Pa
Experiment 2: p_{set} = 92 Pa − 57 Pa = 35 Pa

where p_{min} = 53.2 Pa and p_{min} = 57 Pa and the critical zones are zone "2" and zone "1" in Experiment 1 and Experiment 2, respectively. The critical zones are wide open in the third step as shown in the Figure 7 and Figure 8. Since the pressure drop in zone "1" and zone "2" in Experiment 2 are almost the same, both zones are wide open in the third step. Applying SPR control has lowered the power use by 27% and 36% compared to the second step in Experiment 1 and Experiment 2, respectively.

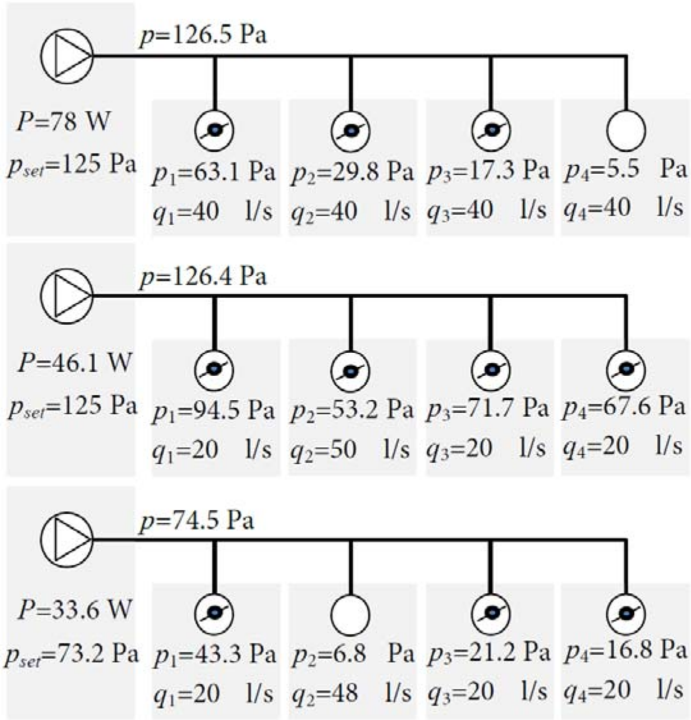


Figure 7: Experiment 1 – The airflow rates, the static pressures at the four terminal dampers as well as the static pressure at the main fan outlet and main fan power use during pressure reset strategy. The airflow rate setpoints were changed from $q_{1\text{-set}} = q_{2\text{-set}} = q_{3\text{-set}} = q_{4\text{-set}}$ = 40l/s to $q_{1\text{-set}}$ = 20l/s, $q_{2\text{-set}}$ = 50l/s, $q_{3\text{-set}}$ = 20l/s, $q_{4\text{-set}}$ = 20l/s

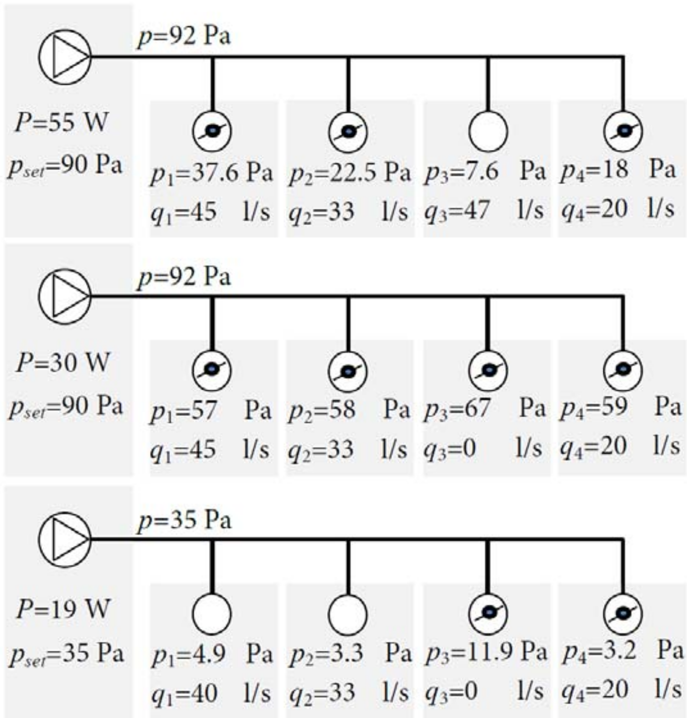


Figure 8: Experiment 2 – The airflow rates, the static pressures at the four terminal dampers as well as the static pressure at the main fan outlet and main fan power use during pressure reset strategy. The airflow rate setpoints were changed from $q_{1\text{-set}}$ = 45l/s, $q_{2\text{-set}}$ = 33l/s, $q_{3\text{-set}}$ = 47l/s, $q_{4\text{-set}}$ = 20l/s to $q_{1\text{-set}}$ = 45l/s, $q_{2\text{-set}}$ = 33l/s, $q_{3\text{-set}}$ = 0l/s, $q_{4\text{-set}}$ = 20l/s

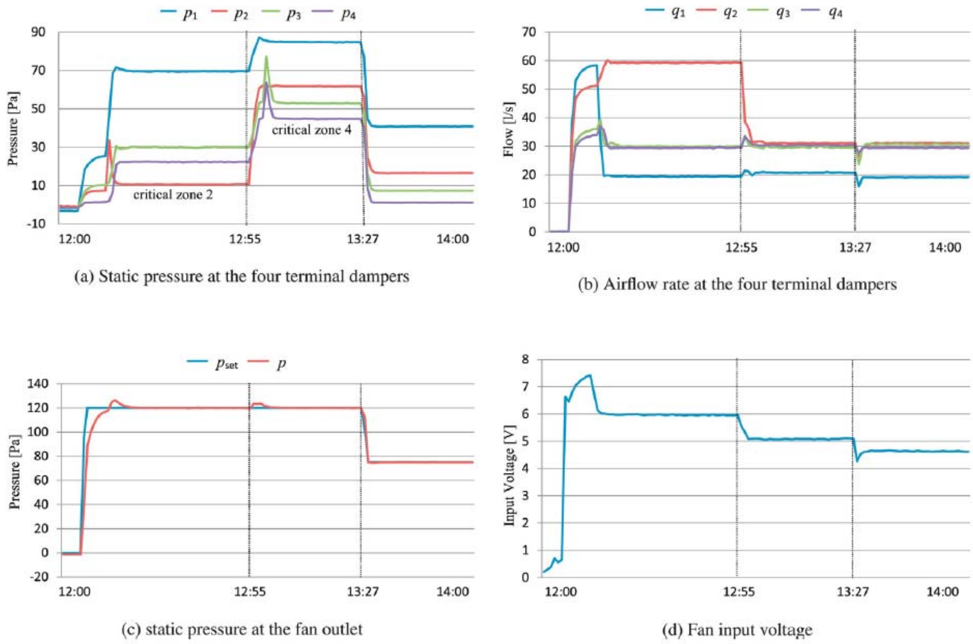


Figure 9: Recorded measurements during a 2-h experiment applying SPR control on the damper system. The airflow rate setpoints were changed from $q_{1\text{-set}}$ = 20l/s, $q_{2\text{-set}}$ = 60l/s, $q_{3\text{-set}}$ = 30l/s, $q_{4\text{-set}}$ = 30l/s to $q_{1\text{-set}}$ = 20l/s, $q_{2\text{-set}}$ = 30l/s, $q_{3\text{-set}}$ = 30l/s, $q_{4\text{-set}}$ = 30l/s

Figure 9 shows the recorded measurements during a 2-h experiment. Then, the transient response after any change can also be seen before the system reaches balance. The measurements were logged every 2 s and the average values over 1 min were calculated and shown in the figure. The system was run from standstill to provide the total airflow rate of 140 l/s. The required pressure setpoint was 120 Pa at this load. The highest pressure drop occurred at zone “2” as seen in graph (a). This zone was wide open. At 12:55, the airflow demand decreased at zone “2” from 60 l/s to 30 l/s. After the system reached balance, zone “4” had the highest pressure drop and became the critical zone. At 13:27, the pressure setpoint was reset to 75 Pa which was obtained as below:

$$p_{\text{set}} = p(13:27) - p_{\min}(13:27) = 121.2\text{Pa} - 46.2\text{Pa} = 75\text{Pa}$$

Figure 9(c) shows the measured static pressure at the fan outlet in which, the performance of the PI controller can be seen. At the beginning, it took 12min until the measured pressure reached the setpoint. However, at the time of pressure reset, the pressure setpoint was followed almost immediately. Figure 9(d) shows the input voltage to the fan from the controller during the three steps. The average power use during these steps are P = 62 W, P = 42 W and P = 33 W, respectively. The power use has decreased by 21% after pressure reset compared to the constant static pressure.

For more elaboration on the critical zone reset method, Figure 10 shows the system characteristic curves in the three steps for the last experiment. The curves are obtained from Eq. (2), where the average value of the pressure drops and the airflow rates in the three steps are used to calculate the related specific resistance. Velocity exponent is assumed n = 2, since the airflow was fully developed at the measuring points in the laboratory setup. The operating point for the fan is the intersection between the system characteristic curve and the fan characteristic curve which are available from the factory tests. Suppose, the fan operates to provide q = 140 l/s at the design condition. The design system curve is marked (1) in Figure 10. The operating point for the fan is point (A) with p = 120.1 Pa and q = 138.2 l/s. The fan speed is n_1 = 3000 rpm at point (A). At the partial load condition (q = 110 l/s), the system curve moves upward due to the change in the position of the dampers. The new system curve is marked (2) and the new operating point for the fan is point (B) with p = 120 Pa, q = 111.9 l/s and n_2 = 2580 rpm. The fan outlet pressure is kept almost constant from point (A) to point (B). Keeping pressure constant at the fan outlet, the operating point for the fan moves along the green dashed line, denoted as fan modulation curve – constant pressure, when the load changes during the system operation.

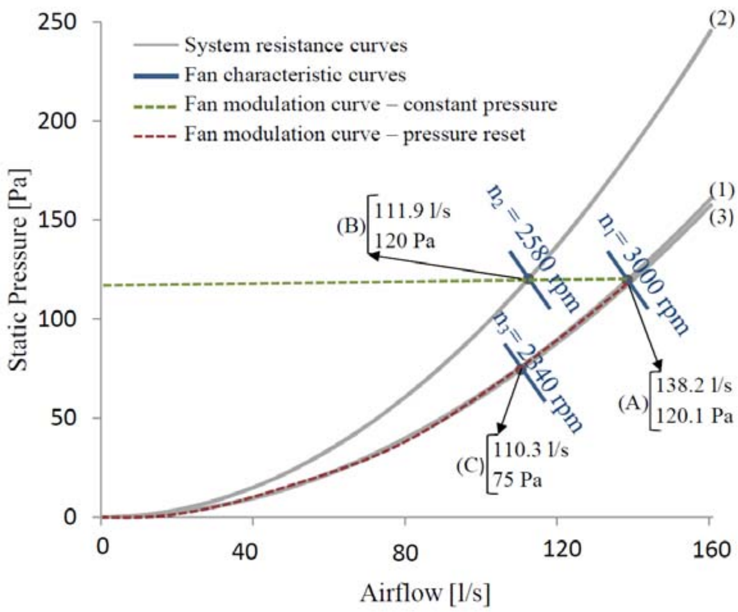


Figure 10: System characteristic curves at the design and partial load conditions

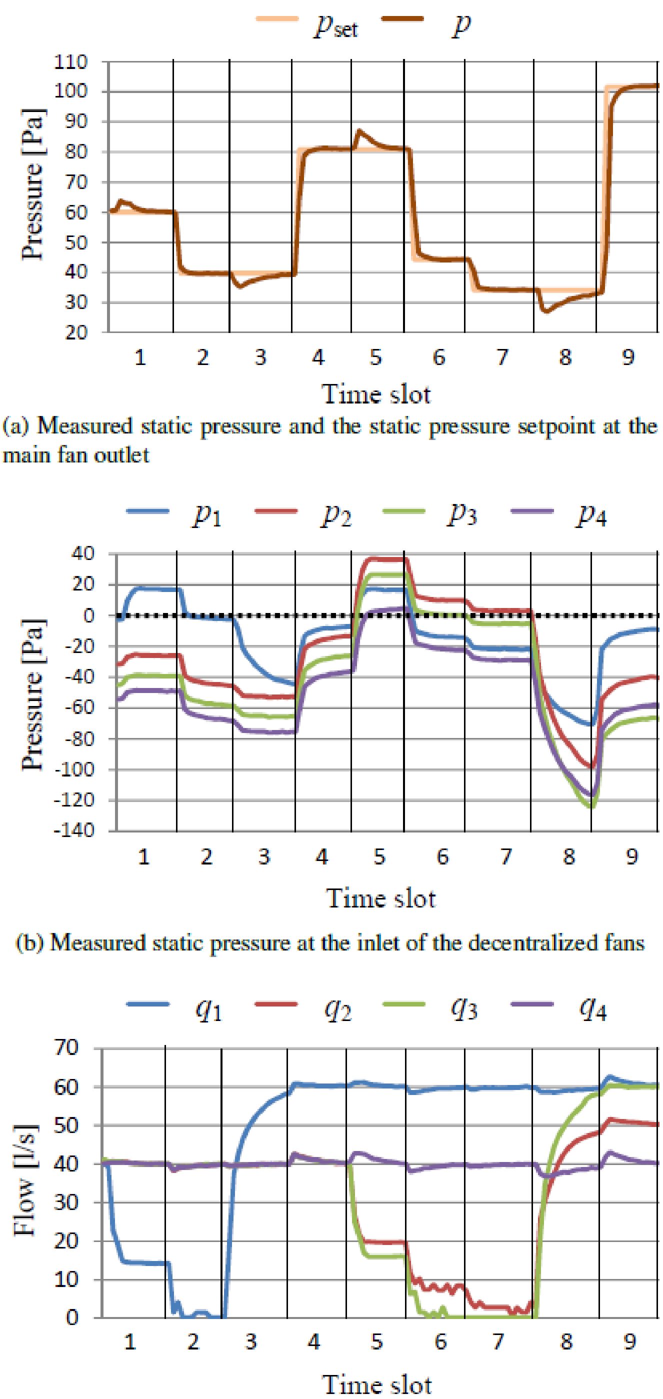
However, the same airflow can be provided at a lower pressure with the pressure reset strategy. The system characteristic curve after pressure reset is marked (3) in Figure 10. As shown in the figure, the system curve after pressure reset is almost identical to the design system curve. Ideally, these curves should be exactly identical. The fan duty point after reset is point (C) with a lower pressure (p = 75 Pa), almost the same airflow (q = 110.3 l/s) and a lower fan speed (n_3 = 2340 rpm). With pressure setpoint reset, the operating point for the fan moves along the design system curve when the load changes. The red dashed line in Fig. 6 shows the fan modulation curve with pressure reset.

Measurement results – SPR control on the fan system

The proposed SPR control setup was applied on the experimental mock-up at the laboratory. The recorded measurements are shown in Figure 11. The measured static pressure as well as the static pressure setpoint at the main fan outlet is shown in Figure 11(a) and the measured static pressure at the inlet of the decentralized fans is depicted in Figure 11(b). The measured airflow rates in each branch are shown in Figure 11(c). The measurements were recorded in nine time slots in which either the airflow rate setpoints or the static pressure setpoint were changed according to the values given in T bale 1. Each time slot lasted 10 min. In a real full-scale ventilation system, the airflow rate setpoints are given values which indicate the ventilation demand in zone areas. However, the static pressure setpoint changes based on the proposed SPR control automatically. In the laboratory experiment, the static pressure setpoint was also set manually to evaluate the performance of the proposed control setup. The initial set values at the beginning of the experiment were:

$q_{i\text{-}set} = 40$l/s (i = 1… 4)
p_{set} = 60Pa

which provided the static pressure around zero (p_1 = 0.5Pa) at the decentralized fan inlet in zone “1”, i.e. the critical zone.



(a) Measured static pressure and the static pressure setpoint at the main fan outlet

(b) Measured static pressure at the inlet of the decentralized fans

(c) Measured airflow rates in each branch (The airflow rate setpoints are given in Tab. 1)

Figure 11: Recorded measurements from the experimental mock-up of a VAV ventilation system with decentralized fans. The measurements were recorded in nine time slots each of duration 10 min.

Table 1: The airflow rate setpoints in the four branches and the static pressure setpoint at the main fan outlet related to the experiment with the measurements shown in Figure 11

Time slot	$q_{1\text{-set}}$	$q_{2\text{-set}}$	$q_{3\text{-set}}$	$q_{4\text{-set}}$	p_{set}
	l/s	l/s	l/s	l/s	Pa
1	0	40	40	40	60
2	0	40	40	40	39.8
3	60	40	40	40	39.8
4	60	40	40	40	80.9
5	60	0	0	40	80.9
6	60	0	0	40	44.3
7	60	0	0	40	34.2
8	60	50	60	40	34.2
9	60	50	60	40	101.6

The ventilation mock-up operation in above-mentioned time slots is as follows:

1 In time slot 1, the airflow rate setpoint in zone "1" changed to $q_{1\text{-set}}$ = 0l/s, while p_{set} remained unchanged. As shown in Figure 11(c), the airflow rate setpoint in zone "1" could not be satisfied with the same static pressure setpoint (60Pa).

2 In time slot 2, the pressure setpoint was reset according to the proposed SPR control as:

$p_{max} = p_1 = 20.2\text{Pa}$
$p_{set} = p_{set} - p_{max} = 60\text{Pa} - 20.2\text{Pa} = 39.8\text{Pa}$

With this change in the pressure setpoint, the ventilation system could satisfy the airflow rate setpoint in all zones.

3 In time slot 3, the airflow rate setpoint in zone "1" changed to $q_{1\text{-set}}$ = 60l/s while p_{set} remained unchanged (39.8Pa). Again, the ventilation system could not satisfy the airflow rate setpoint in the first branch.

4 In time slot 4, the pressure setpoint was reset as:

$p_{max} = p_1 = -41.1\text{Pa}$
$p_{set} = p_{set} - p_{max} = 39.8\text{Pa} + 41.1\text{Pa} = 80.9\text{Pa}$

5 In time slot 5,the airflow rate setpoint in zone "2" and zone "3" changed to $q_{2\text{-set}} = q_{2\text{-set}}$ = 0l/s. As expected, the pressure setpoint reset was required. Otherwise, the airflow rate setpoints were not satisfied in all zones (see Figure 11(c)).

6 The pressure setpoint was reset in time slot 6 as:

$p_{max} = p_2 = 36.6\text{Pa}$
$p_{set} = p_{set} - p_{max} = 80.9\text{Pa} – 36.6\text{Pa} = 80.9\text{Pa}$

This time, the critical zone was zone "2". However, more than one pressure setpoint reset was required to provide zero static pressure (almost zero in practice) at the decentralized fan inlet in zone "2".

7 The pressure setpoint was reset again in time slot 7 as:

$p_{max} = p_2 = 10.1Pa$
$p_{set} = p_{set} - p_{max} = 44.3Pa – 10.1Pa = 34.2Pa$

With the pressure setpoint reset for the second time, the airflow rate setpoints were satisfied with an acceptable deviation in all zones.

8 In time slot 8, the airflow rate setpoint in zone “2” and zone “3” changed to $q_{2\text{-}set}$ = 50l/s and $q_{3\text{-}set}$ = 60l/s respectively.

9 To satisfy the new airflow rate setpoints, the pressure setpoint changed to p_{set} = 101.6Pa based on the SPR control in time slot 9.

Measurement results – fan system versus damper system

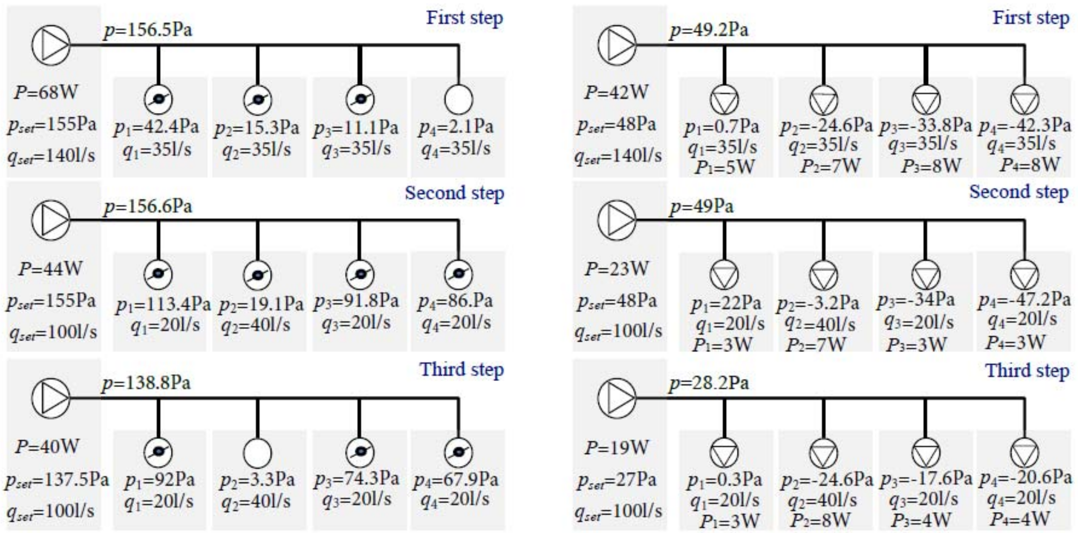
Figure 12 shows the recorded measurements from two separate experiments performed in the laboratory denoted Experiment 1 ((a),(b)) and Experiment 2 ((c),(d)), where (a) and (c) show the measurements from the damper system, whereas (b) and (d) show the measurements from the fan system. The instantaneous measurements are illustrated at three steps. The airflow rate setpoints in one or several zones were changed from the first step to the second step, but the pressure setpoint remained unchanged. In the third step, the pressure setpoint was reset according to the SPR control for the damper system and the fan system. The terminal dampers at the critical zone were set wide open in the first and third steps for the damper system. Likewise, the measured static pressure at the critical zone was near zero in the first and third steps for the fan system.

As noted, it is possible to make balance in the damper system without static pressure reset. However, static pressure reset can reduce the main fan energy use. For instance, the main fan power lowered from 44W to 40W in Experiment 1 (a) and from 31W to 16W in Experiment 2 (c). The amount of power reduction can vary in different operating points and depends on the size of ventilation system though. For the fan system, making balance may not be possible without pressure reset. For instance, in Experiment 2 (d), the airflow rate setpoint in zone “3” ($q_{3\text{-}set}$ = 0l/s) was not met in the second step. The measured airflow rate varied between 0l/s to 13l/s. Thus, for comparing the two ventilation systems from power use point of view, only the first and the third steps are considered. The system was in balance in these steps and the airflow rate setpoints were met in all zones. Zone “4” was only starved 2l/s in Experiment 2 (c) in the third step which is negligible.

Table 2 summarizes the total power use of the two ventilation systems for the two experiments. Except for the total airflow rate of 75l/s, the power use difference between the two ventilation systems is equal or less than 2W. For the total airflow rate of 100l/s, the power use of the fan system is lower, whereas for the rest is higher than the damper system.

Table 2: Power use of the fan system and the damper system in two separate experimnets

	Total airflow rate (l/s)	Total power use damper system (W)	Total power use fan system (W)
Experiment 1	140	68	70
	100	40	38
Experiment 2	115	47	48
	75	16	27



(a) Experiment 1 - Damper system: Zone "4" is the critical zone in the first step. Zone "2" is the critical zone in the second and third steps.

(b) Experiment 1 - Fan system: Zone "1" is the critical zone in all steps

(c) Experiment 2 - Damper system: Zone "3" is the critical zone in the first step. Zone "4" is the critical zone in the second and third steps.

(d) Experiment 2 - Fan system: Zone "1" is the critical zone in all steps

Figure 12 : The airflow rates, the static pressures at the four terminal dampers/Dec. fans and the power use of fans during pressure reset strategy. The airflow rate setpoints were changed from $q_{1\text{-set}} = q_{2\text{-set}} = q_{3\text{-set}} = q_{4\text{-set}} = 35$l/s to $q_{1\text{-set}} = 20$l/s, $q_{2\text{-set}} = 40$l/s, $q_{3\text{-set}} = 20$l/s, $q_{4\text{-set}} = 20$l/s in Experiment 1 and from $q_{1\text{-set}} = q_{2\text{-set}} = q_{4\text{-set}} = 25$l/s, $q_{3\text{-set}} = 40$l/s to $q_{1\text{-set}} = q_{2\text{-set}} = q_{4\text{-set}} = 25$l/s, $q_{3\text{-set}} = 0$l/s in Experiment 2

The following condition should be held in order to reduce the power use and hence achieve energy saving in the fan system:

$$P + \sum_{i=1}^{k} P_i \,|_{\text{fan system}} < P \,|_{\text{damper system}} \tag{3}$$

i.e. the power use of main fan plus the total power use of decentralized fans in the fan system should be lower than the power use of main fan in the damper system. The fan power demand depends on the provided airflow rate and the fan pressure as well as the fan efficiency (see Eq. (1)). To understand the experimental results, Figure 13 shows the main fan characteristic curves and the system characteristic curves at the total airflow rate of 140l/s for the two ventilation systems (Experiment 1 (a) and (b)-First steps).

The main fan efficiency curves provided by the manufacturer are also shown from which the main fan efficiency at the tested operating points is defined. The calculated efficiencies based on measurements are lower than the efficiencies defined from the efficiency curves.

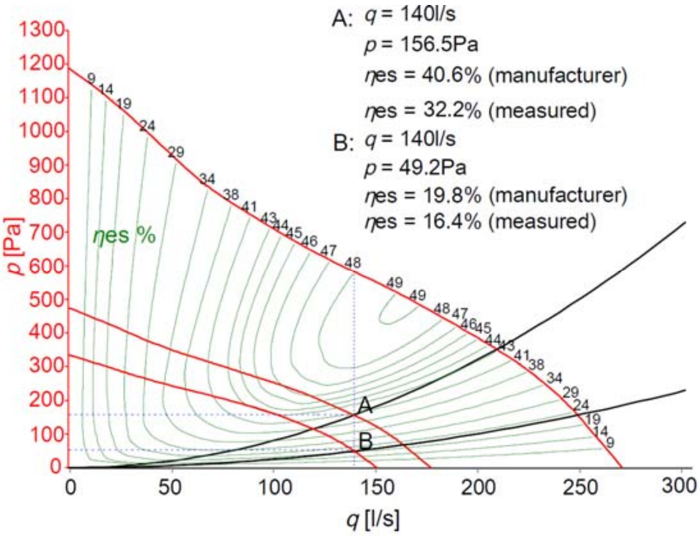


Figure 13: The main fan e_ciency and the main fan characteristic curves (provided by the manufacturer ebmpapst) as well as and the system characteristic curves at the total airflow rate of 140l/s for the damper system (A) and the fan system (B).

For the experiment shown in Figure 13, the measured static pressure at the main fan outlet lowered by around 68% (from p = 156.5Pa to p = 49.2Pa) from the damper system to the fan system at the total airflow rate of 140l/s. However, the fan power demand only lowered by around 38% (from P = 68W to P = 42W). The reason is the decrease in the main fan efficiency (from 32.2% in the damper system to 16.4% in the fan system) which affects the power use according to Eq. (2). The main fan power use would be around P = 21W and the total power use of the fan system would be around $P+P_1+P_2+P_3+P_4$ = 49W if the main fan efficiency remained constant from the damper system to the fan system. This translates to around 28% decrease in the total power use of the fan system compared to the damper system at the total airflow rate of 140l/s.

Figure 14 shows the energy saving potential for the four total airflow rate listed in Table 2, assuming the same efficiency for the main fan in the two ventilation systems. At the total airflow rate of 115l/s and 100l/s, the power use (equivalently, the energy use) of the fan system would be lowered by around 30% and 32%. However, for the total airflow rate of 75l/s, the power use of the fan system would be still higher even with the same main fan efficiency. Negative saving in the figure indicates the higher power use of the fan system compared to the damper system.

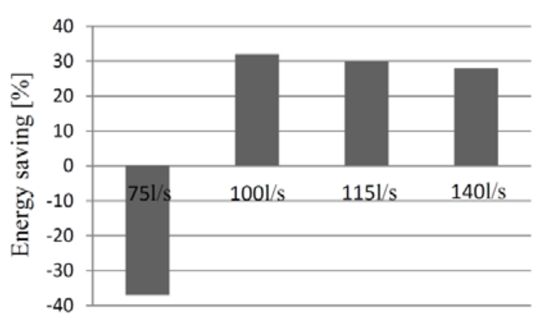


Figure 14: Energy saving potential in the ventilation systems with decentralized fans compared to the ventilation system with dampers assuming the same efficiency for the main fan in both systems.

In addition to the main fan efficiency, the efficiency of decentralized fans also affects the amount of energy saving in the fan system. As mentioned above, the main fan basically satisfies the demand at the critical zone according to the SPR control. The decentralized fan at the critical zone just works to overcome the pressure drop across the fan, i.e. a little pressure drop which was no measured at the laboratory. Thus, the efficiency of decentralized fan is rather low at the critical zone. Apart from the decentralized fan at the critical zone, the higher the decentralized fans efficiency at other zones, the greater the energy saving.

The maximum efficiency calculated for the decentralized fans based on experimental measurements is 26.7% (Dec. fan 4 providing 35l/s airflow rate in Experiment 1 (b)). At the total airflow rate of 75l/s (Experiment 2 (d)-Third step), the efficiency of decentralized fans is rather low. The decentralized fans for zone “2” and zone “4” had the efficiency of 3% and 8% respectively at the tested operating points. For zone “3”, the airflow rate setpoint is zero, i.e. no airflow request at this zone. The measured airflow rate was zero with the available measuring instrument at the laboratory. In fact, the actual airflow rate was insignificant and almost zero. This shows the possibility of having zero airflow rates at one zone with the fan system even if there is no terminal damper. However, this decentralized fan also worked at a low efficiency which cannot be calculated based on measurements from the available measuring devices at the laboratory.

Full-scale experimental study

In the laboratory setup, there were no real zone areas and the airflow rates were assumed to be known at each terminal damper. In order to evaluate the proposed SPR method in a full realistic ventilation setup and to investigate the potential for energy savings in the long term, the SPR control has been implemented in a real VAV system serving three classrooms in Southern University of Denmark (SDU).
Figure 15 shows a simplified picture of the VAV ventilation system at the SDU campus. Classrooms 501 and 502 have the designed airflow rate of 1440 m^3/h and classroom 503 has the designed airflow rate of 2218 m^3/h. In the automatic mode, the ventilation plant starts and stops via passive infrared sensors (PIR) in the zones.

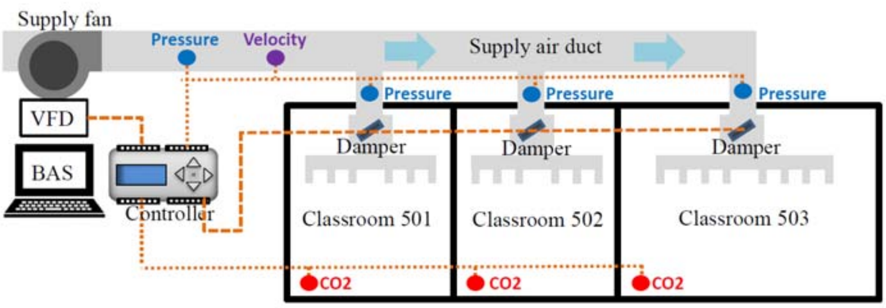


Figure 15: Simplified picture of the VAV ventilation system at SDU. The supply fan is controlled via a variable frequency drive (VFD) through the building automation system (BAS). The dotted and the dashed lines show the measuring and control signals respectively.

The existing ventilation plant was running as follows: The supply fan was controlled to keep a constant static pressure setpoint at the fan outlet. The pressure setpoint was either zero (when the system was off) or a constant value, i.e. no reset strategy had been applied for the ventilation system. At the zone-level, the local dampers were controlled based on CO_2 sensors in the classrooms. The dampers were opened to the maximum level when the measured CO_2 concentration was greater than a desired value.

In the first step, the SPR control of the damper system was implemented at SDU. Other than the existing pressure sensor installed on the main supply air duct, three additional pressure transmitters have been added before the terminal dampers in each classroom in order to implement the SPR control. Moreover, an energy meter, a velocity meter and three indoor climate meters (IC meter) have been installed to log the energy use of supply fan, the total air velocity and the CO_2 concentration in the classrooms respectively.

The basic sequence of the proposed SPR control is shown in Figure 16. The SPR algorithm checks whether there is a decrease or increase in the total air velocity (or equivalently total airflow rate) compared to the previous sample time. In case of decrease in the air velocity and when the difference is greater than dv, the static pressure setpoint p_{set} is reset to a lower value as p_{set} – K p_{min} where, p_{min} is the minimum of static pressures measured before the terminal dampers in each classroom and K is a constant value. The associate damper, in which the static pressure is equal to p_{min} is also set to 90% open. When $v(t) - v(t\text{-}\tau) < -dv$ does not hold, no change is required in the static pressure setpoint, unless the CO_2 concentration level violates the maximum value, C_{max} in at least one of the classrooms. In this case, the setpoint is reset immediately to the $p_{set\text{-}max}$, being the maximum pressure setpoint at which the maximum demand is satisfied without having a starved zone. Then the same reset strategy is repeated after a waiting period. Except in a few points, the basic sequence is same as the SPR control implemented at the laboratory (see Figure 2). At the University campus, the critical damper is set to 90% open after the pressure reset, rather than 100% open at the laboratory environment. This reduces the chance of having a starved zone. Another practical solution to avoid starved zones is to subtract a percentage of p_{min} from the current pressure setpoint in the reset strategy, i.e. $p_{set} = p_{set} - \mathrm{K}\ p_{min}$. To avoid the frequent pressure reset in the ventilation system, a constant value, dv, can be considered. The constant values in this study are as follows: K = 1, dv = 0, $p_{set\text{-}max}$ = 25 Pa and C_{max} = 1000 rpm.

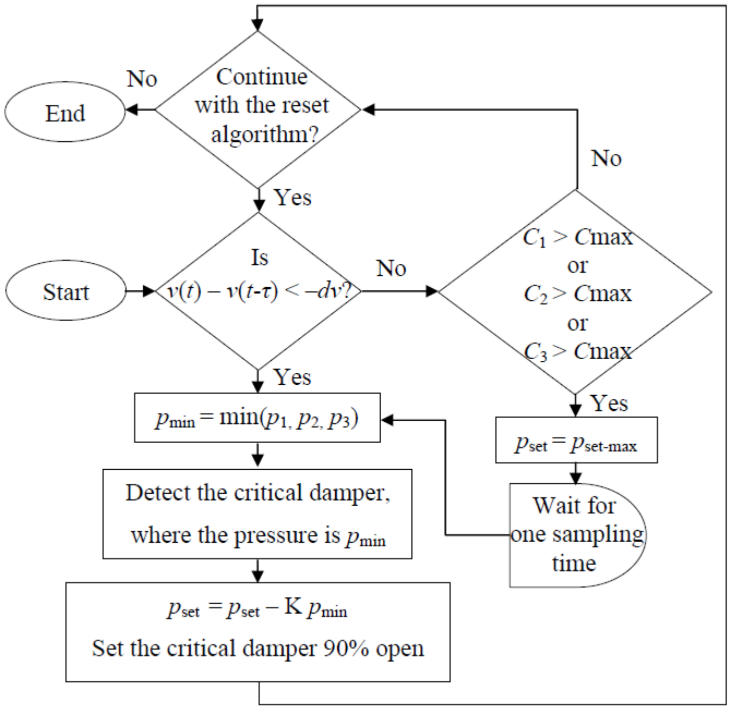


Figure 16: p_i and C_i (i=1,2,3) are the measured static pressure before the terminal dampers and the measured CO_2 level in the three classrooms respectively. v is the measured total air velocity in the main duct. p_{set} and τ denote the static pressure setpoint and the sample time at which the measurements were taken respectively. K, $p_{set\text{-}max}$, C_{max} and dv are constant values.

Ventilation system measurements at Southern University of Denmark

Figure 17 shows the main duct static pressure measurements for one week before and one week after the SPR control implementation. The measurements were carried out from Monday 8:00 until Sunday 23:50 in November 2016 and January-February 2017. Before the SPR implementation, the static pressure setpoint was held constant at 25 Pa when the system was running. The measured static pressure varies around 25 Pa as can be seen in the top graph of Figure 17. The measured static pressure is around 3 Pa most of the time when the system is off. After the SPR control implementation, the static pressure setpoint changes according to the sequence described in Figure 16. As can be seen in the bottom graph of Figure 17, the measure static pressure varies around values below 25 Pa, which indicates the lower static pressure setpoint than 25 Pa.

The CO_2 concentration measurements in the three classrooms for the same periods are also shown in Figure 18 in order to examine the indoor air quality before and after the SPR control implementation. According to the Danish

Building Regulations 2015 (BR15) the CO_2 content of the indoor air must not exceed 0.1 % for extended periods. Exceeding the limit for short periods is acceptable according to BR15. As can be seen in Figure 18, except in a few points, the CO_2 concentration level was kept below 1000 ppm for the whole periods before and after the SPR control implementation. In fact, the CO_2 concentration level after the SPR control is lower than the CO_2 concentration level before the SPR control. Figure 19 shows the position of terminal dampers (percentage of openness) in the three classrooms after the SPR control. As expected from the sequence described in Figure 16, at least one of the terminal dampers was 90 % open (known as the critical damper) at any time during the system operation. For example, the terminal damper in classroom 503 was 90 % open on Monday morning, whereas, on Monday afternoon, the terminal damper in classroom 501 was the critical damper. This is also expected according to CO_2 concentration levels for these periods shown in Figure 18.

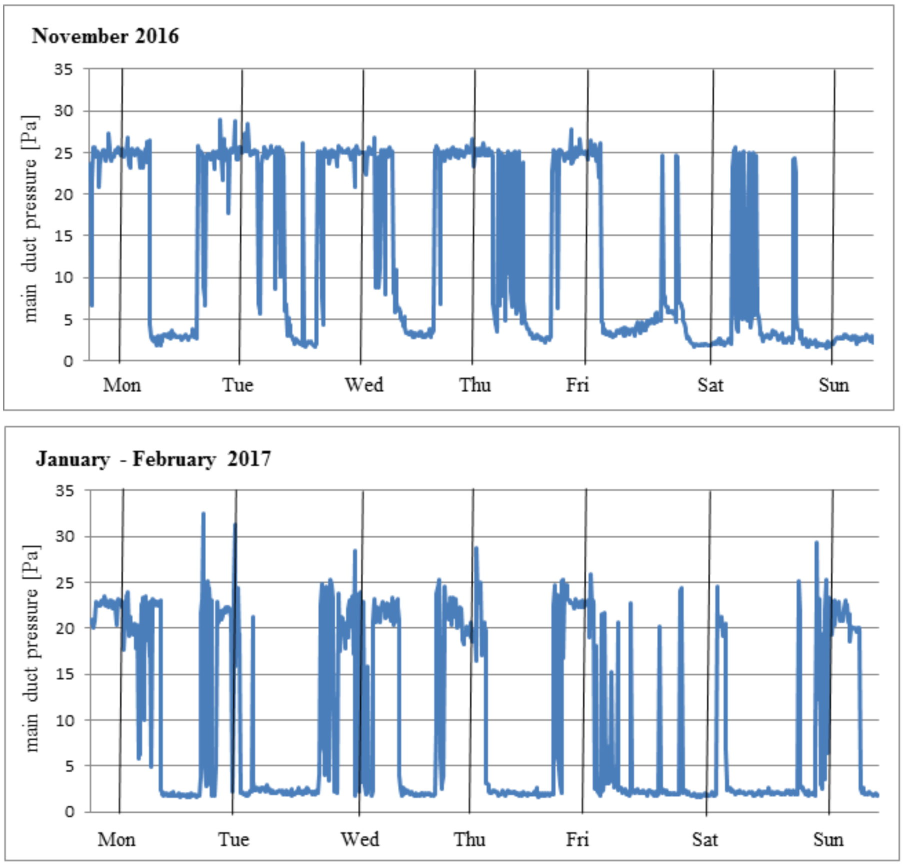


Figure 17: Main duct static pressure measurements before the SPR control (on the top) and after the SPR control (at the bottom)

Total energy uses of the supply fan for the two aforementioned one-week periods are listed in Table 3. As can be seen, the total energy use after the SPR control implementation is greater than the total energy use before the SPR control implementation. However, the total provided airflow rate is also greater during the period after the SPR control implementation, but relating the total energy uses to the total provided airflow rates for the two periods indicates lower average energy use per 1 m^3/h provided airflow rate after the SPR control implementation.

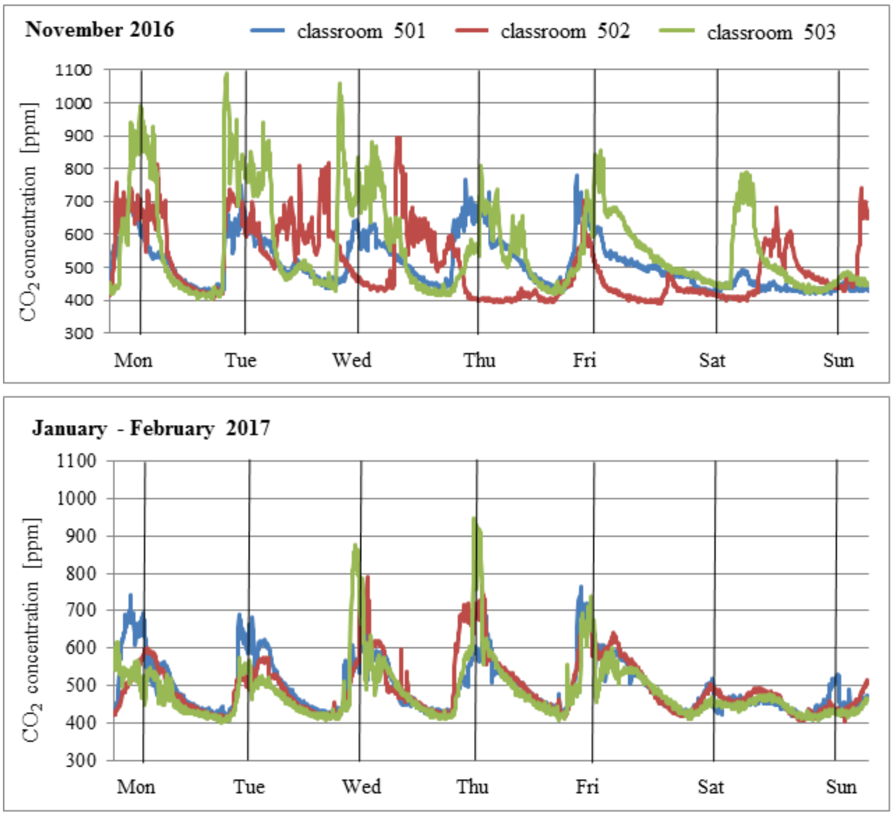


Figure 18: CO_2 concentration measurements before the SPR control (on the top) and after the SPR control (at the bottom) in the three classrooms

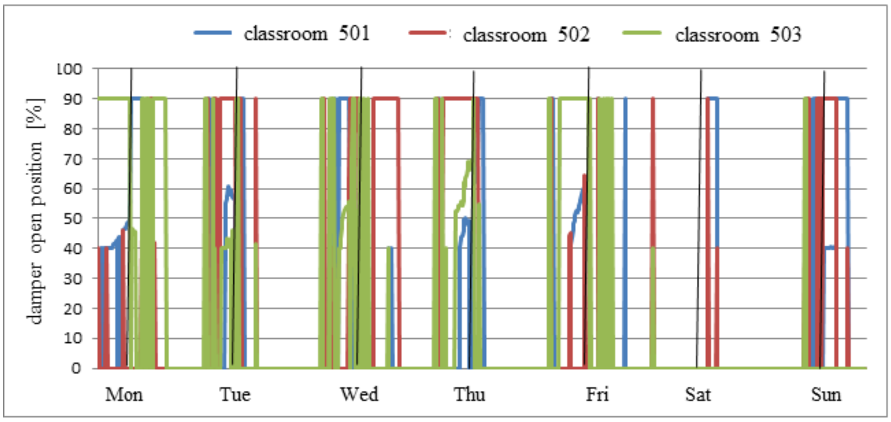


Figure 19: Terminal damper open position in the three classrooms after the SPR control

Table 3: Energy use of the supply fan during two periods of one week each at SDU

	Without SPR	With SPR
Total energy use of the supply fan	11.12 kWh	12.12 kWh
Total provided supply airflow rate	618626.70 m³/h	690982.18 m³/h
Average energy used per 1 m³/h airflow rate	1.80×10^{-5} kWh	1.75×10^{-5} kWh

Conclusion

The present research project has studied a novel VAV ventilation system in which terminal dampers are replaced with decentralized fans. The purpose of the study was to investigate two issues:

1. The possibility of making balance in such a ventilation system, i.e. to control the speed of the main fan and the decentralized fans in relation to each other such that the ventilation demands are satisfied in all zones.
2. The potential for energy saving in comparison with a VAV ventilation system with terminal dampers.

The conventional VAV ventilation system with terminal dampers was studied first as a reference for the comparison purpose. The outcomes are as follows:

– A practical method was proposed to apply the static pressure reset (SPR) strategy based on measuring the static pressure at the terminal dampers to energy efficient control of the conventional ventilation system. Implementing the proposed strategy is straightforward in practice. The pressure instruments can be added to an existing ventilation system without requiring any substantial change.

– The experimental results from the five separate experiments have shown a reduction of 48%, 36%, 27%, 21% and 9% in fan power use with pressure reset compared to constant static pressure in different total airflow rate. Thus the amount of power reduction can vary in different operating points and also depends on the size of ventilation system.

– As shown in the experimental results, there is a risk for having starved zones with the proposed SPR strategy for the VAV ventilation system with dampers. A practical solution can be to consider a permanent safety factor in applying the pressure reset. Rather than subtracting p_{min} from the current pressure setpoint, a percentage of that can be subtracted. This reduces the risk for having starved zones, but also the amount of energy saving.

– The SPR control proposed for the VAV ventilation system with terminal dampers has been adapted to the VAV ventilation system with decentralized fans.

– The experimental results have indicated the ability of the proposed SPR control to make balance in the ventilation system with decentralized fans. It was even possible to have zero airflow rates in one zone without using a damper as shown in the laboratory experiments.

– The experimental results from four pair of experiments providing four different total airflow rates have shown almost the same power use for both ventilation systems, except in the lowest tested airflow rate (75l/s). This happened while the same main fan was used in both systems, meaning a less efficient main fan in the novel ventilation system at the tested operating points. However, calculations have shown the potential for energy savings.

– Calculations have shown an average reduction of 30% in power use with the ventilation system with decentralized fans when having the same efficiency for the main fan in both ventilation systems. Thus, a right choice for the main fan will lead to energy saving in the novel ventilation.

– In addition to the main fan efficiency, the efficiency of decentralized fans also affects the amount of energy saving in the fan system. Energy saving is not achieved in rather low airflow rates even with using the main fan with the same efficiency as the damper system (for example, for the total tested airflow rate of 75l/s in the present research). A minimum airflow rate is required in order to obtain energy saving in the fan system. This aspect needs to be further investigated in the future work.

– The SPR control of the VAV ventilation system with terminal dampers was implemented on a ventilation system serving three classrooms at a university campus (SDU) in Odense, Denmark. The measurements taken during two separate weeks, one week when the system was running based on the constant static pressure strategy and one week when the system was running based on the SPR control strategy, were studied. The measurement results indicated that both control strategies can provide an acceptable indoor air quality according to BR15 in terms of CO_2-content in the indoor air while the SPR control strategy uses less energy for providing the required airflow rate on average. Although applying the SPR control strategy has reduced the average energy use, the measurements carried out was not sufficient to quantitatively compare the two control strategies from energy efficiency point of view. The measurements covered rather short periods (one week each) and the measurements were taken in two different times of a year. Experiment on the full-scale ventilation plant at SDU is still on-going. The future papers will cover long-term measurements in which the effect of seasonal change can be investigated.

Future work

– Three decentralized fans were installed at the three classrooms in SDU and the SPR control has been implemented for the ventilation system with decentralized fans. The existing terminal dampers were set wide open to be inactive in the plant. Taking measurements from the SDU ventilation plant is still ongoing. Up to the time of writing this report, CO_2 concentration measurements from the classrooms indicates an acceptable indoor air quality. A long-term measurement is required to compare the energy use of the system with the VAV ventilation system with terminal dampers. This is considered as the future work.

– The present project has evaluated the performance of the VAV ventilation system with decentralized fans through a particular experimental mock-up. The potential for energy saving in such a ventilation system is considerable, but it can vary from system to system. The amount of energy saving depends on the size of the ventilation system, the airflow rates etc. Future works can study the involved parameters which have influence on the amount of energy saving.

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