Air - for Health and Comfort, An Analysis of HVAC Systems' Performance in Theory and Practice

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Department of Building Physics

LUND INSTITUTE OF TECHNOLOGY
Lund University

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Fredrik Engdahl

Department of Building Physics
Lund University
Box 118
S-221 00 Lund
Sweden

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Abstract

One part of the objective of this study is to analyze how different ventilation systems perform in practise when it comes to supplying and exhausting designed air flow in different outdoor and indoor conditions. The other part is to analyze the design criterias and the energy use of a variable air volume system (VAV) based on controlled static pressure at the branch duct level and supplying outdoor air only. To investigate the technical status of ventilation systems, the result from the compulsory testing and examination of ventilation systems (OVK) is used. A multi-zone model (COMIS) based on the mass balanced equation is used to study the air flow in a multi-family building. Two ventilation systems are analyzed combined with the same building; the mechanical exhaust air and the mechanical supply and exhaust air system. Fundamental pressure loss equations and a computer program (PFS) are used to determine how much the air flow at the air terminals on a branch with controlled static pressure differs compared to the designed air flow depending on duct design. A model is developed and tested to determine the optimal supply air temperature with respect to heating, ventilation and air conditioning (HVAC) energy use. The energy use for a VAV system only using outdoor air dependent on control strategies of the supply air temperature and average U-value of the building envelope is analyzed with climate data from northern Europe. An average of 34% of the studied ventilation systems performed as intended. Both the mechanical exhaust and the mechanical exhaust and supply air system showed sensitiveness to outdoor temperature, building airtightness and wind. When the static pressure is controlled to be constant at the branch duct level it is possible to vary the air flow to different zones without measuring the individual flow and without significantly influence the air flow to other zones. When using 100% outdoor air in a VAV system, the indoor air quality will be improved during most of the year. There is great potential in controlling the supply air temperature optimally to reduce the HVAC energy use. The optimal U-value of the building envelope in an energy use perspecive is most often 0 W/(m²·°C) for northern European countries.

Keywords: Ventilation, buildings, infiltration, variable air volume, VAV, indoor climate, indoor air quality, thermal climate, energy, HVAC, optimization, U-value.

This thesis consists of an introductory paper together with research articles. The articles are listed below and will be referred to in the text as Paper I to Paper V.

I	F. Engdahl, Evaluation of Swedish Ventilation Systems, Building and Environment, Vol. 33, No. 4, pp. 197-200, 1998.
II	F. Engdahl, Stability of Mechanical Exhaust System, Indoor Air, No. 9, pp. 282-289, 1999.
III	F. Engdahl, Stability of Mechanical Exhaust and Supply Systems, Proceedings of the 7th International Conference on Air Distribution in Rooms, pp. 1207-1212, 2000.
IV	F. Engdahl, A. Svensson, Pressure Controlled Variable Air Volume System. Submitted to Energy and Buildings, 2002.
V	F. Engdahl, D. Johansson, Optimal Supply Air Temperature with Respect to Energy Use in a Variable Air Volume System, Submitted to Energy and Buildings, 2002.

Preface

This work was initiated by Professor Anders Svensson in 1995 at the department of Building Physics, Lund University, Sweden. Professor Anders Svensson introduced me to this research field and has guided me through the work. I thank him for his support and sharing of knowledge.

I am grateful to the financiers of the project. The financiers are The Swedish Council for Building Research (BFR, Formas), The Development Fund of the Swedish Construction Industry (SBUF), Byggrådet (Föreningen för samverkan mellan byggsektorn och högskolorna) and Föreningen V. Special thanks to BFR, Byggrådet and Åke och Greta Lissheds stiftelse for financing my research period at Lawrence Berkeley Laboratory, USA.

There are several persons that I have been working with and I want to thank Professor Arne Elmroth for supervising and always giving an extra angle to the problem. Dennis Johansson for interesting discussions and a good co-operation. Lilian Johansson for the work with the layout and the figures in the thesis. Stephen Burke for correcting my English and making it understandable. Professor Lars Jensen for dynamic pressure discussions.

I am grateful to the people working at Stifab Farex for their understanding and support during the last period of this work. I thank Göran Engdahl for sending the project ad to Spain and for enlightening me on the practical and economical aspects of ventilation systems. Finally I thank Linda Ahnfeldt for marrying me, though I look in ceilings for air diffusers.

Lund in April 2002
Fredrik Engdahl

Contents

1. Introduction

The perceived indoor climate can be divided into thermal climate, acoustic climate, air quality and visual climate. Demands on the indoor climate are stated in standards [1] and national regulations [2], which also give required minimum amounts of supply air in relation to activity. The air exchange made by the ventilation system to improve the indoor air quality (IAQ) must have a low influence on the other parts of the indoor climate. At a certain amount of supply air flow the IAQ depends on parameters such as the emissions from the building and its interior, the outdoor air quality and other pollutant sources. In some standards these parameters are used as input when designing the supply air flow.

An average person spends 90% of his or her lifetime indoors [3]. During this time the person is exposed to the indoor climate of a variety of buildings. Each indoor climate may have different effects on humans; some may have negative effects on human health. Symptoms [4] such as mucous membrane irritation, dryness, headache and fatigue are often related to the indoor climate. The appearance of such symptoms among many persons in the same building is often identified as "Sick Building Syndrome" (SBS). Two common denominators in buildings with problems are insufficient ventilation [4] and a buildup of moisture.

A great deal of research has been done on the effect of IAQ on human health and performance. Milton et al. [5] studied the short-term sick leave of 3700 employees in 40 buildings depending on supply air flow. The supply air flow was rated as moderate at 12 l/(s·person), and high at 24 l/(s·person). Those in the study who were exposed to moderate outdoor supply air flow had an attributable risk of short-term sick leave of 35% compared to those who were exposed to high supply air flow. Fanger and Wargocki et al. [6, 7] showed that for each twofold increase in supply air flow, the performance of office workers improved on average by 1.7%. The air flow were in the interval 3-30 l/(s·person). The temperature (22°C), the relative humidity (40%) and all other environmental parameters remained unchanged. The study result also showed that an increased outdoor air supply rate reduced the percentage of subjects who expressed dissatisfaction with the air quality, increased the perceived freshness of air, and decreased the sensation of dryness of mouth and throat. The increased supply of outdoor air also eased difficulty in thinking clearly and made the test persons feel generally better. Weschler and Shields [8] showed that there is a reaction among indoor pollutants that generates reactive and irritating products. For this reaction to occur there must be sufficient time for the pollutants to interact. The ventilation efficiency and supply air flow determine the time available for such interactions; that is, the supply air flow influences indoor chemistry. A higher supply air flow rate not only decreases the indoor pollutant concentrations, it also limits reactions among indoor pollutants.

Another study [9] on multi-family buildings in Sweden investigating the indoor climate indicated that many users experience a poor acoustic climate because of high noise levels from the ventilation system and a poor thermal climate because of draft. In the study, IAQ problems such as stuffy and dry air are also pointed out.

The IAQ has an important effect on humans well-being and it is dependant on the pollution sources and on the supply air flow. The best way to improve the IAQ is to reduce the pollution sources. As they can not be totally removed, the supply air flow is of great importance.

As supply air flow counts air coming through air terminals and air leaking from the outdoors through the building envelope into the building, i. e. infiltration. Therefore, not only the designed value of the supply air flow coming through air terminals decides the actual supply air flow, but also the building construction and its interaction with the outdoor climate.

Increasing the supply air flow in a specific ventilation system and building would improve the IAQ, but also have the effect of increasing the energy use. The energy use has always to be taken into account when designing the supply air flow, the ventilation system and the building.

1.1 Ventilation systems

A ventilation system is installed in a building to supply and exhaust air flow in order to remove pollutants. The transport of air from the outdoors to the indoors and then back out again can be achieved in several ways, using different ventilation systems. Natural ventilation is the oldest system used to ventilate buildings, and it is still used in new buildings to some extent. With natural ventilation the air is supplied through large ducts or leakage in the facade. The air is driven through the building via ducts and by thermal forces and wind forces. The forces are relatively weak and therefore the system requires large ducts with low pressure losses. One of the advantages of natural ventilation is that it is silent.

The mechanical exhaust air system is commonly used in residences. In this system, the air is supplied by slots in the facade, often placed somewhere around the windows. The air is then extracted by a fan via a duct system from rooms such as kitchens and bathrooms. When introducing the mechanical exhaust air system, the exhaust air flow can be controlled. The fan makes possible the use of smaller ducts, thereby increasing the area available for the building owner to lease out. The only way to recover energy from the exhausted air is to use a heat pump that heats the hot service water and sometimes also the water in the heating system.

The mechanical supply and exhaust ventilation system is used to make preconditioning of the supply air possible, and to control both the exhaust and the supply air flow. In this system, the air is supplied by a fan via ducts and through an air terminal before entering the ventilated space. An exhaust fan exhausts the air via a duct system. The system also makes it possible to recover heat from the exhaust air to the supply air which reduces the energy use for heating. To reduce the risk of persons experience draft, the supply air can be preheated. Usually the air enters the ventilated space at ceiling level, and to create mixing ventilation it enters at a high velocity. The alternative is to supply the air at floor level at a low velocity and at a temperature slightly lower than the room temperature. This is called displacement ventilation. To make displacement ventilation work all seasons, the supply air has to be cooled in the air-handling unit during warmer periods. One of the disadvantages of the mechanical supply and exhaust air system is that the supply air ducts and fan require more space compared to the mechanical exhaust system. The investment cost for this system is higher compared to the mechanical exhaust air system.

In many buildings there is a need not only for ventilation but also for cooling the building to improve the thermal climate. The thermal climate has an even greater effect on human performance [10] than does the IAQ. Therefore, there are systems that not only ventilate but also increase the thermal comfort by supplying air at a lower temperature than the room temperature. The most common system using air to cool buildings is the variable air volume (VAV) system. The VAV system has a unit at every zone that measures and controls the supply air flow depending on room temperature. The unit consists of a controller, a pressure sensor, a damper and an electric motor that moves the damper. In the room there is a wall module that inculdes a temperature sensor for measuring of the room temperature. The user can set its desired room temperature at the wall module, this is called individual temperature control. The air flow required to uphold the IAQ stated in the standards is often not enough to cool the building, and therefore the VAV system uses higher air flow rates than conventional systems resulting in higher space requirements. One alternativ to the VAV system is a system where instead of air, water is used as an energy carrier. The water is chilled and then distributed by a piping system to the rooms where the water is heated by the room air and then transported back to the chiller. The energy per volume unit that can be carried by water is higher than for air, therefore this system requires less space. The building is then ventilated by a mechanical supply and exhaust air system.

1.2 Energy and ventilation

The price that has to be paid for indoor air quality and thermal comfort is the investment of the building and its installations, the cost of maintenance and the cost of energy use. The cost of energy use is measurable not only in monetary terms, but also with respect to the global environment. The energy use in a heating, ventilation and air conditioning (HVAC) system can be divided into two parts. One part is the electrical energy required to run the fans used to transport the air. The other part is energy used to precondition the air, either heating or cooling it. The preconditioning energy is either in the form of a tempered medium or electricity. In some systems, dehumidifying or humidifying of the supply air may also take place, but this is not common in northern European countries.

In most countries where the VAV system is used, the outdoor climate is warmer and more humid than the indoor climate. Using 100% outdoor air as supply air would result in considerable chiller energy use. Therefore the exhaust air is returned to the HVAC unit; this is called return air. If the VAV system were to use 100% outdoor air, the increased air flow would have a positive effect on the IAQ. Using return air has a negative effect on the IAQ in parts of the building where the IAQ is higher than the average of the building. In northern Europe the outdoor temperature is usually lower then the indoor temperature and therefore return air would not significantly reduce the energy use.

The supply air temperature in a VAV system is usually controlled to be constant all the year round or to decrease when the outdoor temperature increases. Depending on the supply air temperature, the power used in the HVAC unit to produce the cooling power will differ. A high supply air temperature results in a higher air flow than a low supply air temperature would. Supplying air of a temperature below the dew point temperature will cause condensation, resulting in increased energy use. Optimaizing the supply air temperature with respect to energy use would improve the energy performace of a VAV system that uses 100% outdoor air. The total HVAC energy use will depend on the efficiencies of the components such as the specific fan power (SFP) value, temperature efficiency of the heat recovery unit and the chiller coefficient of performance (COP).

The energy use of a VAV system can be reduced if air is only supplied when there is a demand for it. Instead of only controlling the supply air flow on temperature, it can also be controlled on the level of occupancy in the room by motion sensors or carbon dioxide sensors. This strategy is called demand controlled ventilation (DCV). It has been shown in papers published as International Energy Agency (IEA) documents [11-13] that the use of occupancy sensors and carbon dioxide sensors have a significantly potential of reducing energy use.

Decreasing the U-value of the building envelope by increasing the insulation will increase the need for cooling when the outdoor temperature is lower than the indoor temperature. A decreased U-value also decreases the need for heating at lower outdoor temperatures and decreases the cooling at outdoor temperatures higher than the indoor temperature. Depending on how high the internal heat loads are and the outdoor temperature, the optimal U-value of the building envelope will differ. The most important factor influencing the need for cooling is the glazing areas facing south. This means that broadly it is the architectural design of the building that determines the need for cooling in a given outdoor climate.

The VAV system is based on a high fan controlled static pressure in the main duct. The static pressure must be maintained at a level that covers the pressure losses in the system at maximum load. This results in unnecessary energy use at every load below maximum [14-21] and a generation of noise from the fan and the duct system. Therefore, the pressure level in the system should be kept as low as possible.

How well a system is balanced also affects the energy use. In a well-balanced system there is little difference between the fan supply and the fan exhaust air flow. Depending on air flow difference between supply and exhausted air, temperatures, facade airtightness and wind, air might be infiltrated or exfiltrated through the facade. If the outside air must be heated before being supplied to the building, then infiltrated air will be heated in the room by the heating system and this will increase the total energy use. Otherwise, the air would have been heated through heat recovery from the exhaust air. To reduce the risk of moister problems in the building envelope, air leakage from the inside to the outside has to be avoided. Therefore the exhaust air flow has to be higher than the supply air flow.

1.3 Other aspects of ventilation systems

One of the major problems with today’s mechanical ventilation systems is that they generate too much noise [9]. Standards and national regulations usually set noise limits for frequencies that are audible, but not for other frequencies. A study [22] shows that low-frequency noise (< 200 Hz) interferes with performance and affects health. The air should be distributed and removed as quietly as possible. Major sources of noise are the fan, the duct system with dampers and the air terminals. Sound attenuaters can reduce generated noise. However, a sound attenuator creates a higher pressure loss than a straight duct. The most energy efficient way to reduce noise is to reduce the generation of noise, and that includes noise of any frequency. This can be done in a VAV system by controlling the fan to create only the amount of pressure required, instead of creating a too high pressure, which then has to be reduced by partly closed dampers.

A study [23] shows that users want a greater degree of individual or group control of the indoor climate, and they want systems that are easy to operate. For example, the layouts of offices change over time, and different users have different needs [23] or wishes. The building owner must adapt the building and its ventilation system to new users. Therefore it is an advantage if the ventilation system can be changed without having to be entirely re-commissioned. As an example, a system designed to meet the demanded air flow for 25 students in a classroom would have to be adapted to provide a higher air flow if the number of students would increase. If the system was flexible, it would be easy to change the air flow to satisfy such new demands.

One way to make it easier to make changes of air flow to different zones without affecting the air flow to other zones could be to control the static pressure to be constant at the branch duct level. A damper with a controller connected to a pressure sensor controls the static pressure at one point on the branch. Several zones can then be connected to the branch and the air flow to a zone can be varied by changing the outlet area of the air terminal or the position of a damper. The damper position or the outlet area of the air terminal will then correspond to an air flow which can be calculated instead of measured if compared to the VAV system described before. For this to work the static pressure in the whole branch must not differ too much from the point where it is controlled to be constant. The placement of the pressure sensor and the duct design will determine how much the air flow will differ from the designed air flow. Here also the fan has to control a static pressure in the main duct and this pressure setpoint can be controled depending on the percentage opening of the branch dampers.

2. Previous work

The preformance of the mechanical supply and exhast system and the mechanical exhaust system has also been studied by Herrlin [24, 25]. Herrlin´s research dealt with air flow in a multi-family building but it did not include the air flow within (inter zonal) the apartment. Comparisons to his work are made in the Discussion chapter.

Most research has been made on the VAV system that uses return air. Most often focus is on the proportions between return air and outdoor air [26, 27] in order to make it work in practice and to reduce the energy use. In a 100% outdoor air VAV system there is no return air and therefore is this not a problem.

Norford et al. [28] simulated with DOE-2 an office building in New Jersey. The energy use was calculated for different constant supply air temperatures and a supply air temperature decreasing with increased outdoor temperature. By changing the supply air temperature the energy use was reduced by 10% in winter time and between 11% and 21% in summer time. Ke et al. [29] simulated eight ventilation control strategies in VAV systems and three of the strategies included a change in supply air temperature. The climate data was from south central Pennsylvania, USA. Their conclusion is that the supply air temperature and supply air flow rate are the two proper optimizable parameters on the air side of the HVAC system. Ke and Mumma [30] simulated the effect on ventilation when changing the supply air temperature in a fan powered VAV system (FPVAV) that uses return air. The climate data was from Harrisburg, PA, USA. Mathews et al. [31] showed other ways, such as air-bypass control on cooling coils and system start-stop times, to reduce the HVAC energy use. The supply air temperature's influence on the energy use in a northern European climate with a VAV system that not uses return air has not been studied.

3. Objectives

The objectives of this thesis are to analyze

- how different ventilation systems performe in practise when it comes to supplying and exhausting designed air flow.
- the mechanical exhaust system and the mechanical supply and exhaust system with respect to their ability to maintain designed supply air flow at different outdoor and indoor conditions.
- the design criterias for a VAV system based on a controlled static pressure at the branch duct level.
- the impact of the supply air temperature on the HVAC energy use of a building ventilated and cooled by a VAV system using only outdoor air in a northern European outdoor climate.
- the impact of the U-value of the building envelope on the energy use when the building is ventilated and cooled by a VAV system in a northern European outdoor climate and at different internal heat gains.

4. Methods

To investigate how different ventilation systems performe in practise, the result from the compulsory testing and examination of ventilation systems (OVK) is used. Technicians inspected ventilation systems in Sweden using checklists. Three kinds of buildings; multi-family buildings, offices and schools were tested. Four different systems were used in the tested buildings; natural ventilation, mechanical exhaust air system and the mechanical supply and exhaust air system with and without heat recovery. The result checklists from 5625 inspections have been implemented into a database and evaluated. In order for a system to pass the test, the ventilation performance had to conform to the Swedish building code that applied when the system was brought into operation.

To analyze the air flow in a building when it is ventilated by either the mechanical exhaust system or the mechanical supply and exhaust system an air flow model of a four story multi-family building is defined. By using the same building but different ventilation systems, the systems can be compared on their ability to maintain the supply air flow. A multi-zone model program, COMIS [32-37], is used to do a parametric study of the air flow depending on outdoor temperature, window opening, building envelope airtightness, airtightness between apartments and staircase, wind speed and direction. COMIS is based on the mass balanced equation and it is well established and widely used in this research field.

How much the air flow at the air terminals on a branch with controlled static pressure differs compared to the designed air flow depending on duct design is analyzed with fundamental pressure loss equations. The equations are also used to determine where the pressure sensor should be placed to minimaize the air flow difference between the air terminals. The results are compared with the results from a computer model, PFS [38], for air flow in duct systems developed at the Department of Building Science at Lund University. To validate the model, air flow measurements are made on a branch with controlled static pressure and five air terminals.

A heat balance of a zone and stady state equations of the power requirements of the fans, chiller, boiler and radiator system are used to study the HVAC energy use depending on supply air temperature and U-value of the building envelope. The equations are also used to optimaize the supply air temperature with respect to HVAC energy use. The zone is ventilated and cooled by a VAV system that uses outdoor air only. The yearly energy use is calculated for two different outdoor climates, Luleå and Sturup (north and south of Sweden), and for different control strategies of the supply air temperature.

5. Results

Out of the 5625 ventilation systems studied in Paper I, only an average of 34% performed as intended. As can be seen in Fig. 1, the mechanical exhaust and supply air systems in offices showed the best result, with 62% passing.

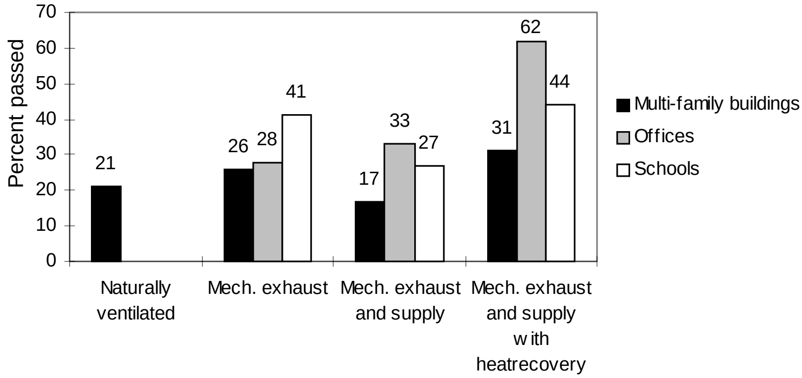

Fig. 1. Percentage of ventilation systems in three different kinds of buildings that passed the test at the inspection (Paper I).

The study also showed that the main reasons for failing the test were lack of maintenance, user influence and change in use of the building. 26 % of the investigated mechanical exhaust systems had defective or missing supply air terminals. In the multi-family buildings with mechanical exhaust and supply air systems, 62% had faulty air flow. The systems without satisfactory maintenance instructions had 50% more remarks than the systems with such instructions.

A 4-story multi-family building with a mechanical exhaust air system was modeled with a multi-zone infiltration program to show how various parameters such as outdoor temperature, building airtightness, wind direction and wind speed influence the air flow in the building. The envelope of the apartments has an airtightness of 1 air exchange per hour (ACH) at 50 Pa, not including the air leakage to the staircase. The systems were commisioned at 20°C outdoor temperature. The exhaust air flow is almost independent of the outdoor temperature, but as seen in Fig. 2 the supply air flow through the air terminals ("Supply" in Fig. 2) is very sensitive to changes in the outdoor temperature. Air through terminals and air infiltration count as supply air. Air coming from the staircase might contain air from other apartments and therefore does not count as supply air.

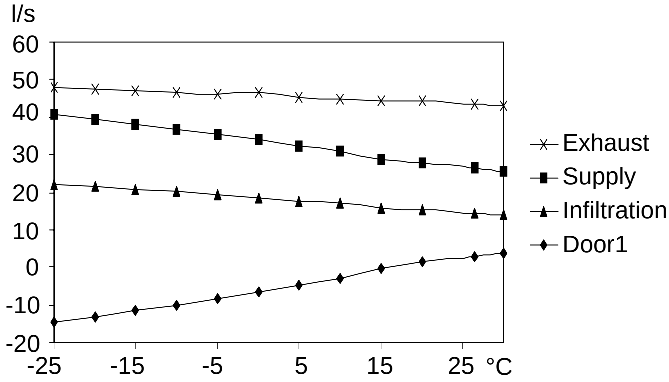

Fig. 2. Air flow in and out of the apartment at the first floor depending on outdoor air temperature. “Door1” is the air leakage from the apartment to the staircase where negative flow is from the apartment to the staircase. “Supply” is the air flow through the air terminals, and “Exhaust” is the fan exhaust air flow. The airtightness of the apartment envelope is 1 ACH at 50 Pa.

There is an air flow from the apartment on floor one to the staircase when the outdoor temperature is below 15°C. At the same time, at floor 4 the air flows from the staircase to the apartment, and therefore there will be an exchange of air between the apartments. The supply air flow on the fourth floor varies between 41 l/s at 20°C outdoor temperature and 26 l/s at -20°C outdoor temperature. If it is not very airtight between the apartments and the staircase there will be an air exchange between the apartments. The supply air flow is also dependent on the airtightness of the building envelope and on wind speed and direction. An open window in one room can cut off the supply air flow altogether in other rooms of the apartment.

The supply air flow to one bedroom room compared to the designed air flow varies between +65% and -10% depending on the wind direction at a wind speed of 6 m/s.

In Fig. 3 the air flow in an identical apartment as in Fig. 2 is shown but here the building is ventilated by a mechanical supply and exhaust air system.

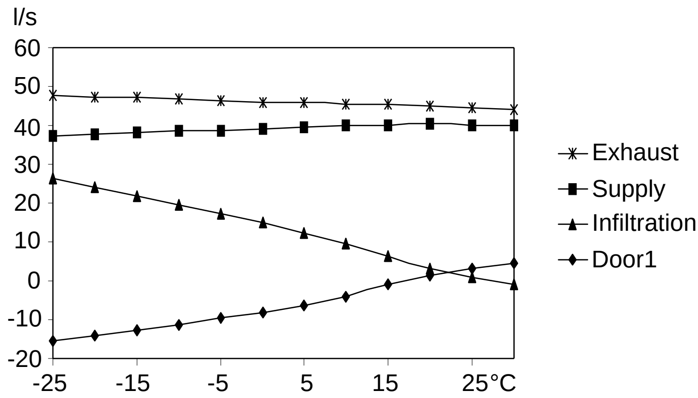

Fig. 3.Air flow in and out of the apartment at the first floor depending on outdoor air temperature. Identical conditions to those in Fig. 2, but with a mechanical exhaust and supply air system. “Door1” is the air leakage from the apartment to the staircase where negative flow is from the apartment to the staircase. “Supply” is the fan supply air flow, and “Exhaust” is the fan exhaust air flow.

Neither the fan supply air nor the exhaust airflow is sensitive to wind speed and temperature changes, but the infiltration varies as shown in Fig. 3. The supply air flow varies at the fourth floor between 43 l/s at 20°C outdoor temperature and 41 l/s at -20°C outdoor temperature. As with the mechanical exhaust system, here also there will be an air exchange between the apartments. An open window in one room will have almost no effect on the fan supply air to the other rooms in the apartment but can increase the air flow from the apartment to the staircase and thereby increase the air exchange between apartments. With this system, as with the mechanical exhaust system, the air flow in the whole building will be changed if some of the users modify the air terminals. Exfiltration can occur in individual rooms depending on wind speed and direction.

Paper IV gives the fundamentals for a system based on a controlled constant static pressure in the branch ducts. By controlling the static pressure at the branch duct level it is possible to vary the flow to different air terminals without measuring the individual air flow and without major changes in air flow to other air terminals. How much the air flow differs is dependant on the design of the duct work and where the pressure sensor at the branch level is located. A dimensionless parameter called *s* is useful when studying the air flow from a straight duct with constant diameter to air terminals that are designed for equal air flow and placed with equal distance between each other. The parameter *s* is dependant on the duct length (L_t [m]), duct diameter (*d* [m]), a duct friction loss coefficient (λ_0 [(m/s)$^{0.2}$] depending on duct diameter) and air velocity (u_0 [m/s]) according to Eq. (1).

$$s = \frac{\lambda_0 \cdot L_t}{2.8 \cdot d \cdot u_0^{0.2}} \quad [-] \qquad (1)$$

When s is less than 1 the static pressure in the branch duct will increase, resulting in a higher air flow at the diffusers at the end of the branch compared to those at the beginning of the branch. For s higher than 1 the static pressure will decrease with the opposite air flow distribution. For s to be higher than 1 a duct with a diameter of 0.315 m has to be more than 50 m long and therefore is s in practise most often less than 1.

Figure 4 shows where the pressure sensor should be placed on the branch duct in order to achieve a minimum of air flow difference between the diffusers on the branch. The placement x_s is dimensionless and can be between 0 and 1 where 0 is a placement at the beginning of the branch and 1 is at the end of the branch.

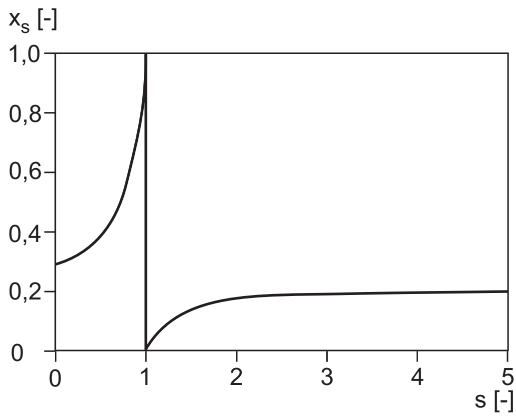

Fig. 4. Placement (x_s) of pressure sensor as a function of s. If x_s equals 1 then the sensor should be placed at the end of the branch.

The parameter s can also be used to calculate the relative flow difference between the diffusers on the branch. Equation 2 gives the relative air flow difference (e [-]) where Δp_{diff} is the pressure loss at a diffuser at maximum flow, u_0 is the air velocity after the first diffuser on the branch and ρ is the air density.

$$e = \sqrt{\frac{\Delta p_{diff} + \frac{\rho \cdot u_0^2}{4} \cdot |1 - s|}{\Delta p_{diff}}} - 1 \qquad [-] \qquad (2)$$

The fan in a system concisting of a supply air fan connected to a main duct to which branch ducts with controlled static pressure is connected has to control a static pressure in the main duct. If the fan static pressure set point is optimized, the fan power requirement will be less during partial loads than if the pressure is constant. Analyzing the power required to transport 50% of maximum air flow through the duct system, a constant static pressure can result in a twofold power requirement compared to when optimizing the fan static pressure.

The HVAC energy use in relation to the supply air temperature in a VAV system that uses 100% outdoor air is studied in Paper V. The HVAC unit and the system is shown in Fig. 5. Limitations such as no solar radiation and a constant coefficient of performance (COP) of the chiller are made and described in detail in Paper V.

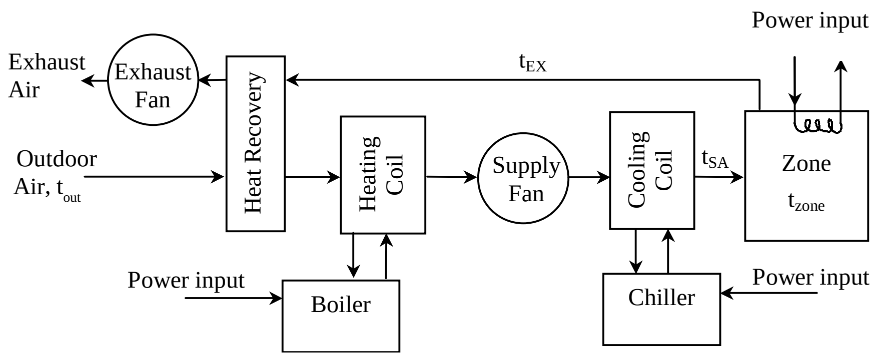

Fig. 5. The components of the HVAC unit and the zone in which the temperature is t_{zone}. t_{EX} is the temperature of the exhaust air and t_{out} is the outdoor temperature. t_{SA} is the supply air temperature.

Three equations are derived to find an optimal supply air temperature with regard to energy use. Which equation is used depends on outdoor conditions and which parts of the HVAC unit are in use. Equation (3) is shown as an example. The equation is valid for the case when the chiller is used without condensation and the zone temperature is higher than the outdoor temperature.

$$t_{SA} = t_{zone} - \frac{k_3}{k_2} \cdot \frac{2}{t_{zone} - t_{out}} \qquad [°C] \qquad (3)$$

t_{SA} is the optimal supply air temperature, t_{zone} is the zone temperature and t_{out} is the outdoor temperature. The constants, k_2 and k_3, are dependent on the fan and chiller efficiencies and the internal load. The constants are described in detail in Paper V.

Figure 6 shows the optimal supply air temperature dependent on the outdoor temperature for a typical HVAC system and one zone with an internal heat load of 30 W/m² floor area. The temperature set point in the zone is 21°C. The heat load that the system has to cool increases with an increasing outdoor temperature becuase of the change in heat transfer through the facade. In Fig. 6 six different sections are marked.

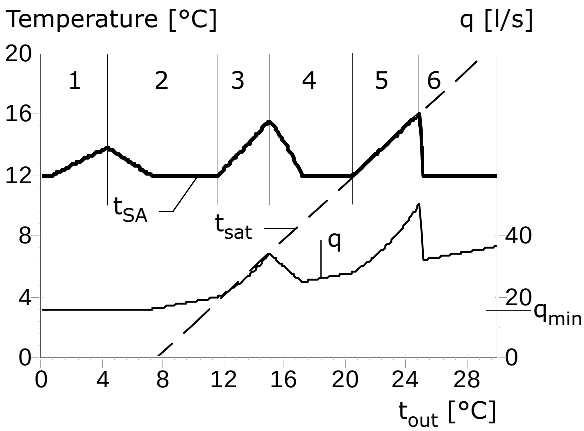

Fig. 6. Optimal supply air temperature set point (t_{SA}) and supply airflow, q , at different outdoor temperatures (t_{out}). The lowest allowed supply air temperature is 12°C and the lowest allowed supply air flow (q_{min}) is 16 l/s. The outdoor relative humidity is 60%. t_{sat} is the saturation temperature. The average U-value of the facade is 1.5 W/(m²·°C) and the facade area is 9 m². The figure is divided inte six sections which are explained in the text below.

Section 1: The heat exchanger is used in its full extension. The increase in t_{SA} is due to increased outdoor temperature. The supply air flow is the minimum supply air flow (q_{min}).

Section 2: The cooling need in the zone increases due to increased outdoor temperature. Therefore the t_{SA} is decreased until the lowest allowed t_{SA} is reached.

Section 3: The cooling need in the zone increases due to increased outdoor temperature. The heat exchanger is no longer used. The supply air is now neither heated nor chilled in the HVAC unit. The air flow increases.

Section 4: Here the energy use is lower if the supply air is chilled (chiller in use). The supply air temperature is now lower than the outdoor temperature. The t_{SA} decreases until the lowest allowed t_{SA} is reached. The air flow decreases.

Section 5: The t_{SA} has now reached the saturation temperature (T_{sat}). Instead of maintaining 12°C, which would result in a loss in energy due to condensation, the t_{SA} follows the saturation temperature.

Section 6: The fan power use is now so high that it is better to chill the air as much as possible (12°C) even though there will be condensation.

Figure 7 exemplifies the energy use during one year for one zone dependent on control strategy. “Optimal t_{SA}” in Fig. 6 indicates that an optimal supply air temperature is used. “Decreasing” is a control strategy where the supply air temperature decreases with an increased outdoor temperature. The strategies named “Constant” indicate that the supply air temperature is constant throughout the year (12, 14 and 16°C respectively).

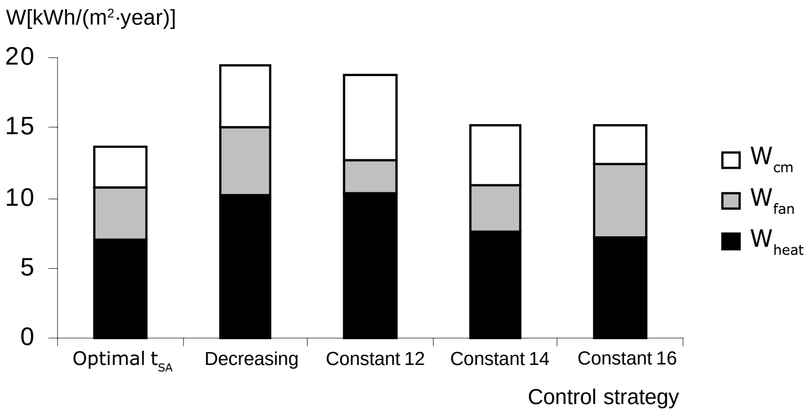

Fig. 7. Energy use per m² floor area for one zone with a facade area of 9 m² and a floor area of 13.5 m² during one year (1977) in daytime in Sturup with a constant internal heat load of 26 W/m² floor area. The average U-value of the building envelope is 1.5 W/(m²·°C) and no infiltration is assumed. W_{cm} is energy used by the chiller, W_{fan} is energy used by the fan and W_{heat} is energy used to heat the air either in the room or in the HVAC unit. Solar radiation is not included.

Increasing the insulation of the building envelope will increase the need for cooling when the outdoor temperature is lower than the indoor temperature, but it will also decrease the need for heating. An optimal U-value depending on outdoor climate, occupied hours and internal heat load is calculated and the result is shown in Fig. 8. It is assumed that there is no solar radiation.

The internal heat load has to be above 135 W/m² in Sturup and above 175 W/m² in Luleå, in order to get an optimal U-value higher than zero when the building is only occupied during day-time hours (06-18).

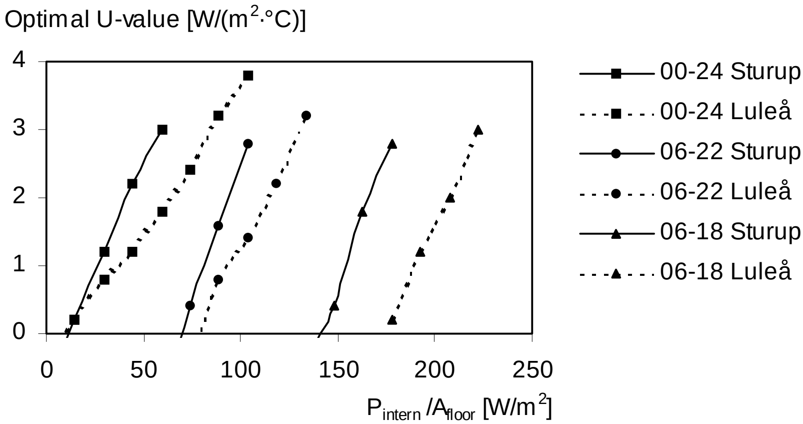

Fig. 8. Optimal U-value of the building envelope depending on internal heat load per m² (P_{intern}/A_{floor}), occupied hours and outdoor climate (Luleå and Sturup). The internal heat load is only active when the zone is occupied.

6. Discussion

6.1 Influence of users on ventilation systems

With the kinds of ventilation systems investigated, the users of the building are very seldom allowed to influence the indoor climate parameters. Mostly, the only thing they can change to any extent is the visual climate, such as lights and colors on the walls, and they can adjust the air temperature by using the radiator thermostats or open a window.

When a person experiences discomfort, this usually results in some action. The action might affect the system's performance. One example is when the user feels a draft from the supply air terminal; he or she might close it or even fill it with some material that prevents the air from entering the room. This is one reason why 26% of investigated mechanical exhaust systems in Paper I had defective or closed supply air terminals. One of the problems with the mechanical exhaust system is how to supply the air during the winter without causing drafts. The air enters the occupied zone at a low temperature, and even though the air velocity might be relatively low, the person will feel a draft. Sometimes the supply air terminal is placed behind a radiator to increase the supply air temperature. A condition is that the radiator must be active, and that is not the case for a wide range of outdoor temperatures when the wall is well insulated and the window has a low U-value.

It is also common that exhaust air terminals generate noise and then users sometimes close or fill the terminal with some kind of material. This action can change the air flow in the whole building. When using constant static pressure at the duct branch level, as described in Paper IV, this action only affects the room in which the terminal is located. Another typical action in response to discomfort is to open a window when it is warm. With the mechanical exhaust air system this action can deprive other rooms of supply air. This will not happen when mechanical exhaust and supply air systems are used but it will increase the air exchange between apartments if air leakage from the apartment to the staircase is possible.

In many cases, the use of a building can change during its lifetime. For example, two office rooms may be converted into a single conference room; a classroom may need to accommodate a larger number of students; or a hospital office may become a treatment room. The systems investigated in Paper I failed the test if the activity had changed and the system was not adjusted to the new activity. The systems had not been adjusted because of the need for, and cost of, re-commissioning the entire ventilation system. One might say that this is not a system error but a lack of flexibility. The lack of flexibility was one reason why the air flow rates of many of the systems were too low. With a constant static pressure at the branch level the airflow rates can to some extent be changed to meet new demands, and this does not require a re-commissioning of the entire ventilation system.

Systems with constant pressure at the branch level can be used to obtain an increase in flexibility and a decreased effect on the air flow when the user makes changes at the air terminals.

A significant cause of the low share of systems that passed the test in Paper I was insufficient or non-existent maintenance. To facilitate maintenance, the HVAC unit must be installed in a way that makes it easy to access. It is recommended that all systems be delivered with maintenance instructions, which was very seldom the case for the investigated systems.

6.2 Indoor climate

The supply air flow when using the mechanical exaust system varies in a wide range depending on outdoor temperature and wind. By using the mechnical supply and exhaust air system the supply air flow is stabilized. The results confirm well to Herrlin’s research [24, 25]. Herrlin’s result shows on the same differences between the systems, but there is a difference in air exchange between the apartments. In his result the air exchange between the apartments is much lower when using the mechanical exhaust system. The reason for this is the difference in airtightness of the staircase. In Herrlin’s study the staircase is less airtight when the mechanical exhaust system is used than when the mechanical supply and exhaust system is used. In this study, it is the opposite because of a ventilator connected to the outdoors and placed at the top of the staircase when the mechanical supply and exhaust air system is used. The conclusion is that no matter what system is used, the staircase should not be airtight if the air exchange between the apartments is to be as low as possible. Having equal staircases, the air exchange between the apartments is less when using the mechanical exhaust system. Therefore, the importance of the airtightness between the apartments and the staircase is greater when using the mechanical supply and exhaust air system compared to the mechanical exhaust air system. In the model of the four story multi-family building no internal leakage between the apartments was assumed. Levin [39] showed by measurment that air flow between apartments are likely to occur and to be in the range of 0 and 18 l/s. Including these leakages in the model would in some extent influence on the result but the overall conclusions would remaine.

Many premises have a long cooling season even though they are located in northern Europe. The long cooling season is a result of internal heat loads, the lack of solar shading and the insulation of the building envelope. A cooling system is often installed in modern buildings. The benefits of the resulting improvement to thermal comfort include, for example, an increase in workers' performance. The period in northern Europe when the outdoor temperature is higher than the requested indoor temperature is relatively short. This fact reduces the need for return air, and therefore 100% outdoor air can be used to cool the building. As described in the introduction, when using air for cooling it is necessary to increase the air flow. When using 100% outdoor air in a VAV system the IAQ will be improved during the part of the year when there is a need for cooling the building.

The constant pressure at the branch level makes it possible to vary the air flow. It is realistic assuming that most rooms are not occupied 24 hours a day. The air flow can be increased when the room is occupied, and decreased to ventilate only building emissions when not occupied. This would make it possible to improve not only the IAQ, but also the acoustic climate because when the system is not running on peak load the velocities in the ducts will be lower and the noise generation will decrease.

A lower fan static pressure lowers the fan speed and as a result reduces noise generation. An open damper generates less noise than a closed damper. By optimizing the fan static pressure, the pressure will be lower at partial load and the dampers will be more open. Therefore, the optimization of the fan static pressure improves the acoustic climate.

6.3 Energy

The mechanical supply and exhaust air system requires the supply and exhaust air flow to be balanced, and this requires a special technique that is time consuming. Paper 1 points out the problem with balancing this kind of system, and this is one reason why 62% of the multi-family buildings had the wrong air flow rate. As described in the introduction, a system that is not balanced can increase the energy use and cause damage to the building. When using constant static pressure at the branches, the balancing procedure is facilitated, and this increases the chance of having systems that are well balanced.

When using the mecanical exhaust system there is almost never a risk for exfiltration irrespective of changes in outdoor temperature and wind. With the mechanical supply and exhaust air system there is a high risk of exfiltration in rooms where the air is supplied. Threrefore it is important that the facade is as airtight as possible. An other possibility would be to increase the exhaust air flow, although this would increase the energy use.

As described before, the IAQ can be improved by ventilating according to demand, and the energy use will depend on how many rooms of the building that are occupied. For example, consider a room that is occupied 8 hours a day and ventilated by a constant air flow of 15 l/s day and night. By reducing the supply air flow to 4 l/s when it is not occupied, the supply air flow can be increased to 22 l/s when the room is occupied without increasing the fan energy use.

Optimizing the fan pressure set point in a VAV system will decrease the energy use when the system is running on partial loads. Therefore the fan energy use during the heating seasons, nights, weekends and vacations will be low.

When a wide range in supply air temperature can be used without the risk of creating drafts, it is possible to vary the supply air temperature. The supply air temperature is usually controlled to be constant year round, or to decrease when the outdoor temperature increases. Paper V describes the theory for an optimal supply air temperature in terms of energy. To be able to control the temperature optimally, data about the fan, chiller, heat recovery unit, internal loads, zone mode, zone temperature set point and the outdoor temperature are needed. All these parameters are known in a building with a modern HVAC system and individual temperature control.

A comparison of the energy use between a constant supply air temperature at 14°C and the optimal strategy shows a difference of 10% in Sturup with an internal load of 26 W/m² floor area (Fig. 7). This is a rather small difference, though this is only true if the internal loads are constant, and that is not the case in practice. If the internal load should change from 26 W/m^2 floor area to 44 W/m^2 floor area, still comparing the same strategies, it would result in an 18% lower annual energy use when controlling the supply air temperature optimally. Therefore, there is great potential in this optimization to reduce the HVAC energy use, provided there is a cooling need for much of the year. When a system has been put into operation it is almost impossible to control the load or the products' efficiencies, but the supply air temperature can be controlled and optimized to decrease the energy use. Therefore, the equations can be efficient to implement in existing 100% outdoor VAV systems if the parameters mentioned above are known.

No solar radiation is included in the energy calculations of different control strategies of the supply air temperature. If solar radiation were present, the absolute calculated HVAC-energy use would be affected. In the comparison between the control strategies, the affection would be in the same direction and therefore the difference would be small. Solar radiation will not affect the supply air temperature optimization because from the system perspective there is no difference in solar gain or internal heat load. Therefore, solar radiation can be treated as a part of the internal heat load.

In most practical applications the internal heat load is below 135 W/m^2 and the building is not used 24-hours a day. As can be seen in Fig. 8, the optimal U-value regarding energy use is only above zero when the internal loads are very high or when the loads are active more than 12 hours a day. The energy use might be reduced at a higher U-value but the penalty might be a poorer thermal comfort and an increase in energy use if the use of the building is changed.

7. Future work

The 100% outdoor air variable volume should be compared to other systems that provide both cooling and ventilation. The comparison should be done in a life cycle perspective. One problem will be that the systems will probably not create identical thermal climate and IAQ. Whether and how these two parameters should be integrated in a life cycle perspective needs to be studied. The energy use in practice of a 100% outdoor VAV system and maintenance cost should be studied to provide basic data for the life cycle analysis. A cost-effective way to implement the pressure controlled system in residences should be studied to improve the residential IAQ. The behavior of users when given the opportunity to choose a minimum, and to some extent a maximum, supply air flow would have been interesting to study.

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I

Paper I

Evaluation of Swedish Ventilation Systems

By

Fredrik Engdahl

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Evaluation of Swedish Ventilation Systems

It is of great importance to the indoor air quality that the ventilation systems, not only directly after taken into operation, operate as they were designed and according to the regulations. The objective of the paper is to show how well the systems satisfy the national regulations after being in operation for several years and what the main defects are. The requirement for a system to pass the test is that the ventilation performance conforms to the regulations that applied when the system was brought into operation. Of the 5625 evaluated systems, 34% passed the test.

1. INTRODUCTION

It is of great importance to the indoor air quality that ventilation systems perform as they were constructed and designed to. Immediately after a system has been taken into operation, it is tested to confirm that the system satisfies the national regulations. The objective of this paper is to show how well the systems satisfy the national regulations after being into operation for several years and what their main defects are.

In Sweden the compulsory testing and examination of ventilation systems [1] has been going on since 1993. The systems are inspected at certain intervals [3] depending on building type.

Three types of buildings have been studied; schools, offices and multi-family houses. The systems were divided into four categories: naturally ventilated, mechanical exhaust air, mechanical exhaust and supply air, and mechanical exhaust and supply air with heat recovery. A specially designed check-list is used during the inspection. The check-list consists of five main parts: operation and maintenance instructions, contamination of the supply air system, contamination of the exhaust air system, system functions, and an indoor climate part. The inspections are carried out by authorised ventilation technicians.

The systems in this paper have been inspected for the first time. The requirement [2,3] for a system to pass the test is that the ventilation performance conforms to the regulations [3] that applied when the system was brought into operation. Of the 276 naturally ventilated multi-family houses, 21% passed the test. Of the 1736 exhaust air ventilated buildings, 31% passed the test. Of the 2201 exhaust and supply air ventilated buildings, 29% passed the test. Of the 1410 exhaust and supply air with heat recovery ventilated buildings, 50% passed the test. The main reason for disallowance was that the air flow rate was too low. Of those which did not pass the test, 31% did not have maintenance instructions. The paper shows that the use of standards and regulations only directly after the system is brought into operation is not a guarantee of a well-functioning system during a longer period. The systems must be easy to maintain and operate. They must also match the quality and potential of the maintenance organisation.

2. METHOD

Specially designed check-lists from the compulsory testing and examination of ventilation systems in Sweden have been used. The check-list consists of five main parts: operation and maintenance instructions, contamination of the supply air system (CSA), contamination of the exhaust air system (CEA), system functions, and an indoor climate part. The sections of each main part are shown in Table 1.

Table 1. Check list from the compulsory testing and examination of ventilation systems

Documents	**CEA**	**Functions**
drawings	filter	filter
operation and maintenance instructions	heating and cooling coils	heating and cooling coils
inhabitants point of view	heat recovery unit	heat recovery unit
CSA	fan	dampers
air intake	ducts	control and regulation device
filter	exhaust air terminals	fan
heating and cooling coils	fan room	air flow rates
heat exchanger	**Indoor climate**	humidification unit
humidification unit	temperature	refrigerating unit
fan	odour	operation times
ducts	draught	air terminals
supply air terminals	noise	temperature
fan room		

The inspections were carried out by different authorised ventilation technicians [3]. The technician measured [4] the air flows and compared those with the air flows according to the regulations. Those functions in Table 1 that existed in the checked system were inspected and the supply air temperature was measured. The contamination parts in Table 1 were checked subjektively by the technician without any instruments to see if any parts were in need of cleaning. If the checked part of the system affected any function due to contamination, this resulted in one remark in the function section and one in the contamination section. If the technician made a remark on any of the sections, he also wrote down the reason for the remark on a separate protocol. A fan that is broken results in a remark in the fan section and should also result in a mark in the air flow rate section. Since the inspections were carried out by different technicians, this might not always be the case. Some technicians did not put a remark in the air flow rate section.

The reason for a remark in the air flow rate section is not always due to a defective system. For example, if the air flow rate into a class room was designed for 20 students, but nowadays is being occupied by 30 students, this will result in a remark in the air flow rate section.

Number of buildings, the building types and ventilation system types are shown in Table 2. The requirement for a system to pass the test was that the ventilation performance conformed to the regulations that applied when the system was brought into operation.

Due to the integrity of the building owners the location of the building were not known, but the buildings were mainly from the major cities in Sweden; Stockholm, Gothenburg and Malmoe.

The age of the building was known, but not the age of the system. The median value of the building years is presented in Table 2. Some buildings have been renovated, which means that the median age of the systems is lower than the building median age.

Table 2. The evaluated systems

Building type	System type	Median building year	Number of systems
Multi-family building	naturally ventilated	1929	278
Multi-family building	mech. exhaust	1951	1073
Multi-family building	mech. exhaust and supply	1973	108
Multi-family building	mech. exhaust and supply with heat recovery	1984	213
School	mech. exhaust	1958	544
School	mech. exhaust and supply	1960	1081
School	mech. exhaust and supply with heat recovery	1956	562
Office	mech. exhaust	1950	119
Office	mech. exhaust and supply	1964	1012
Office	mech. exhaust and supply with heat recovery	1978	635
			Σ 5625

3. RESULTS

The results of how many of the systems conformed to the regulations that applied when the system was brought into operation are shown in Fig. 1.

The numbers given in brackets are per cent of evaluated systems of the current system type and building type.

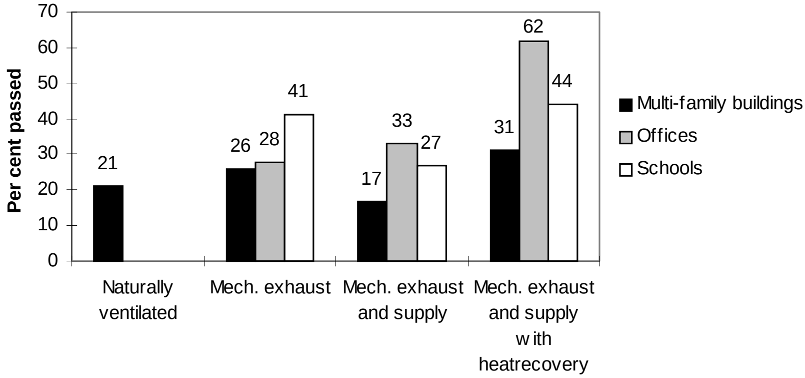

Fig. 1. Percentage of systems in each category that conformed to the regulations that applied when the system was brought into operation.

3.1 Naturally ventilated multi-family buildings

The main reasons for not passing the test were: defective or missing supply and exhaust terminals (36%), too low air rate flow (35%) and contamination of exhaust air ducts (23%).

3.2 Mechanical exhaust air ventilated buildings

In multi-family buildings, the main reasons for not passing the test were: too low air flow rate (64%), defective or missing supply and exhaust terminals (26%) and contamination of exhaust air ducts (23%). Of the systems, 30% were without satisfactory maintenance instructions.

In offices the main reasons for not passing the test were: too low air flow rate (48%), defective fans (17%) and contamination of exhaust air ducts (15%). Of the systems were 39% without satisfactory maintenance instructions.

In schools, the main reasons for not passing the test were: too low air flow rate (31%), defective fans (14%) and defective control unit (8%). Of the systems, 59% were without satisfactory maintenance instructions.

3.3 Mechanical exhaust and supply air ventilated buildings

In multi-family buildings, the main reasons for not passing the test were: too low air flow rate (73%), contamination of supply air filter (26%) and defective fans (14%). Of the systems, 53% were without satisfactory maintenance instructions.

In offices, the main reasons for not passing the test were: too low air flow rate (47%), defective control unit (15%) and contamination of supply air fan (14%). Of the systems were 54% without satisfactory maintenance instructions.

In schools the main reasons for not passing the test were: too low air flow rate (44%), contamination of supply air filter (21%) and contamination of supply air intake (18%). Of the systems, 67% were without satisfactory maintenance instructions.

3.4 Mechanical exhaust and supply air with heat recovery ventilated buildings

In multi-family buildings, the main reasons for not passing the test were: too low air flow rate (62%), contamination of supply air filter (8%) and defective fans (5%). Of the systems, 12% were without satisfactory maintenance instructions.

In offices the main reasons for not passing the test were: too low air flow rate (25%), contamination of supply air filter (8%) and contamination of supply air fan (8%). Of the systems, 40% were without satisfactory maintenance instructions.

In schools the main reasons for not passing the test were: too low air flow rate (28%), defective control unit (13%) and contamination of supply air intake (9%). Of the systems, 66% were without satisfactory maintenance instructions.

3.5 Maintenance

The systems without satisfactory maintenance instructions had 50% more remarks (the remark on satisfactory maintenance instructions not included) compared to those with satisfactory maintenance instructions.

4. CONCLUSIONS AND DISCUSSION

In naturally ventilated buildings, the users often install kitchen fans and toilet fans which change the air flow rate and flow pattern in the building. The reason why the users install fans is insufficient air flow rate, for such reasons as present-day shower behaviour and the smell of cooking in the whole apartment and not only in the kitchen. Problems with odour from other apartments are common, and caused by the installed fans. Most of the naturally ventilated buildings are from the beginning of the century, and the ducts have never been cleaned; this, of course reduces the air flow rates.

A problem with the mechanical exhaust air ventilated buildings is how to supply the air. The terminals for supply air often cause draught, noise and filth from the street. This makes the users close or remove the supply air terminals.

The exhaust ducts have hardly ever been cleaned which reduces the air flow rates.The fans are often located in the attics where they are difficult to reach for maintenance. The fans are often defective due to broken or worn out belts. These defects are easy and cheap to repair.

In mechanical exhaust and supply air ventilated buildings, the major problem is the contamination of the supply air system. The contamination is often caused by defective filters and air intakes. The filters are often found to be leaky, wet, full of dirt or missing. The systems are complex and the caretakers often have problem of over-viewing the systems and handling the control system. Defects in the control system are for example fire dampers that do not respond and problem with control of the supply air temperature.

Mechanical exhaust and supply air with heat recovery ventilated offices are those system with the highest share passing the test. The reason for this might be the users' stronger economy and knowledge of the relation between good indoor air climate and productivity, and therefore they may make higher demands on the design of the installation. The main problem in this case is also the contamination of the supply air system. The reason why these systems do not have as many remarks as the mechanical exhaust and supply air systems might be that the systems are younger and therefor a deficient maintenance does not show yet. Another reason might be an improved filter technique.

The many remarks in the air flow rate section are often due to unbalanced systems. There is a big difference between the supply and exhaust air flow rate. The difficulty in balancing the systems and the lack of following up the balancing is one important reason why the systems are unbalanced.

The existence of operation and maintenance instructions is no guarantee of a well-maintained system, but in any case this is a necessity. A properly functioning system requires that it is well maintained and regularly cleaned. Since many systems, after being into operation for a while, do not operate as they were intended to, the compulsory testing and examination of ventilation systems is an imperative necessity. A well-maintained system also results in lower energy costs and an increased life span of the components.

The systems must be robust, easy to maintain and to operate. They must also be flexible in order to adjust for changes in the ventilated spaces, such as changes in number of occupants and their behaviour. The system must also match the quality and potential of the maintenance organisation, since many of the caretakers are having problem with over-viewing and understanding the complex systems.

Acknowledgements-- This study has been supported by The Swedish Council for Building Research, SBUF - The Development Fund of the Swedish Construction Industry, Föreningen V, Byggrådet LTH and The National Board of Housing, Building and Planning.I thank Prof. Anders Svensson.

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II

Paper II

Stability of Mechanical Exhaust System

By

Fredrik Engdahl

Indoor Air 1999; 9: 282-289

Stability of Mechanical Exhaust Systems

Abstract The mechanical exhaust system, where outdoor air is supplied through infiltration and devices in the building shell, is a common ventilation technique in multi-family buildings in Europe. The objective of this paper is to determine how well the system meets indoor air quality standards and regulations and how sensitive it is to disturbances, such as window opening, temperature differences and resident behavior, how different building construction parameters affect the airflow. A simple model of a multi family building has been simulated with a multi zone infiltration program. It was found that the system almost never fulfills the regulations and that the actions of one resident can often affect the airflow in other apartments. The use of standards and regulations defining indoor air quality and energy efficiency will be ineffective for these systems because they are not stable, that is, sensitive to disturbances such as weather changes and resident behavior.

Key words Ventilation; Mechanical exhaust; Multi family building; Modeling; Infiltration; Stability.

Introduction

Standards and national regulations (ASHRAE, 1989; Boverket, 1994) specify and recommend airflow rates to achieve a certain energy efficiency and indoor air quality. A system design goal is to achieve these airflow rates or a higher level of energy efficiency and indoor air quality specified by the customer. Once the system is in operation it is checked by a technician to ensure that it fulfills the design criteria. The system may at this time satisfy customer specifications and the regulations, but it may not perform well when the occupants move in and when outdoor conditions change. No standard or regulation gives the customer the ability to specify the system stability, that is, the sensitivity to disturbances such as weather changes and resident behavior.

A common ventilation system design for multi-family buildings, and sometimes offices, is the mechanical exhaust system. The system is common because it is inexpensive and does not require much space compared to other systems. Air is exhausted from the kitchen, the bathroom and the clothes closet, and is supplied through slot openings, located in bedrooms and the living room under the windows behind the radiators. The air supply devices are often poorly maintained (Engdahl, 1998) and operated improperly because often no instructions are given. Krüger (1996) has described the thermal comfort next to this kind of supply air devices and found through measurement (Krüger and Kraenzmer, 1996) that it was poor.

Levin and Eriksson (1988) made measurements in a tight (0.8 ACH at 50 Pa) four-story building with a mechanical exhaust system. Only a small affect on total air change rate in the apartment due to wind (1-7 m/s) was found, but individual rooms could show significant changes in supply airflow.

A model of a four-story multi-family building with a staircase and one apartment on each floor is used to examine how the airflow changes under different conditions. The multi-zone infiltration program COMIS 3.0 (Feustel and Rayner-Hooson, 1990; Feustel and Smith, 1997), developed by nine countries at Lawrence Berkeley Laboratory, was used to simulate the model.

The system can hardly ever fulfill the regulations and during most of the year an exchange of air between the apartments exists. The system is sensitive to how the residents behave and one resident's action has a significant impact on the ventilation of the other apartments. The use of standards and regulations defining indoor air quality and energy efficiency is rather useless if the system is not stable.

Method

A model of a four story multi-family building with a staircase and one apartment at each level was examined. The system is modeled with COMIS 3.0, a multi zone infiltration program based on the mass balanced equation (Feustel and Rayner-Hooson, 1990):

$$0 = \sum_{l=0}^{m} \left\{ \sum_{j=0}^{k} \left[\rho \cdot C_{j,l} \cdot \left| P_{0_{j,l}} - P_i \right|^{n_{j,l}} \cdot \left(\frac{P_{0_{j,l}} - P_i}{\left| P_{0_{j,l}} - P_i \right|} \right) \right] \right\} \tag{1}$$

where:

m	number of zones
k	number of flow paths in zone l
ρ	density of air
$C_{j,l}$	flow coefficient for flow path j of zone l
$P_{0\,j,l}$	external pressure for flow path j of zone l
P_i	internal pressure
$n_{j,l}$	flow exponent for flow path j of zone l

The program represents the airflow through components via the widely used power law, i.e.:

$$\dot{m} = C_S \cdot \Delta p^n \qquad \text{(kg/s)} \qquad (2)$$

where C_S is the air mass flow coefficient [kg/(s·Pa^n)] and n is the exponent [dimensionless]. C_S is corrected to temperature according to Tamura and Wilson (1967):

$$C_{STcorr} = C_S \left(\frac{\rho}{\rho_0} \right)^n \cdot \left(\frac{\mu}{\mu_0} \right)^{1-2n} \qquad \text{(kg/(s·Pa}^n\text{))} \qquad (3)$$

where:

C_{STcorr} C_S corrected for temperature

ρ density, kg/m^3

ρ_0 reference density at 293.15 K, kg/m^3

μ dynamic viscosity, Pa·s

μ_0 reference dynamic viscosity at 293.15 K, Pa·s

n flow exponent, -

Figure 1 shows the apartment at floor one, where R12 is room 2 at floor 1, LR1 is living room at floor 1, K1 is kitchen at floor 1, T1 is bathroom at floor 1 and Door1 is the door between the apartment at floor 1 and the staircase. Every floor consisted of an identical apartment. There was no internal leakage between the rooms and no leakage through the roof.

Ventilation

The system was designed to provide a supply airflow of 8 L/s to the bedrooms, R11, R12 and R13 in Figure 1. 15 L/s air was exhausted from the kitchen (K1), 20 L/s from the bathroom (T1) and 10 L/s from the closet (Closet1). The exhaust airflow rates fulfill Swedish regulations [14] and ASHRAE Standard 62-1989 (ASHRAE, 1989). The supply airflow rates to the bedrooms fulfill the Swedish regulations for a two-person bedroom (8 L/s) and the ASHRAE standard (7.5 L/s) for one-person bedroom.The supply airflow rate to the living room, 16 L/s, fulfills the ASHRAE standard (0.35 ACH) and the Swedish regulations (0.35 L/sm^2). The total exhaust airflow in each apartment is 45 L/s, 0.584 ACH. The exhaust system was showed in Figure 2.

SupplyAir Devices

The devices were placed 0.5 meters above floor level under a window and behind a radiator to prevent draught during colder seasons. The residents can adjust the opening level of the device to increase or decrease the flow rate. If the resident experiences draught, an alternative to decreasing the airflow is to increase the radiator power. At the time of installation, the supply devices were set to provide 8 L/s at a pressure difference of 10 Pa, and the flow exponent is 0.5.

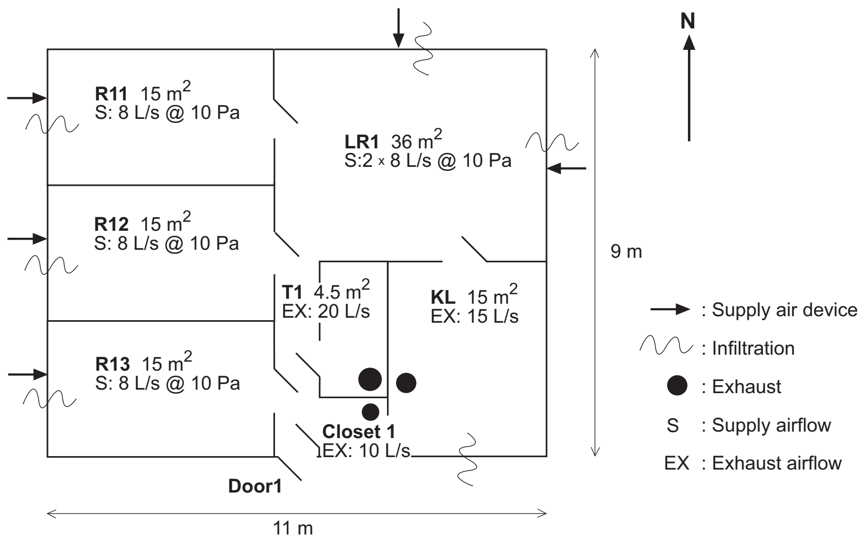

Fig. 1 The apartment at floor one with supply, infiltration and exhaust flows

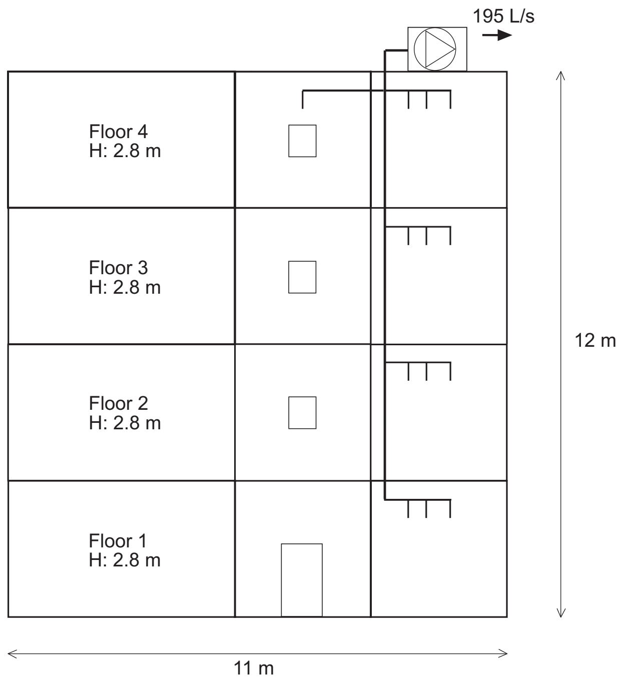

Fig. 2 Front view of the building and the exhaust system

Exhaust Air Devices

These devices were used to provide the correct exhaust airflow rates. Pressure-drops were between 35 and 65 Pa, depending on where in the system they were placed. The flow exponent is 0.5 (Stifab Farex, 1995). There are no dampers in the system so the devices were used to divide the flow between the rooms and the apartments.

Infiltration

The infiltration through the walls provided 1 ACH at 50 Pa. This value does not include the flow through the door to the staircase. The crack sizes were calculated from the known leakage flow at 50 Pa, 20°C, and fitted to a curve where the exponent n=0.7 (ASHRAE Handbook, 1989). The leakage area was divided to the three bedrooms, the living room and the kitchen in proportion to their floor area.

Doors and Windows

The door height was 2.2 meters and the width was 1 meter. All doors were closed when the system was in operation. The flow through an internal door at a pressure difference of 75 Pa was 143 L/s. The flow through a door from the apartment to the staircase was 75 L/s at a pressure difference of 75 Pa. The flow exponent was 0.55 for all doors (ASHRAE Handbook, 1989). The door from the staircase to the outside had the same leakage area as the other doors in the staircase.

The discharge coefficient (C_d) for an open internal door was calculated according to Pelletret et al. (1994):

$$C_d = 0.609 \cdot Hrel - 0.066 \qquad (-) \qquad (4)$$

where *Hrel* (-) is the ratio between door height and room height.

The discharge coefficient (C_d) for an open window was calculated according to Santamouris et al. (1995):

$$C_d = 0.08 \cdot \left(\frac{Gr}{\mathrm{Re}^2} \right)^{-0.38} \qquad (-) \qquad (5)$$

where:

Re Reynolds number, dependent on wind velocity and zone depth

Gr Grashof number, dependent on window size and temperature difference

Staircase

The exhaust flow at the top of the staircase is 15 L/s, this was a typical design to ensure that the staircase is always ventilated. The volume of the staircase is 108 m^3.

Temperatures

The apartments had no cooling system. The temperature was the same in all apartments and at all heights. The staircase has a heating system but no cooling system. If the outside temperature was above 20°C, then the staircase temperature was higher. The temperature gradient is set to 0.2°C per meter, so the temperature at the top of the staircase was 2.4°C higher than at the bottom. The indoor and outside air temperatures are shown in table 1.

Table 1 Outside and indoor air temperatures

Outside temperature [°C]	Temperature at bottom of staircase [°C]	Room temperature [°C]
30	25.1	25
25	22.6	22.5
20	20	20
<20	17.6	20

Fan

The fan curve is described by the following equation:

$$\dot{m} = 0.4704 - 1.31 \cdot 10^{-5} \cdot \Delta p^2 \qquad \text{(kg/s)} \tag{6}$$

A filter was placed before the fan to protect it from particles in the exhausted air. The filter had a pressure drop of 50 Pa at 195 L/s and the flow exponent is set to 0.5.

Kitchen Hoods

Each kitchen hood contained a damper that made it possible for the resident to double the exhaust flow from 15 L/s to 30 L/s.

Wind

The wind pressure, p_v, is given by Bernoulli's equation, assuming no height change or head losses:

$$p_v = C_p \cdot \frac{\rho \cdot v^2}{2} \qquad \text{(Pa)} \tag{7}$$

where

ρ density of air, kg/m^3
v wind speed, m/s
C_p surface pressure coefficient, -

The C_p coefficients were 0.6 at the windward side, -0.35 at the lee side and -0.5 at the other two sides (Akins et al., 1979). The wind velocity profile exponent (α) was 0.14, that is for a level surface with small obstructions. The wind speeds are given at the reference height, h_{ref}, which was 10 m. The wind speed, v_h (m/s), at height, h (m), was calculated according to ASHRAE (1989):

$$v_h = v_{ref} \cdot \left(\frac{h}{h_{ref}} \right)^{\alpha} \text{ (m/s)} \tag{8}$$

Scenarios

Table 2 shows nine different scenarios that were modeled. Conditions when the system was taken into operation are shown in scenario no. 0 in Table 2. Supply air devices were open at the same level in all scenarios.

Table 2 Modeled scenarios.

Scenario no.	Exhaust [L/s] per apartment	Outside tempera-ture [^{o}C]	Infiltration [ACH@ 50Pa]	Windows	Doors	Door leakage L/s@50 Pa	Wind [m/s]
0.	45	10	1	closed	closed	75	0
1.	45	-25 - +30	1	closed	closed	75	0
2.	45	10	0-2.5	closed	closed	75	0
3.	45	-10	1	closed	closed	0-200	0
4.	45	20	1	R11 open	closed	75	0
5.	45	20	1	R11 open	R11 open	75	0
6.	K1 increased	10	1	closed	closed	75	0
7.	K1, K4 increased	10	1	closed	closed	75	0
8.	45	20	1	closed	closed	75	3, 6
9.	Decreased exhaust pressure-drops	-25 - +30	1	closed	closed	75	0

Results

Outdoor Temperature Changes, Scenario 1

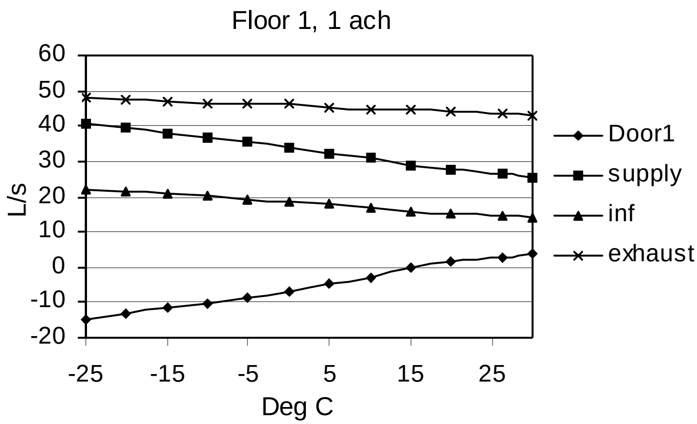

Fig. 3 Flows at floor 1 depending on outdoor air temperature. Door1 is the door to the staircase where negative flow is from the apartment to the staircase

The total exhaust airflow at floor 1, shown in Figure 3, decreases at 0.1 L/s per degree C. Line inf in Figures 3 and 4 shows the total rate of infiltration, not including doors, to each apartment. Line supply in Figures 3 and 4 is the total rate of air coming through the supply air devices in each apartment. Decreasing outdoor air temperature increases the infiltration, the supply airflow and the flow to the staircase.

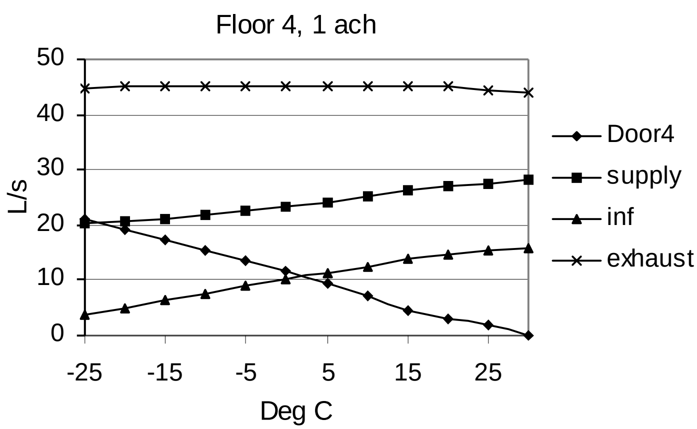

Fig. 4 Flows at floor 4. Door 4 is the door to the staircase where the flow is from the staircase to the apartment

The exhaust airflow at floor 4, shown in Figure 4, shows almost no change. Decreasing outdoor air temperature decreases the infiltration, the supply airflow and increases the flow from the staircase.

The apartment at floor 1 will be over-ventilated and the apartment at floor 4 could have poor indoor air quality. When the outdoor temperature is below 15°C, air from the apartment at floor 1 will flow into the staircase and then into the other apartments above.

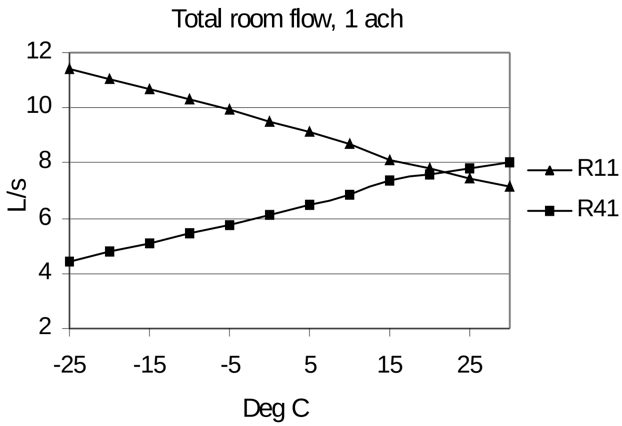

Fig. 5 Air flows through infiltration and supply air device to room 11 at floor 1 and room 41 at floor 4

Because of the increasing stack pressure when the outdoor air temperature decreases the infiltration and supply airflow at floor 1 increase (Figure 5). This increases the flow to the staircase, which increases the flow to the apartment at floor 4. The increased flow from the staircase to the apartment decreases the infiltration and the supply airflow.

The design airflow of 8 L/s to room 1 at floor 1 will be achieved when the outdoor air temperature is below 15°C if the resident chooses not to close his devices. The airflow to room 41 at floor 4 is only above 8 L/s when the temperature is 30°C or higher.

Changing Infiltration Rate, Scenario 2

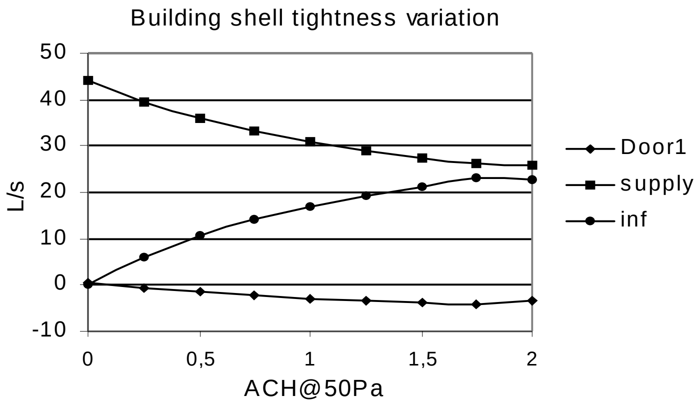

Fig. 6 The effect of making the building shell tighter. Door1 is the door to the staircase where negative flow is from apartment 1 to the staircase. Inf is infiltrated air and supply is air supplied through the supply devices

The effect on the airflow at apartment 1as a result of making the building shell tighter but not the doors to the staircase is shown in Figure 6. The building tightness decides the proportion of how much air will be supplied through the devices and how much will be supplied through infiltration. Tightening the building shell decreases the flow from the apartment at floor 1 to the staircase.

Door Tightness, Scenario 3

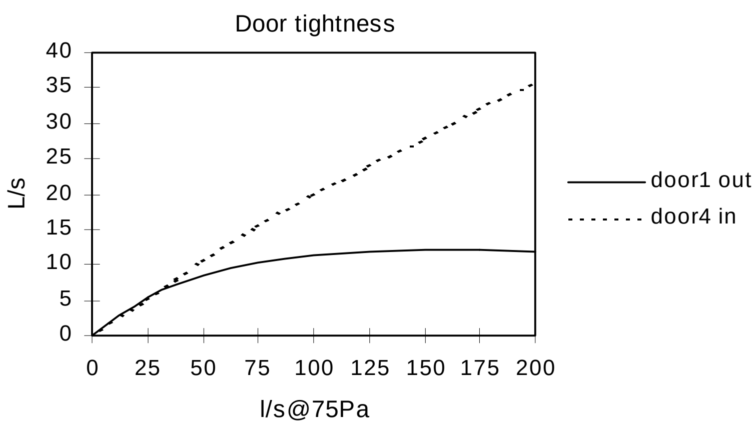

Fig. 7 The airflow out of apartment 1 through door 1 and the airflow into apartment 4 through door 4 at different levels of tightness at -10°C outdoors

Figure 7 shows the effect of tightening the doors from the apartments to the staircase and the entrance door to the staircase. Tightening the entrance door to the staircase decreases the outside stack pressure effect on the staircase. The pressure in the staircase decreases compared to the pressures in the apartments. This effect and a tighter door to apartment 4 decreases the flow from the staircase to the apartment. If the door to apartment 1 was not also tightened, then the lower pressure in the staircase would increase the flow from the apartment.

Window Opening, Scenarios 4 and 5

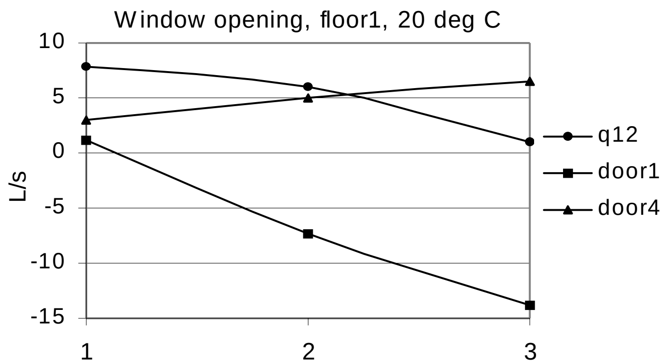

Fig. 8 q12 is the airflow through intake and infiltration to room 12 at floor 1. Airflow through door is positive when the flow is going in to the apartment

Door 1 and door 4 are doors between the apartments at floor 1 and floor 4 and the staircase. At start point 1 in Figure 8 windows and doors are closed. At point 2, the window in room 11 at floor 1 is open at 25% (22.5° door angle). At point 3, both the door and the window in room 11 are open at 25%, at this time the airflow to room 12 at floor 1 is below 1 L/s.

Increased Kitchen Flow, Scenarios 6 and 7

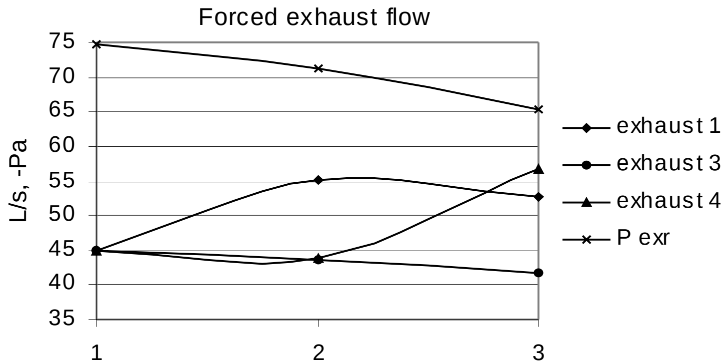

Fig. 9 Total exhaust flows and pressure before the filter (P exr)

At point 1 in Figure 9, the kitchen flow is not increased. At point 2, the flow in kitchen 1 at floor 1 is increased to 30 L/s (5 L/s is taken from the bathroom and the closet). At point 3, both kitchen 1 and kitchen 4 have increased flows. These flows, of course, also depend on the derivative of the fan curve.

Wind Effects, Scenario 8

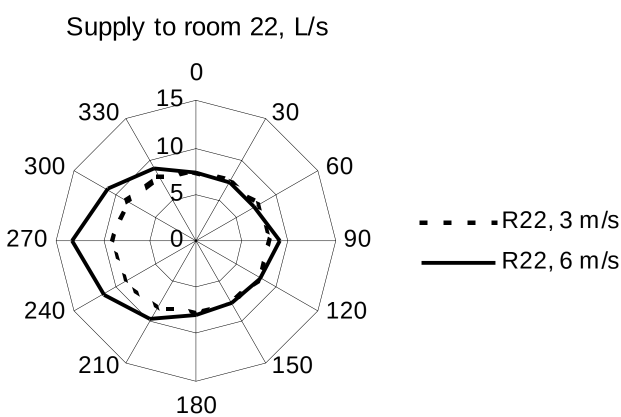

Fig. 10 Infiltration (1 ACH at 50 Pa) and flow through device depending on wind angle and wind speed. Outdoor temperature is 20°C. The Y axis' unit is L/s

The wind effect on the exhaust airflow from the apartments is very small. 0 degrees wind angle is wind blowing from south to north and 270 degrees wind angle is wind blowing from west to east. Figure 10 shows the sum of infiltrated and supplied air to room 22 at floor 2. The C_d values for the supply air devices are set to 0 (independent of wind changes). The designed airflow to room 22 is 8 L/s. At 3 m/s the flow varies depending on the wind direction between +13% and –8%, and between +65% and –10% at 6 m/s. If the supply air devices were as sensitive to the wind (C_d device = C_d infiltration) as the infiltration, then these flow variations would be more than the double.

Decreasing Pressure Drops, Scenario 9

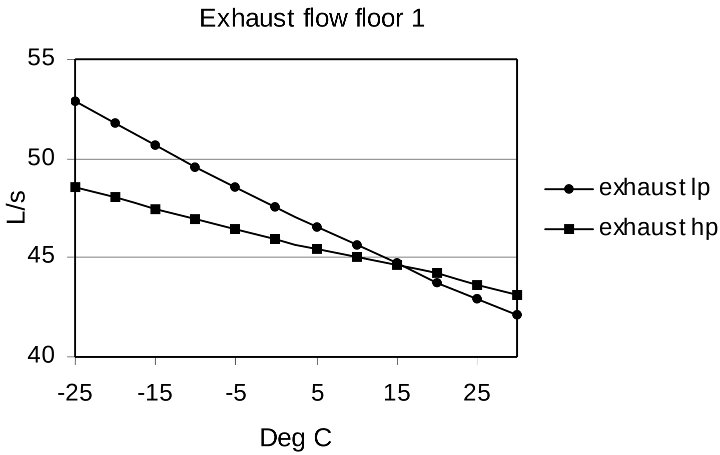

Fig. 11 Decreased pressure drops at exhaust air devices. High pressure (hp) and low pressure (lp)

Decreasing the pressures in the system at the same air flow rates by decreasing the fan speed and the pressure-drops at the exhaust air devices to the level where an increased kitchen flow still can be achieved increases the flow variations as shown in Figure 11. The average pressure reduction at the exhaust air devices is –57%. The flow change depending on temperature is now 0.2 L/s per degree C, compared to 0.1 at higher pressures. This pressure reduction makes it possible to decrease the fan speed by 10%. Decreasing the pressures also increases the flow variations when the kitchen flow is increased.

Conclusions

The supply airflow rates are sensitive to changes such as wind, temperature differences and window opening. The airflow rates to ventilated areas are almost always either too low or too high and can not fulfill the regulations. Low airflow rates can lead to poor indoor air quality and high rates increase energy losses. The exhaust airflow rates are stable and do fulfill the regulations.

If the residents do not regulate their supply air devices properly, then the only way to prevent air going from one apartment to another is to have tight doors to the staircase. The tightness of the doors in the staircase has a major impact on the amount of supplied outdoor air at the upper floors. Decreasing pressure drops at the exhaust devices increases the exhaust airflow variations. The system does not manage to increase the exhaust kitchen airflow with 100% in 50% of the apartments at the same time. Opening a window and a door in one room causes other rooms to be almost without supply air and it also increases the air exchange between the apartments. Therefore, my suggestion is that different ventilation techniques should be classified depending on how much the inter zonal airflow varies when outdoor conditions and resident behavior changes. This classification must also take into account how tight the building construction is.

Discussion

The use of standards and regulations defining indoor air quality and energy efficiency is ineffective if the system is not stable, i. e., sensitive to disturbances such as weather changes and resident behavior. A way of classifying different ventilation strategies should be introduced. This classification should consider the system sensitivity to outdoor changes, window opening, resident's behavior and risk of air exchange between different zones within the building. A stability definition would help the customer to specify his demands without specifying a certain ventilation strategy.

If the resident at floor 1 does not close his supply air devices there will be an exchange of air between the floors when the outdoor temperature is below 15°C. This might happen because he might choose to increase the power of the radiators if he experiences draught.

Sometimes air from the staircase cannot count as a supply of outdoor air because it might contain contaminated air from other apartments or become contaminated in the staircase. If this is considered, then the amount of supply air is reduced from 45 L/s to 37.8 L/s at floor 4 (1 ACH infiltration at 50 Pa) when taken into operation.

If the infiltration air, which might be contaminated from the wall materials of the building, is not counted as a supply of outdoor air, then the supply of outdoor air is reduced to 25.2 L/s, 56% of the required flow.

Tightening the building would increase the flow through the supply air devices and this might cause thermal comfort problems. The sensitivity for wind in individual rooms would decrease if the building where tighter and if the supply air devices are not sensitive to wind changes.

Decreasing pressure drops at the exhaust devices in order to save energy and reduce noise makes the system even more sensitive to changes. The intended energy savings might not occur since the flow will increase during colder seasons.

Figure 9, increased kitchen flow, also indicates what would happen, if for instance, a resident experiences the noise too loud in the bathroom and therefore decides to decrease the pressure drop at the exhaust device. This would affect the other apartments exhaust airflow. A way to reduce this problem and to make increased kitchen flow possible is to install a variable speed fan and keep the pressure constant before the filter. This would also maintain the exhaust airflow when the pressure drop at the filter is increasing with time and create possibilities for demand controlled ventilation in the bathrooms.

If mechanical exhaust system are used then care should be taken into make the doors as tight as possible and not place a mail drop in the door. The residents should be informed about how the system works and how their action might affect the airflow in their apartment and in the building.

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III

Paper III

Stability of Mechanical Exhaust and Supply Systems

By

Fredrik Engdahl

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STABILITY OF MECHANICAL EXHAUST AND SUPPLY SYSTEMS

ABSTRACT

The mechanical supply and exhaust system is a common ventilation technique in multi-family buildings in Europe. The objective of this paper is to determine how well the system meets indoor air quality standards and regulations and how sensitive it is to disturbances, such as window opening, temperature differences, wind and how building tightness affect the airflow. The effects of these parameters are compared with the effects in an identical building but with a mechanical exhaust system. A simple model of a multi family building has been simulated with a multi zone infiltration program (COMIS). The system is not as sensitive to disturbances as the mechanical exhaust system. The customer should be aware of these differences between different systems and therefor the systems should be divided into different stability classes.

KEYWORDS

Ventilation, Mechanical supply and exhaust, Multi family building, Modeling, Infiltration, Stability

INTRODUCTION

Standards and national regulations (ASHRAE, 1989; Boverket, 1994) specify and recommend airflow rates to achieve a certain energy efficiency and indoor air quality. A system design goal is to achieve these airflow rates or a higher level of energy efficiency and indoor air quality specified by the customer. Once the system is in operation a technician to ensure that it fulfills the design criteria checks it. The system may at this time satisfy customer specifications and the regulations, but it may not perform well when the occupants move in and when outdoor conditions change. No standard or regulation gives the customer the ability to specify the system stability, that is, the sensitivity to disturbances such as weather changes, resident behavior and how much it depends on the building tightness.

A previous paper (Engdahl, 1999) shows that mechanical exhaust systems are not stable. The mount of supplied air varies a lot with outdoor conditions and occupant behavior. Another way of ventilating a building is the mechanical

supply and exhaust. This system design is used for multi-family buildings, offices and schools.
It is easy to measure the air flows and the system gives opportunities such as heat recovery and the possibility to cool or heat the supply air. Air is exhausted from the kitchen, the bathroom and the clothes closet, and is supplied in the bedrooms and the living room. The systems are often poorly maintained (Engdahl, 1998) and operated improperly, because often no instructions are given. Especially in schools and multi family buildings the performance is low. This way of supplying the air has advantages compared to the slot devices (Krüger, 1996) (Krüger and Kraenzmer, 1996) used in mechanical exhaust systems. The possibility to heat the air and the design of the supply devices makes the risk of draught less compared to the slot devices. A problem that might occur is a higher sound level.

A model of a four-story multi-family building with a staircase and one apartment on each floor is used to examine how the airflow changes under different conditions. The multi-zone infiltration program COMIS 3.0 (Feustel and Rayner-Hooson, 1990; Feustel and Smith, 1997), developed by nine countries at Lawrence Berkeley Laboratory, is used to simulate the model.

METHOD

A model of a four-story multi-family building with a staircase and one apartment at each level is examined. The system is modeled with COMIS 3.0, a multi zone infiltration program based on the mass balanced equation (Feustel and Rayner-Hooson, 1990). The equations used by COMIS are described in detail in a previous paper (Engdahl, 1999) where the same method was used.

Figure 1 shows the apartment at floor one, where R12 is room 2 at floor 1, LR1 is living room at floor 1, K1 is kitchen at floor 1, T1 is bathroom at floor 1 and Door1 is the door between the apartment at floor 1 and the staircase. Every floor consists of an identical apartment. There is no internal leakage between the rooms and no leakage through the roof.

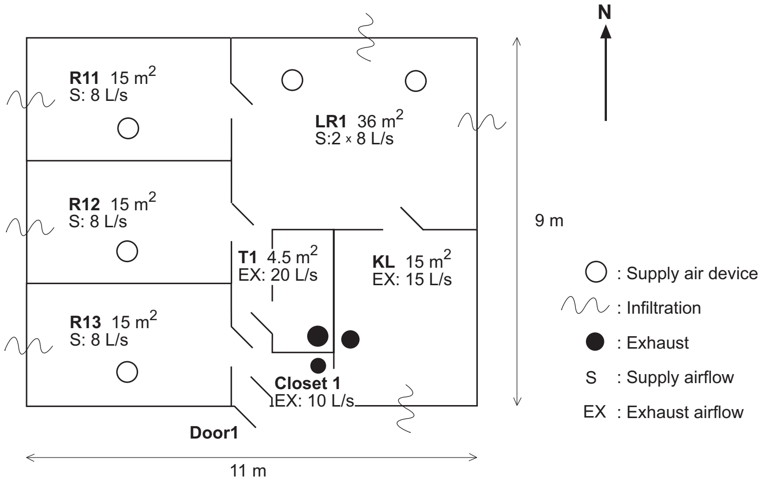

Figure 1: The apartment at floor one with supply, infiltration and exhaust flows

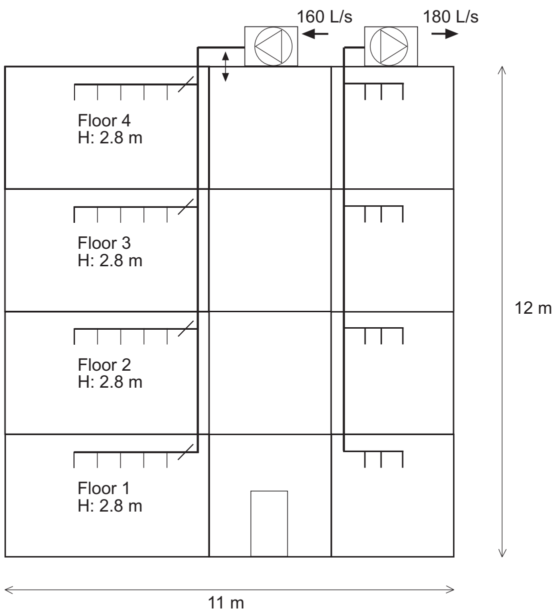

Figure 2: Front view of the building and the ventilation system.

Ventilation

The system is designed to supply airflow of 8 L/s to the bedrooms, R11, R12 and R13 and 16 L/s to the living room, LR1 in figure 1. 15 L/s air is exhausted from the kitchen (K1), 20 L/s from the bathroom (T1) and 10 L/s from the closet (Closet1). The exhaust airflow rates fulfill Swedish regulations [14] and ASHRAE Standard 62-1989 (ASHRAE, 1989). The supply airflow rates to the bedrooms fulfill the Swedish regulations for a two-person bedroom (8 L/s) and the ASHRAE standard (7.5 L/s) for one-person bedroom.

The supply airflow rate to the living room, 16 L/s, fulfills the ASHRAE standard (0.35 ACH) and the Swedish regulations (0.35 L/sm^2). The total exhaust airflow in each apartment is 45 L/s, 0.584 ACH. The supply and exhaust system is showed in figure 2.

Temperatures

The apartments have no cooling system. The temperature is the same in all apartments and at all heights. The staircase has a heating system but no cooling system. If the outside temperature is above 20°C, then the staircase temperature is higher. The temperature gradient is set to 0.2°C per meter, so the temperature at the top of the staircase is 2.4°C higher than at the bottom. The indoor and outside air temperatures are shown in table 1.

TABLE 1

OUTSIDE AND INDOOR AIR TEMPERATURES

Outside temperature [°C]	Temperature at bottom of staircase [°C]	Room temperature [°C]
30	25.1	25
25	22.6	22.5
20	20	20
<20	17.6	20

Scenarios

Table 2 shows seven different scenarios that were modeled. Conditions when the system was taken into operation are shown in scenario no. 0 in table 2. When taken into operation the total amount of supplied air at each floor is 40 L/s and 45 L/s is exhausted. All flows, except for the mechanical exhaust, are

TABLE 2
MODELED SCENARIOS.

Scenario no.	Ventilator in staircase	Outside tempera-ture [^{o}C]	Infiltration [ACH@ 50Pa]	Windows	Doors	Door leakage *L/s@50* Pa	Wind [m/s]
0.	Yes	10	1	closed	closed	75	0
1.	Yes	-25 - +30	1	closed	closed	75	0
2.	Yes	-10	0.2-2.0	closed	closed	150-30	0
3.	Yes	20	1	R11 open	closed	75	0
4.	Yes	20	1	R11 open	R11 open	75	0
5.	Yes	20	1	closed	closed	75	3, 6
6.	Yes/No	-25-30	1	closed	closed	75	0

RESULTS

Outdoor temperature changes, scenario 1

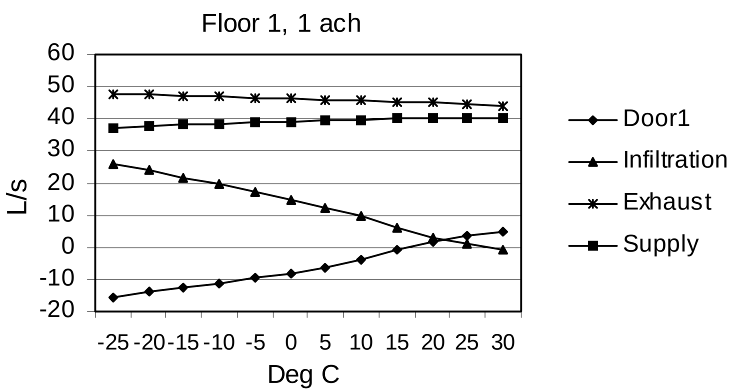

Figure 3: Flows at floor 1 depending on outdoor air temperature. Door1 is the door to the staircase where negative flow is from the apartment to the staircase.

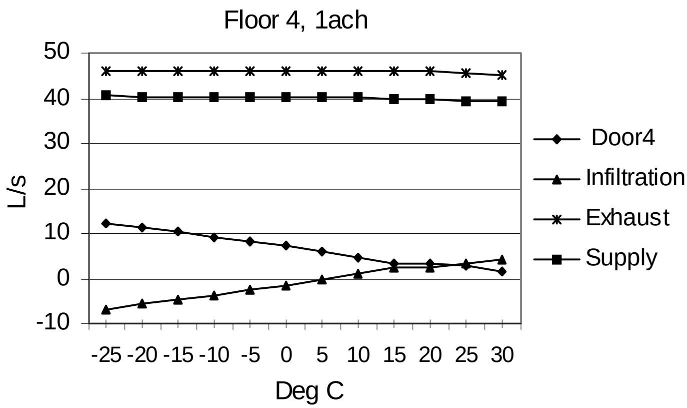

Figure 4: Flows at floor 4. Door4 is the door to the staircase where the flow is from the staircase to the apartment.

Line Infiltration in fig. 3 and 4 shows the total rate of infiltration, not including doors, to each apartment. Line Supply in fig. 3 and 4 is the total rate of air coming through the supply air devices in each apartment. Decreasing outdoor air temperature increases the infiltration and the flow to the staircase at floor 1. Decreasing outdoor air temperature decreases the infiltration to the apartment and increases the flow from the staircase at floor 4. The supply and exhaust flow is almost constant at both floors.The apartment at floor 1 will be over-ventilated by infiltration and the apartment at floor 4 will have an over pressure in some rooms. When the outdoor temperature is below 15 °C, air from the apartment at floor 1 will flow into the staircase and then into the other apartments above.

Changing infiltration rate, scenario 2

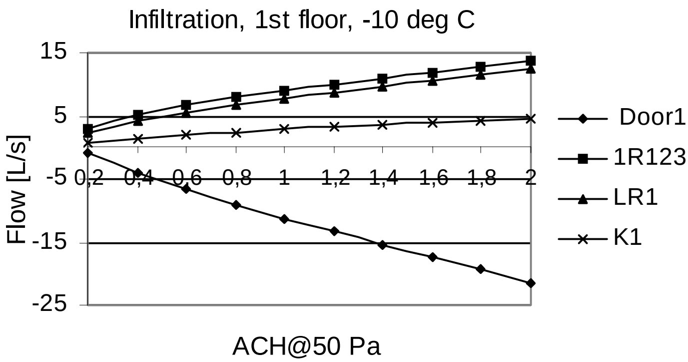

Figure 5: The effect at floor 1 of making the building shell and doors tighter. Door1 is the door to the staircase where negative flow is from apartment 1 to the staircase.

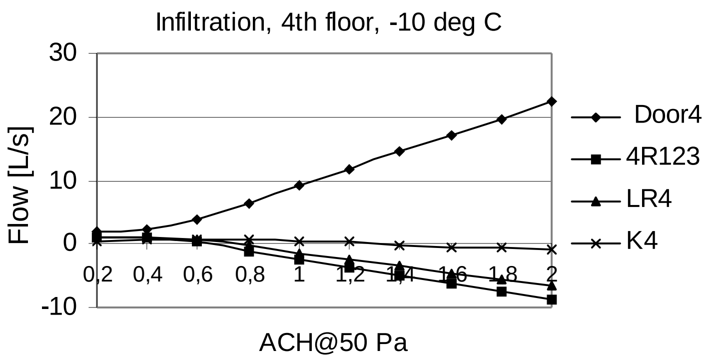

Figure 6: The effect at floor 4 of making the building shell and doors tighter. Door4 is the door to the staircase where negative flow is from apartment 4 to the staircase.

The effect on the airflow, at floor 1, of making the building shell and doors tighter is shown in fig. 5. The door between the apartment (Door1) and the staircase at floor 1 must almost be tight to prevent air going out of the apartment at –10 degrees C. Line 1R123 is the sum of infiltration to R11, R12 and R13. The air change rate at 50 Pa must be less then 0.6 to prevent an overpressure in R41, R42 and R43.

Window opening in R12 and wind effects, scenario 3, 4 and 5

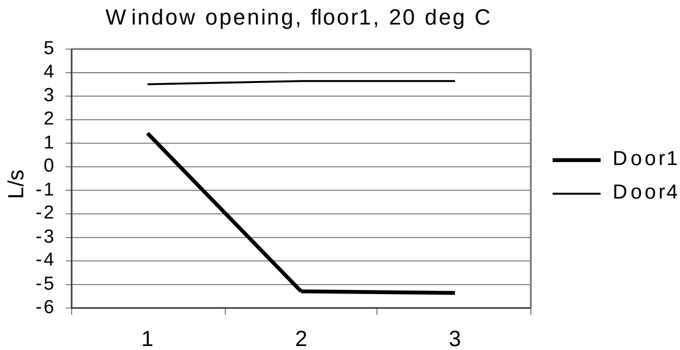

Figure 7: Airflow through door is positive when the flow is going in to the apartment.

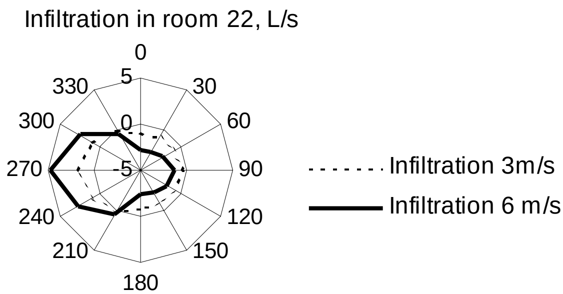

Figure 8: Infiltration (1 ACH at 50 Pa) depending on wind angle and wind speed. Outdoor temperature is 20 °C. The Y axis' unit is L/s.

Door 1 and door 4 are doors between the apartments at floor 1 and floor 4 and the staircase. At start point 1 in fig. 7 windows and doors are closed. At point 2 the window in room 12 at floor 1 is open at 25% (22.5° door angle). At point 3 both the door and the window in room 12 is open at 25%. The window opening does almost not affect the supply air to the rooms next to R12.

The wind effect on the exhaust airflow from the apartments is very small. 0 degrees wind angle is wind blowing from north to south and 270 degrees wind angle is wind blowing from west to east. Figure 8 shows the infiltrated air to room 22 at floor 2. At a wind speed of 6 m/s the wind will extract air from the room at 240 degrees out of 360 degrees.

With and without ventilator in staircase, scenario 6

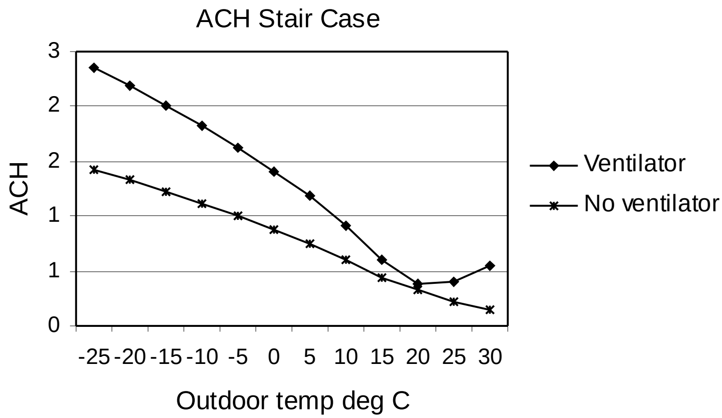

Figure 9: Total air change rate in staircase, with and without ventilator in staircase.

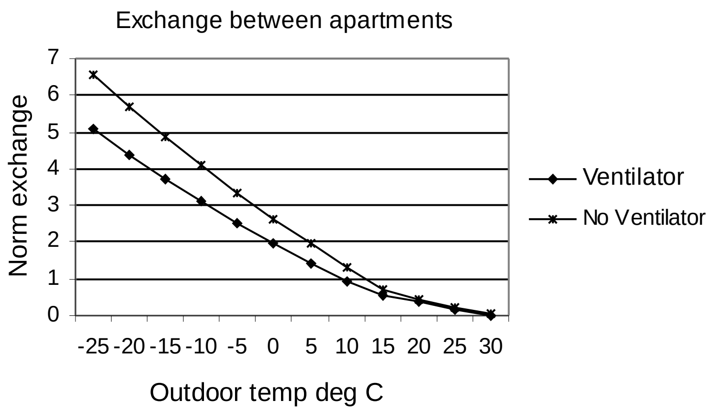

Figure 10: Normalized air exchange between apartments, with and without ventilator in staircase.

As shown in figure 9 and 10 the ventilator increases the air change in the staircase and decreases the air exchange between the apartments.

CONCLUSIONS

The supply and exhaust airflow rates are not sensitive to changes such as wind, temperature differences and window opening. The exhaust and supply airflow rates are stable and do fulfill the regulations. There is almost always exchange of air between the apartments. Window opening almost only affects the flow in the apartment where the window is opened. Opening a window does not cause other rooms in the apartment to be without supply air as in the case with a mechanical exhaust system.

The use of a ventilator in the staircase reduces the air exchange between the apartments but it also increases the air change rate and by that the energy consumption. The wind pressure causes easy overpressure in rooms where the air is supplied through supply air devices.

Comparing these results with the mechanical exhaust system shows that there is a need for a classification of different ventilation techniques. The techniques should be classified depending on how much inter zonal airflow varies when outdoor conditions and resident behavior changes. This classification must also take into account how tight the building construction is.

DISCUSSION

The use of standards and regulations defining indoor air quality and energy efficiency is ineffective if the system is not stable, that is, sensitive to disturbances such as weather changes and resident behavior. A way of classifying different ventilation strategies should be introduced. This classification should consider the system sensitivity to outdoor changes, window opening, resident’s behavior and risk of air exchange between different zones within the building. A stability definition would help the customer to specify his demands without specifying a certain ventilation strategy.

The overpressures in rooms where the air is supplied causes the air to flow from the room to the outside, this might in some climates create problems with moister in the faced construction. If a resident changes the pressure drop over a supply or exhaust device is a problem that remains with the mechanical supply and exhaust system. This change of pressure drop will affect other apartments’ airflow. A way to reduce this problem is to install a variable speed fan and keep the static pressure constant in the main ducts. This would also maintain the airflow when the pressure drop at the filter is increasing with time and create possibilities for demand controlled ventilation. Keeping the static pressure constant also gives the possibility to decrease pressure drops and thereby the sound level. When mechanical supply and exhaust system are used then care should be taken into make the doors as tight as possible and not place a mail drop in the door. The residents should be informed about how the system works and how their action might affect the airflow in their apartment and in the building.

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IV

Paper IV

Pressure Controlled Variable Air Volume System

By

Fredrik Engdahl and Anders Svensson

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Pressure Controlled Variable Air Volume System

Abstract

There is one primary goal and several secondary goals to fulfill when designing a heating, ventilation and air conditioning (HVAC) system. The main goal is to create an indoor climate that satisfies the user and this should be achieved in the most energy efficient way and with a system that also function in a long term perspective. Higher supply air flow have the potential of increasing office workers performance and decreasing short-term sick leave. To increase the supply air flow, a cooling function is added to the air and 100% outdoor air is supplied. The main objective of this paper is to give the fundamentals for a system design that takes good indoor climate, energy efficiency and long term operation into account. When controlling the static pressure at branch level, it is possible to vary the air flow to different zones without measuring the individual flow and without affecting the air flow to other zones. An equation for the placement of the pressure sensor is shown and the realtive flow difference between the diffusers on the branch is calculated and measured. These principals are the fundamentals for a pressure controlled Variable Air Volume system (VAV). The fan pressure set point is optimized resulting in a decreased fan power requirement and sound generation. The flow to each zone is controlled at the diffuser outlet which prevents draft and therefore a wide range in supply air temperature can be used.

1. Introduction

Most scientific papers about ventilation focus on either system energy performance or indoor climate even though the result of a system design is an indoor climate at a certain energy cost. A system design that creates a good indoor climate is not interesting if it uses much energy, and vice versa. A system should not only operate well during startup, it should also operate well over its entire life time. Most systems do not have a long term operation [1, 2]. The main objective of this paper is to give the theoretical fundamentals for a system design that takes good indoor climate, energy efficiency and long term operation into account.

When designing a ventilation system there is one primary goal and two secondary goals to achieve. The primary goal is to

- Get as many as possible of the users satisfied
  and the two secondary goals are to
- To achieve the primary goal at a low energy use, that is fan energy and cooling/heating energy
- Make the HVAC system user-friendly when it comes to operation and maintenance and by this achieve a long term operation

1.1 User satisfaction

There are several parameters that have to be taken into account when it comes to user satisfaction. One important parameter is the supply air flow. The main reason for supplying air is to create a healthy environment for the occupant. This is achieved by supplying enough air to keep pollutants from materials and occupants at an acceptable and healthy level.

Weschler and Shields [3] showed that there are chemical reactions among indoor pollutants that generates reactive and irritating products. For this reaction to occur, there must be sufficient time for the pollutants to interact. The supply air flow rate determines the time available for such interactions, that is, the supply air flow rate influences indoor chemistry. A higher air flow rate does not only decrease the indoor pollutant concentrations. It also limits reactions among the indoor pollutants. Low pollutant concentrations can be achieved either by increasing the supply air flow or by reducing the pollutants, but the best results will appear when doing both.

Fanger and Wargocki et al. [4, 5] showed that for each twofold increase in ventilation rate, the performance of office workers improved on average 1.7%. The results also showed that an increased outdoor air supply rate decreased the percentage of subjects dissatisfied with the air quality, increased the perceived freshness of air and decreased the sensation of dryness of the mouth and throat. The increased outdoor air supply rate also eased difficulty in thinking clearly and made subjects feel generally better.

Milton et al studied [6] the short-term sick leave of 3700 employees in 40 buildings depending on supply air flow. The ventilation was rated as moderate at 12 l/s and person and high at 24 l/s and person. The employees that were exposed to moderate outdoor supply rates had a 35% attributable risk of short-term sick leave compared to those who where exposed to high supply air flow.

A way to increase the supply air flow is to add one more function to the supply air, that is room temperature control by varying the supply air flow in spaces with varying heat loads. By using outdoor air for cooling, the room temperature set point will be reached with an improved indoor air quality, assuming the outdoor air is fresher than the extracted air. When the supply air is used for cooling, the air flow must be high enough to meet the cooling need in the room.

The relative humidity should be kept above 30% [7, 8] to prevent dehydration and discomfort. The risk of supplying dry air is most common during cold periods when the outdoor air is heated and then supplied to the room at a low relative humidity. Therefore, it should be possible to lower the air flow during cold periods.

According to ASHRAE standard 62 [9] the indoor concentration of carbon dioxide should be kept below 1000 ppm and approximately 10% will be dissatisfied if the concentration is 700 ppm.

The supply air temperature is another parameter that has to be considered. The supplied air temperature should be low enough so that the cooling power is ample to achieve the room temperature set point, decided by the user. The temperature must not be lower than the current dew point temperature in the ventilated zone to avoid condensation. The risk of feeling drafts [10] is higher when the supplied air temperature is low, which must also be taken into consideration. Additionally the air velocity should be kept as low as possible in the occupied zone.

The only now known way of achieving a high satisfaction level, is to let the individuals decide and control their own room air temperature and air flow. A study [11] showed that users want a greater degree of individual or group control, and systems that are easy to operate. Both indoor temperature and indoor air quality should be controlled by the user.

One of the major problems in today's mechanical ventilation systems is that they generate too much noise [12]. Standards and national regulations usually set the limits for noise in the frequencies that are possible for the ear to register and not for other frequencies. A study [13] showed that low frequency noise (< 200 Hz) interferes with performance and effects the health. The air should be distributed and removed as silently as possible. Major sources to noise generation are the fan, the duct system with dampers and the diffusers. Noise can be reduced by sound attenuaters which create a higher pressure loss than a straight duct. The most energy efficient way to reduce noise is to reduce the production of noise, and that is noise of any frequency.

1.2 Energy use

When designing a building and its systems the fundamental element for energy savings is good building design. Outdoor heat sources like solar radiation and indoor heat sources like lighting and computers should be minimized and, if possible, techniques like night cooling should be used to decrease the energy use for cooling.

The energy input to a HVAC system can be divided into two parts, cooling and heating of air and fan energy. In many premises like offices and hospitals in northern Europe there is a cooling need a major part of the year, even though the outside air temperature is below the required indoor temperature. When the outside temperature is lower than the required temperature in the building, outside air can be used for cooling with a decreased cooling energy use as a result. This is called free cooling. For this reason free cooling should be used whenever possible. When heating is needed, the energy should be taken from the exhausted air with a heat exchanger and, if possible, no heating coil should be used in order to reduce pressure losses. The potential of energy savings with heat exchangers in HVAC systems is shown by Jagemar [14]. The other part of the energy input is to use fan energy to transport the air through the supply system to the conditioned spaces and then exhaust it. The common VAV system is based on a constant static pressure in the main duct after the fan and then VAV units including the controllers, air flow measuring equipment and the dampers. The static pressure must be kept at a level that covers the pressure losses in the system at maximum load. This results in unnecessary energy use at every load below maximum and a generation of noise from the fan and the duct system. Therefore, the pressure level in the system should be kept as low as possible. Adopting the static pressure set point depending on load in order to reduce energy use has been applied on VAV systems by Wang and Burnett [15].

Air leaking from ducts increases the air flow through the HVAC unit which results in an increased energy use of both fan energy and energy needed to achieve the required supply air temperature. By keeping the pressure low, air leaking from ducts will decrease which will in return decrease the energy use compared to a system with a constant static pressure set point in the main duct.

Ventilation and cooling should only occur when there is a demand and where it is demanded, everything else is a waste of energy. This strategy is called demand controlled ventilation (DCV). Energy use could be minimized by reducing the ventilation air flow when only half of the office workers are in the office. When a cell office or any other room is not occupied, the air flow should be reduced to a rate that is just enough to remove building material emissions. It has been shown in papers published as International Energy Agency (IEA) documents [16, 17, 18] that the use of occupancy sensors and carbon dioxide sensors have a high potential of reducing energy use. Reducing the air flow as much as possible in the whole system also makes it as silent as possible due to low air velocities and fan speed.

How well a system is balanced also affects the energy use. Depending on flow difference between supply and exhausted air, temperatures, facade construction and wind air might be infiltrated or exfiltrated through the facade. If the outside air must be heated before being supplied to the building, then infiltrated air will either be heated in the room or it will lower the exhaust air temperature which is used to heat the supply air. This will increase the energy use and therefore infiltration should be avoided in this situation. Similarly this situation will occur when cooling a building in a hot and humid outdoor climate. The other thing to take into account is the risk of condensation in the facade construction. If it is colder on the outside than on the inside, the relative humidity in the air passing the facade from the inside to the outside (exfiltration) will increase and there is a risk for condensation or to high relative humidity in the facade. Infiltration creates the same risk when it is warm and humid outside. In a warm and humid climate the supply air flow should be higher than the exhaust air flow in order to create an overpressure in the building. This avoids infiltration, which in this case might cause condensation in the facade and an increase in energy use. In a cold climate there should be a negative pressure in the building to prevent condensation in the facade but the pressure difference should be as small as possible to reduce the energy loss due to infiltration. It is important that the HVAC system manage to keep the difference between supply and exhaust air flow during its life time and this is mostly not the case in today's systems [1, 2].

1.3 Make the HVAC system user-friendly when it comes to operation and maintenance in order to achieve a long term function

The layout of offices changes with time and different users have different needs [11]. The building owner must adopt the building and its HVAC system to the new user. This costs and therefore the HVAC system must be easy to change without having to re-commissioning the entire HVAC system. When parts of the building or individual rooms are not used, it should be possible to shut off these parts of the system without affecting the rest of the system. The system should be easy to inspect in order to make it easy to detect errors and to make changes in the system during its life time. Potential sources of errors should be minimized.

Lack of maintenance [1, 2] is a common reason for system failures and therefore, critical parts like the HVAC plant should be easy to access. When commissioning the system there should be no need for traditional, time consuming, commissioning methods. In order to make the system easy to inspect and to track possible errors, it should be built with a graphic computer-interface.

Dust and particles in the duct system might affect the sensors and result in a false value. This will change the balance between supply and exhaust air flow. To prevent this from happening, the components used in the system should not be sensitive to dust and the number of flow or pressure measuring points should be kept at a minimum. Nor should the sensors need to be cleaned or have filters that must be replaced every year.

1.4 System design

A system with static pressure control at the branches is one way to enable variable air flow from several diffusers using only one pressure sensor per branch. In the design process, duct diameters, pressure sensor location and air flow differences have to be considered. These parameters are analyzed and described. The relative air flow difference between the diffusers on a branch depends on the diameter, placement of pressure sensor and the air velocity. The supply air flow from a branch with 5 diffusers have been measured to confirm the calculations. Diffusers with variable outlet area are used for air flow control to improve the indoor climate by decreasing the risk of draft. To reduce the power requirement during partial load, an optimization of the fan static pressure set point is described.

Nomenclature

a	Numerater in CO_2 concentration calculation (mg/s)
A	Surface area inside the ventilated space on which pollutant can be absorbed (m^2)
A_k	The quotient between the duct area after and before a transition (-)
A_{max}	Diffuser outlet area at maximum air flow (m^2)
A_0	Diffuser outlet area (m^2)
b	Denominater in CO_2 concentration calculation (m^3/s)
C_c	Diffuser constant (-)
C_{SS}	Steady-state indoor concentration of pollutant (mg/m^3)
C_x	Outdoor concentration of pollutant (mg/m^3)
d	Duct diameter (m)
e	Relative air flow difference (-)
E_v	Ventilation effectiveness (-)
G_o	Generation rate for pollutant by an occupant (mg/s)
k	Constant depending on main duct construction (-)
k_c	Volume flow constant for diffuser ($l/(s\cdot Pa^{0.5})$)
k_d	Deposition velocity on A for pollutant (m/s)
L	Distance from first bifurcation on branch (m)
L_t	Total length of duct (m)
n	Number of branches (-)
N	Number of occupants (-)
p_{dyn}	Dynamic pressure (Pa)
p_{static}	Static pressure in branch (constant) (Pa)
p_t	Total pressure (Pa)
P	Filter penetration for pollutant (%)
P_{conv}	The power used to transport the air through the duct system (not including the air handling unit) when not optimizing the pressure (W)
P_o	Penetration of pollutant through human lung (%)

P_{opt}	The power used to transport the air through the duct system (not including the air handling unit) when optimizing the pressure (W)
q	Air flow (m^3/s)
q_{branch}	Air flow in one branch (m^3/s)
q_i	Infiltration air flow (m^3/s)
q_{max}	The highest diffuser air flow from the branch duct to a diffuser (l/s)
q_{min}	The lowest air flow from the branch duct to a diffuser (l/s)
q_o	Average respiratory flow for a single occupant (m^3/s)
q_v	Ventilation air flow (m^3/s)
Re	Reynolds number (-)
s	Branch duct characterstic constant (-)
t_{c_in}	Temperature cold side in (°C)
t_{c_out}	Temperature cold side out (°C)
t_{h_in}	Temperature hot side in (°C)
u	Air velocity (m/s)
u_0	Air velocity in branch duct at L=0 (m/s)
v_{max}	Maximum outlet air velocity (m/s)
v_0	Outlet air velocity (m/s)
x_m	Throw length (m)
x_{max}	Throw length at maximum air flow rate (m)
x_r	Relative throw length compared to throw length at maximum air flow (-)
x_s	Position on branch duct for pressure sensor (-)
x_t	Position on branch duct for transition (-)

Greek letters

ΔL	Length of duct (m)
Δp	Pressure loss (Pa)
Δp_{diff}	Total pressure loss over supply air diffuser (Pa)
Δp_{main}	Pressure loss main duct (Pa)
Δp_{sd}	Pressure loss straight duct (Pa)
Δp_{90}	Pressure loss when turning 90° in a T-junction (Pa)
Δt_0	Temperature difference between supply air and room air (°C)
ε	Material absolute roughness factor , $4.6 \cdot 10^{-5}$(m)
η_t	Temperature efficiency (-)
λ_f	Duct friction coefficient (-)
λ_0	Duct friction coefficient, independent of velocity ($m^{0.2}/s^{0.2}$)
ρ	Air density (kg/m^3)
ν	Kinematic viscosity, $1.5 \cdot 10^{-5}$ (m^2/s)
ξ	Pressure loss coefficient (-)

2. Method

In sections 2.1 and 2.2 the basic equations used by the flow calculation program PFS are described. In 2.3 the theory for a branch with intended constant static pressure and constant diameter is developed. The location of the pressure sensor, expected differences in air flow and the use of a transition are described in sections 2.4 to 2.6. Section 2.7 describes the measurements of the air flow from a duct with a constant diameter to 5 diffusers. Sections 2.8 to 2.11 show equations for diffusers with variable outlet area, indoor air quality, static pressure after fan and heat exchanger temperature efficiency.

2.1 Pressure drop at T-junction

To calculate the air flow and pressure losses in the circular duct system, a computer program called PFS [19] was used. The following equations [20] are used by PFS.

$$\Delta p = \xi \cdot p_{dyn} = \xi \cdot \frac{\rho \cdot u^2}{2} \qquad \text{[Pa]} \qquad (1)$$

The pressure loss coefficient, x, when going straight through the T-junction, is calculated with Eq. (2) and with Eq. (3) when turning 90 degrees.

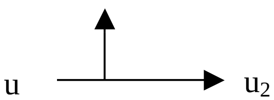

Fig. 1. Velocities in T-junction Eq. (2).

$$\xi_1 = 0.35 \cdot \left(\left| \frac{u_2}{u_1} - 1 \right| \right)^{1.5} \qquad \text{[-]} \qquad (2)$$

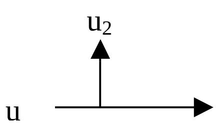

Fig. 2. Velocities in T-junction Eq. (3).

$$\xi_2 = 0.52 \cdot \left(\left| \frac{u_2}{u_1} - 0.55 \right| \right)^{1.5} + 0.90 \qquad [-] \qquad (3)$$

2.2 Pressure loss in straight duct

The pressure loss through a straight duct, Δp_{sd}, is calculated with Eqs.(4)-(6).

$$\Delta p_{sd} = \lambda_f \cdot p_{dyn} \cdot \frac{\Delta L}{d} \qquad [\mathrm{Pa}] \qquad (4)$$

$$\frac{1}{\sqrt{\lambda_f}} = -2 \cdot \log^{10} \left(\frac{\varepsilon / d}{3.71} + \frac{2.51}{\mathrm{Re} \cdot \sqrt{\lambda_f}} \right) \qquad [-] \qquad (5)$$

For Re > 3500

$$\mathrm{Re} = \frac{u \cdot d}{\nu} \qquad [-] \qquad (6)$$

2.3 Constant static pressure in circular branch duct with constant diameter

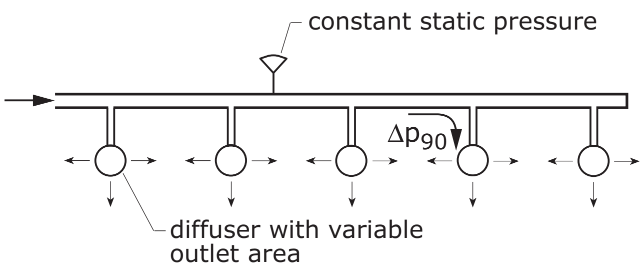

Fig. 3. Duct with constant diameter with 5 diffusers and the static pressure constant at one point.

The air flow through a diffuser is dependent on the total pressure loss, Δp_{diff}, and the design of the diffuser, k_c in Eq. (7). By changing the design of the diffuser, in this case the outlet area, the air flow will change.

$$q = k_c \cdot \sqrt{\Delta p_{diff}} \qquad [\mathrm{l/s}] \qquad (7)$$

Knowing the pressure, the air flow can be calculated depending on the outlet area. If the static pressure is controlled to be constant (Fig. 3) at one position on a branch with several diffusers, the differences in pressure loss from this position to the diffusers have to be negligible in order to use the constant pressure setpoint to determine the air flow in all diffusers. The differences in diffuser air flow will depend on changes in static pressure in the branch and on changes in dynamic pressure. The main assumption is that the static pressure in the branch is the determinant parameter for the air flow at a diffuser with a known outlet area. This is true if the dynamic pressure in the branch is equal to the pressure loss in the T-junction. By studying the difference between the dynamic pressure in the branch and the pressure loss through the T-junction when turning 90° (Eqs. (1) and (3)), it can be verified that the differences in pressure are in the range of -1 to 2.5 Pa (Fig. 4). This is valid for a branch air velocity varying between 1 and 8 m/s and a velocity of 3 m/s at the diffuser. The velocity 3 m/s is a commonly used design criterion due to sound requirements. If this pressure difference is negligible depends on the pressure that is used by the diffuser which in practice is at least 20 Pa. Branch duct air velocities higher than 3.4 m/s will result in a higher pressure at the diffuser than for velocities below 3.4 m/s.

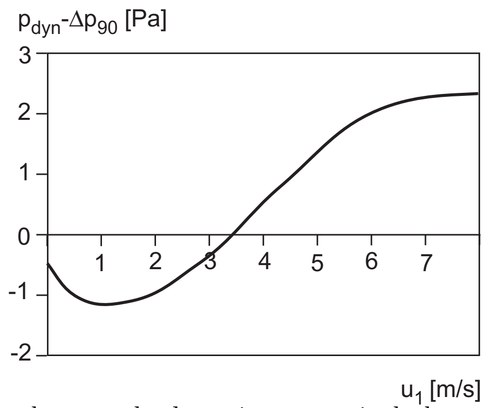

Fig. 4. The difference between the dynamic pressure in the branch duct and the pressure loss over the T-junction when turning 90° as a function of the air velocity (u_1) in the branch duct. The velocity after turning 90° is 3 m/s (Eqs. 1 and 3).

If every diffuser is receiving equal air flow from the branch duct, the air velocity will decrease according to a staircase as shown in Fig. 5. To make calculations possible, the decrescent velocity is approximated according to Eq. (8) and shown in Fig. 5 with a straight line.

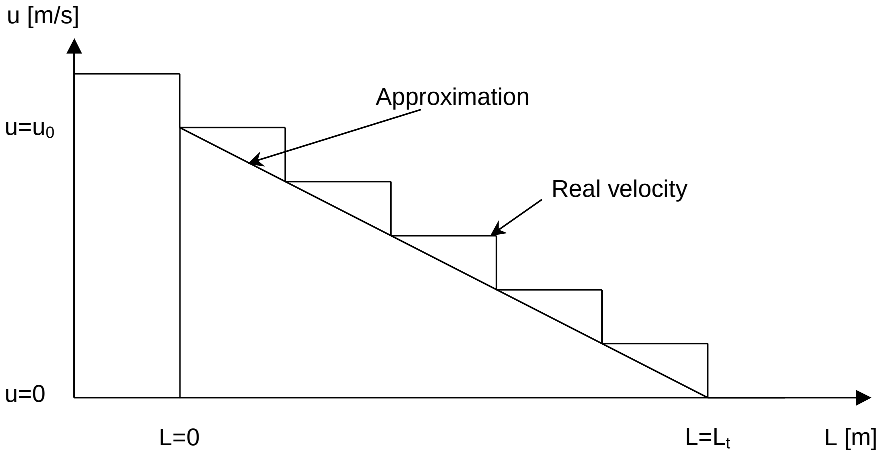

Fig. 5. Real and approximated air velocity decrescenting in a duct with six diffusers.

$$u(L) = u_0 \cdot \left(1 - \frac{L}{L_t}\right) \qquad \text{[m/s]} \qquad (8)$$

The straight duct pressure loss described in Eq. (4) has to be rewritten for a decreasing velocity. This is described by Eq. (9) where *L* equals zero at the beginning of the branch. The pressure losses when when going straigh through the T-junctions are not included.

$$\Delta p_{sd} = \frac{\rho \cdot u_0^2}{2} \cdot \int_0^L \left(1 - \frac{L}{L_t}\right)^2 \cdot \frac{\lambda(u,d)}{d} \cdot dL \qquad \text{[Pa]} \qquad (9)$$

The friction loss coefficient, λ, is dependent of the velocity according to Eqs. (5) and (6). For velocities between 1 and 8 m/s, the coefficient's velocity dependence can be simplified and described according to Eq. (10), where λ_0 is dependent on the duct diameter. This is shown in Fig. 6.

$$\lambda = \frac{\lambda_0}{u^{0.2}} \qquad \text{[-]} \qquad (10)$$

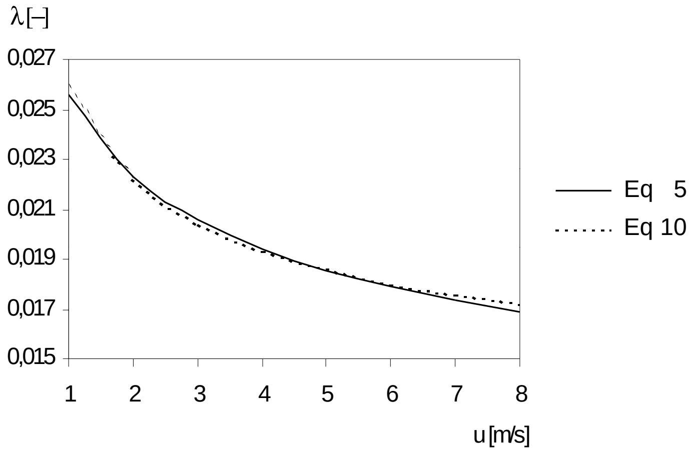

Fig. 6. Friction loss coefficient λ as a function of air velocity (d=0.315 m λ_0=0.0257 $m^{0.2}/s^{0.2}$) depending on Eq. (5) and Eq. (10).

Equations (9) and (10) results in Eq. (11).

$$\Delta p_{sd} = \frac{\rho \cdot u_0^2}{2} \cdot s \cdot \left(1 - \left(1 - \frac{L}{L_t}\right)^{2.8}\right) \qquad \text{[Pa]} \qquad (11)$$

Where:

$$s = \frac{\lambda_0 \cdot L_t}{2.8 \cdot d \cdot u_0^{0.2}} \qquad \text{[-]} \qquad (12)$$

The difference in dynamic pressure between L=0 and L is described by Eq. (13).

$$\Delta p_{dyn} = \frac{\rho \cdot u_0^2}{2} \cdot \left(1 - \left(1 - \frac{L}{L_t}\right)^{2}\right) \qquad \text{[Pa]} \qquad (13)$$

If the straight duct pressure loss for the total duct length (L_t) equals the difference in dynamic pressure, this would result in an "ideal" duct diameter for the specific total length. If L equals L_t and Eq. (11) is equal to Eq. (13) then this results in: s=1.

The dimensionless quotient, s, can be used to identify whether the static pressure is increasing (s<1) or decreasing (s>1) in the branch.

2.4 Placement of pressure sensor

The objective is to get as equal flow as possible in the diffusers. The pressure is kept constant at one point x_s (Eq. (15)). At this point the static pressure difference from the first diffuser to the point is equal to the static pressure difference from the point to the last diffuser according to Eq. (14). With this location, the difference between the diffuser supplying the highest air flow and the designed air flow will be equal to the difference for the diffuser supplying the lowest air flow. This minimizes the air flow difference.

$$\Delta p_{sd_{0\rightarrow L}} - \Delta p_{dyn_{0\rightarrow L}} = \Delta p_{sd_{L\rightarrow L_t}} - \Delta p_{dyn_{L\rightarrow L_t}} \quad \text{[Pa]} \quad (14)$$

$$x_s = \frac{L}{L_t} \quad \text{[-]} \quad (15)$$

Using Eqs. 9-15 results in Eq. 16.

$$2 \cdot s \cdot (1 - x_s)^{2.8} - 2 \cdot (1 - x_s)^2 - s + 1 = 0 \quad \text{[-]} \quad (16)$$

2.5 Air flow difference

If the pressure is kept constant at the point described by Eq. (16) then the relative air flow difference can be calculated according to Eq. (17).

$$e = \sqrt{\frac{\Delta p_{diff} + \frac{\rho \cdot u_0^2}{4} \cdot |1 - s|}{\Delta p_{diff}}} - 1 \quad \text{[-]} \quad (17)$$

In Eq. (17), Δp_{diff} is the total pressure loss for the diffuser and the duct between the branch and the diffuser. The velocity u_0 is according to Fig. 5 the velocity after the first diffuser.

2.6 Branch duct with two diameters

A transition to a duct with less area can be used to decrease the space requirements of the duct when the static pressure is increasing (s<1). The transition should be placed at the position (x_t) on the branch where the reduction of the static pressure when passing the transition (equals the increase in dynamic pressure) equals the difference in static pressure from the first bifurcation to the position. This is described with Eq. (20). The duct area before the transition is A_1, the area after is A_2 and the quotient is described in Eq. (18). When reducing with one dimension this quotient is around 0.65 for standard dimensions.

$$A_k = \frac{A_2}{A_1} \qquad [-] \qquad (18)$$

The decrease in static pressure when passing the transition at the length L is described in Eq. (19).

$$\Delta p_{A_1 \rightarrow A_2} = \frac{\rho \cdot u_0^2}{2} \cdot \left(1 - \frac{L}{L_t}\right)^2 \cdot \left(\frac{1}{A_k^2} - 1\right) \qquad [\mathrm{Pa}] \qquad (19)$$

The decrease in Eq. (11) should respond to the increase in static pressure from the beginning of the branch to the length L.

$$\Delta p_{dyn_{0\rightarrow L}} - \Delta p_{sd_{0\rightarrow L}} = \Delta p_{A_1 \rightarrow A_2} \qquad [\mathrm{Pa}] \qquad (20)$$

Rewriting the equations results in Eq. 21.

$$s \cdot (1 - x_t)^{2.8} - \frac{1}{A_k^2}(1 - x_t)^2 - s + 1 = 0 \qquad [-] \qquad (21)$$

2.7 Air flow measurements

The air flow to each diffuser were measured on a branch duct with five diffusers, shown in Fig.7. The diffusers had a pressure loss of 34 Pa at 65 l/s, which is the designed air flow. The branch duct is 14 m long (L_t) and has a diameter (d) of 0.315 m. The diameter of the duct between the branch and the diffuser is 0.160 m. The pressure sensor is placed at 4.6 m (x_s=0.33). The pressure is measured at diffuser 1 to 5 with a manometer (Digitron type P600) and kept constant at x_s. The air flow error in the measurements is approximately 5%.

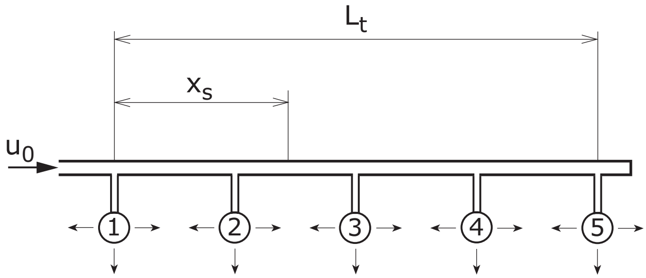

Fig. 7. Branch duct with diffusers (1-5) used for air flow measurments. x_s is the pressure sensor location, L_t the length of the branch duct and u_0 the inlet air velocity.

Air flow measurements were made for all diffusers at maximum air flow and for each diffuser when all the others were closed (0 l/s).

2.8 Diffusers

Equation (22) [21] was used to show the throw length, x_m, depending on the outlet area, A_0. It is assumed that the diffuser constant, C_c, was constant even though the outlet area changes.

$$\frac{x_m}{\sqrt{A_0}} = 2.1 \cdot C_c \left(\frac{v_0^2}{\Delta t_0 \cdot \sqrt{A_0}} \right)^{0.5} \qquad [-] \quad (22)$$

$$q = v_0 \cdot A_0 \qquad [m^3/s] \quad (23)$$

A constant diffuser outlet area, A_0, results in Eq. (24) which describes the relative throw length, x_r.

$$x_r = \frac{x_m}{x_{\max}} = \frac{v_0}{v_{\max}} \qquad [-] \quad (24)$$

A constant outlet air velocity, v_0, results in Eq. (25).

$$x_r = \frac{x_m}{x_{\max}} = \sqrt{\frac{A_o}{A_{\max}}} \qquad [-] \quad (25)$$

2.9 Indoor Air Quality

The indoor air quality is shown by calculations of the steady-state indoor concentration of carbon dioxide, C_{SS} [22].

$$C_{SS} = \frac{a}{b} \qquad [\mathrm{mg/m^3}] \quad (26)$$

Where

$$a = C_x \cdot \left(q_i + \frac{0.01PE_v q_v}{1-0.01P(1-E_v)} \right) + (G_i + NG_o) \qquad [\mathrm{mg/s}] \quad (27)$$

$$b = q_i + q_v + k_d A + Nq_o(1-0.01P_o) \qquad [\mathrm{m^3/s}] \quad (28)$$

2.10 Heat exchangers

To show available supply air temperature, t_{c_out}, depending on the heat exchanger temperature efficiency, η_t, and outdoor temperature, t_{c_out}, Eq. (29) [23, 24] was used. It is assumed that the hot fluid capacity rate equals the cold fluid capacity rate.

$$\eta_t = \frac{t_{c\_out} - t_{c\_in}}{t_{h\_in} - t_{c\_in}} \qquad (29)$$

2.11 Optimization of static pressure after fan

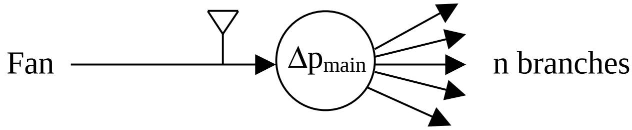

Fig.8. Schematic figure of a supply air system. The static pressure is set after the fan because there is a pressure loss in the main duct when the flow is divided to the branches. The branches have a constant static pressure.

Assuming the supply air system and loads are symmetric then the power, P_{conv}, the fan must deliver to force the air through the duct system can be calculated according to Eq. (30). The maximum air flow sets the static set point (the sum of pressure losses, the static pressure, p_{static}, in the branch and the dynamic pressure, p_{dyn}, in the branch at maximum air flow) which will be constant and not depend on the air flow, q_{branch}.

$$P_{conv} = n \cdot q_{branch} \cdot \left(\Delta p_{main} + p_{static} + p_{dyn}\right) \quad \text{[W]} \quad (30)$$

When the static set point is optimaized with respect to the power requirement, the only thing that is not dependent on the air flow rate is the constant static pressure in the branches. Equation (31) is used to calculate the power, P_{opt}, the fan must deliver to force the air through the duct system when the static set point is optimized.

$$P_{opt} = n \cdot q_{branch} \cdot \left(\Delta p_{main}\left(q_{branch}\right) + p_{static} + p_{dyn}\left(q_{branch}\right)\right) \quad \text{[W]} \quad (31)$$

The pressure loss in the main duct, Δp_{main}, is assumed to depend on the air flow as shown in Eq. (32).

$$\Delta p_{main} = k \cdot n^2 \cdot q_{branch}^2 \quad \text{[Pa]} \quad (32)$$

3. Results

In Fig. 9 a general outline of a system is shown. When controlling the static pressure at the branch duct, as shown in Fig. 9, it is possible to vary the air flow to the zones connected to the branch duct, only using one pressure sensor and without measuring the air flow to each zone.

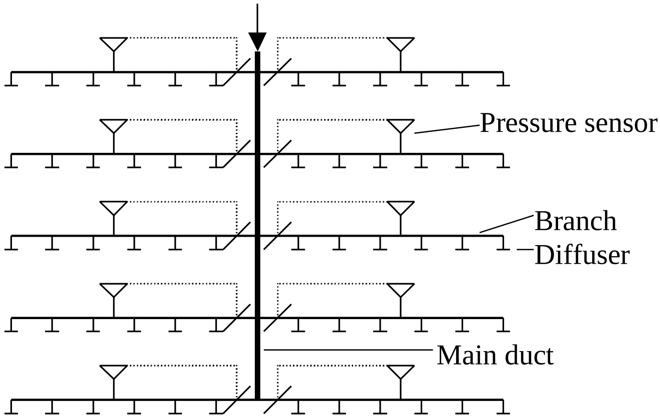

Fig. 9. Supply air system with main duct, branches, dampers, pressure sensors and diffusers with variable outlet area.

The air flow is to each zone is varied by changing the outlet area of the supply air diffuser, a certain outlet area and static pressure in the branch duct will equal an air flow. Sections 3.1-3.3 describe the design parameters, 3.4 the air flow measurments and 3.5 compares PFS calculations with the equations developed in the Method chapter. Section 3.6 shows the throw length of an active diffuser (variable outlet area), 3.7 the IAQ when cooling with air, 3.8 the possibility not to use a heating coil when having a heat exchanger with high efficiency and finally 3.9 shows the fan pressure optimization.

3.1 Controlled static pressure in circular branch duct with constant diameter

When *s* equals 1 in Eq. (12) an “ideal” duct diameter can be calculated depending on air velocity and branch duct length. The diameters that correspond to *s* equals 1 for three air velocities are shown in Fig. 9.

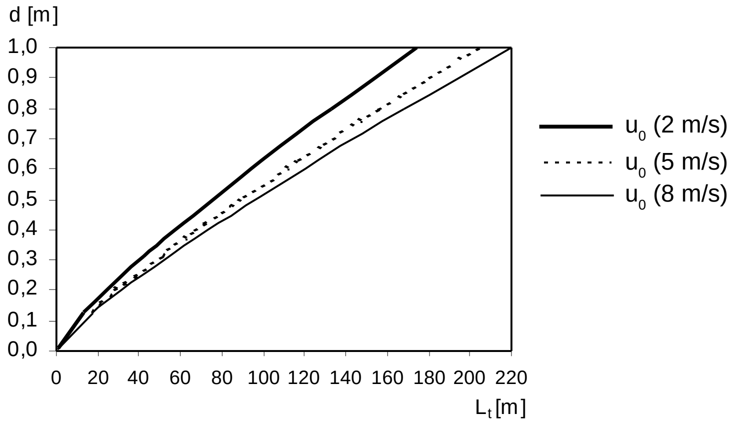

Fig. 10. “Ideal” duct diameter as a function of the duct total length at three different air velocities (u_0).

It can be observed in Fig. (10) that for diameters above 0.315 m the branch has to be longer than 50 m in order to result in a branch in which the static pressure is decreasing (s>1).

Table 1 shows the different λ_0 that are to be used when calculating s for different diameters. The correlation coefficient is above 0.99 for all the λ_0 in Table 1.

Table 1. Duct friction coefficient, λ_0, for different standard dimensions of duct diameter

d [m]	λ_0 [$m^{0.2}/s^{0.2}$]
0,2	0,028543
0,25	0,027096
0,315	0,025707
0,4	0,024378
0,5	0,023226
0,63	0,022114
0,8	0,021045
1	0,020114

3.2 Placement of pressure sensor

In Fig. 11 the solution to Eq. (16) is shown. The pressure sensor should be placed at x_s ($0 \leq x_s \leq 1$) and for s equals 1 the placement is irrelevant.

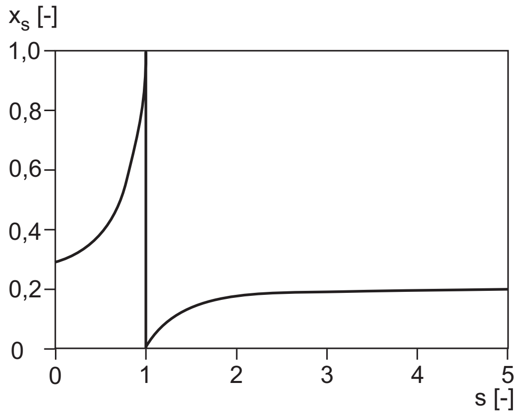

Fig. 11. Pressure sensor position (x_s) as a function of *s* (Eq. 16).

For a duct with increasing static pressure (s<1) the sensor should be placed between 0.29 (=1-0.5$^{0.5}$) and 1 depending on *s*. For a duct with decreasing static pressure it should be placed between 0 and 0.22 (=1-0.5$^{1/2.8}$) depending on *s*.

3.3 Branch duct with different diameters

The solution for where the transition should be placed (Eq. (21)) is shown in Fig.12 for A_k=0.65.

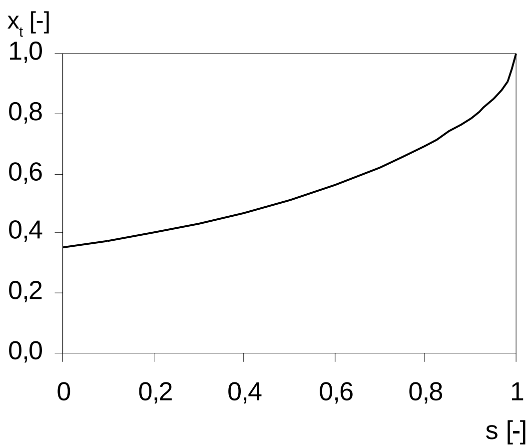

Fig. 12. Placement of transition (x_t) as a function of *s* (A_k=0.65), $0 \leq x_t \leq 1$.

The placement of the pressure sensor has now to be changed. A new x_s has to be calculated based on the velocity after the transition and a new length has to be used that is based on the duct length after the transition.

3.4 Air flow measurements

The calculated air flow in the tables are when using PFS to calculate the air flow. Table 2 is when all the diffusers are at maximum air flow. The static pressure is set to give diffuser 2 the designed air flow of 65 l/s.

Table 2. All diffusers at maximum air flow

Diffuser	1	2	3	4	5
Measured pressure [Pa]	22,5	23,0	26,9	26,5	23,1
Measured air flow [l/s]	64,0	64,7	70,0	69,5	64,9
Calculated air flow [l/s]	64,0	64,9	65,7	66,5	66,4
Relative difference in calculated and measured air flow [-]	0,000	0,002	-0,065	-0,045	0,023

Using Eq. (17) with *s*=0.30659 and u_0=3.336 results in an air flow difference of 3%. Using PFS results in an air flow difference of 2%. Taking Fig. 4 into account (the difference in dynamic pressure) in Eq. (17) the result would be equal to the PFS value. The measured difference is 9% (difference between diffuser 3 and 1) The difference in the measured values can be explained by the method error and by the fact that it is not a ideal “duct” with identical ducts between branch and diffuser.

Table 3 shows the air flow when one diffuser was at maximum flow and the rest were closed (0 l/s). Comparing these values with those in Table 2 gives the range in which the flow to each diffuser will vary depending on the other diffusers air flow. Also in this table the calculated air flow are when using PFS.

Table 3. One diffuser at maximum air flow and the other diffusers at 0 l/s

Diffuser	1	2	3	4	5
Measured pressure [Pa]	22,5	25,0	26,2	24,5	24,0
Measured volume flow [l/s]	64,0	67,5	69,1	66,8	66,1
Calculated volume flow [l/s]	65,0	65,0	65,3	65,2	64,3
Relative difference in calculated and measured air flow [-]	0,015	-0,038	-0,058	-0,025	-0,028

3.5 Comparison between PFS and calculation for duct (s<1) with 12 diffusers

Calculations for a duct with 12 diffusers were made with PFS to be able to compare the equations based on an approximated velocity according to Eq. (8) and Fig. 5. The distance between diffusers is 2.5 m (L_t=27.5) and the designed air flow is 55 l/s at 40 Pa (Δp_{diff}). The branch diameter, *d*, is 0.400 m. The duct between the diffuser and the branch has a diameter of 0.160 m and the length is 0 m. The air velocity after the first diffuser (u_0) is 4.81 m/s.

Using Eq. (12) with λ_0 as 0.024378 according to Table 1 results in *s* equal to 0.438. Using Eq. (16) or Fig. 11 gives a pressure sensor position (x_s) of 0.378. Calculating the air flow difference with Eq. (17) results in 0.048.

Figure 13 is generated by placing the pressure sensor at different positions and for each position calculating with PFS the first diffuser air flow relative difference (e_{min}) and the last diffusers' air flow relative difference (e_{max}). The air flow are relative to the designed air flow. Where e_{max} crosses e_{min} the plus minus relative difference is at its minimum.

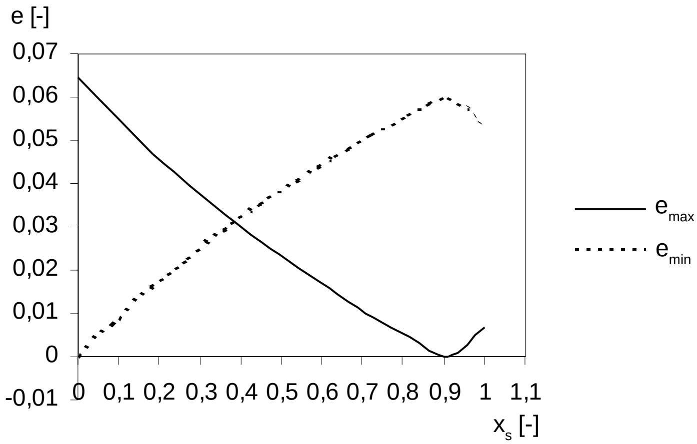

Fig. 13. Relative air flow difference (*e*) depending on where (x_s) the pressure sensor is located. e_{max} is the air flow difference at the last diffuser on the branch and e_{min} is the air flow difference at the first diffuser.

According to Fig. 13 the pressure sensor should be placed at 0.38 which agrees with Eq. (16) (0.378). The relative air flow difference is 0.032 according to Fig. 13, which is lower than the calculated value 0.048. With Fig. 4 (difference in dynamic pressure) in mind, the air flow difference would be 0.036, which is relatively close to the one shown in Fig. 13.

3.6 Supply air diffuser

Varying the opening depending on air flow (active diffuser) makes the air velocity constant when leaving the diffuser. This makes it possible to vary the flow in a wide range without any risk of draft when low air temperature air is supplied. Eqs. (22)-(25) were used to calculate the graphs in Fig. 14.

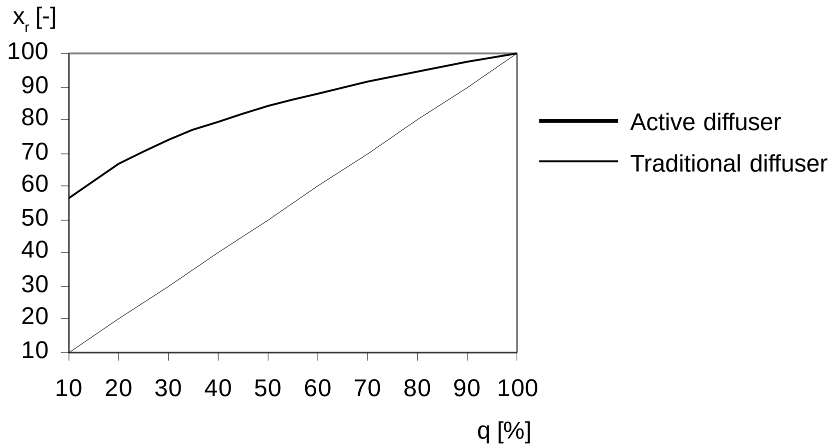

Fig. 14. Relative throw (x_r) depending on the relative air flow (q) for an active diffuser (variable outlet area) compared with a traditional diffuser.

Controlling the static pressure makes a certain opening respond to a certain flow and therefore there will be no need for measuring air flow during operation.

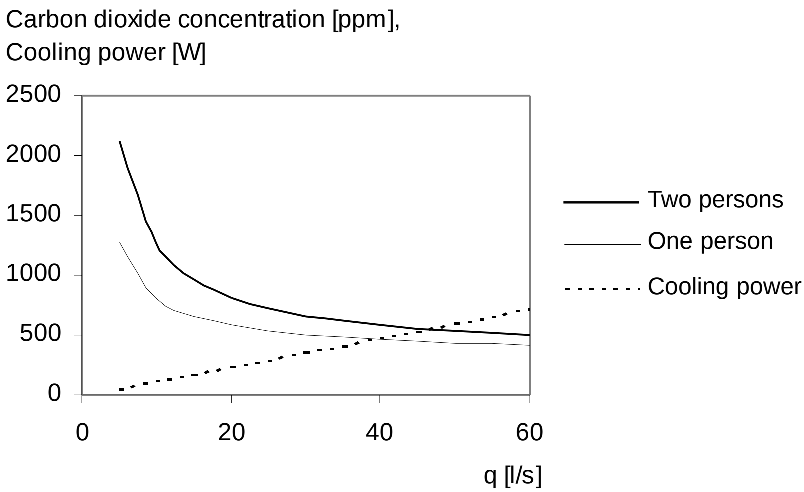

Fig. 15. Carbon dioxide concentration in a room with one or two persons and the supplied cooling power. Outdoor carbon dioxide concentration is 340 ppm. No air leakage or infiltration is assumed.

Equations (26)-(28) were used to calculate the graphs in Fig. 15. If carbon dioxide is used as an indicator of indoor air quality (IAQ) in a single office room the level will be below 600 ppm during the major part of the cooling season (q>20 l/s) even if there are two persons in the room. The dotted graph in Fig. 6 that shows the supplied cooling power is calculated for a supply air temperature 10ºC below room temperature.

3.8 Heat exchanger

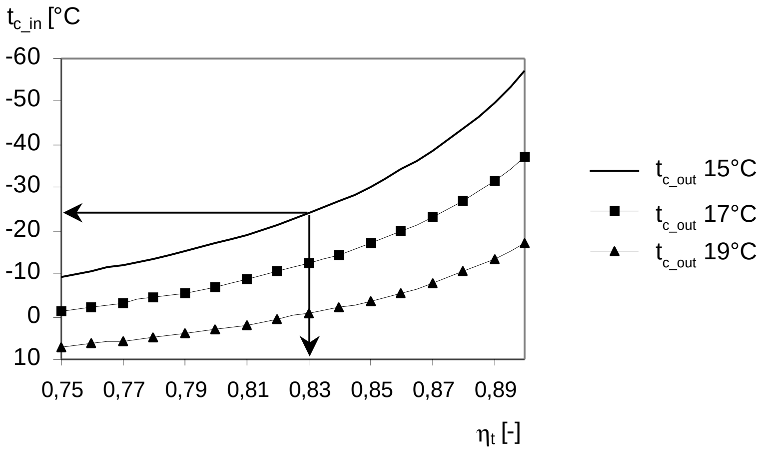

Fig 16. Lowest possible outdoor temperature to maintain 15, 17 or 19ºC as supply air temperature as a function of different temperature efficiencies. The exhaust air temperature (t_{h_in}) is assumed to be 23ºC.

Heat exchangers of the rotating wheel type may have a temperature efficiency of around 80% [4] and even higher at reduced air flow. If the air can be supplied at 15ºC without causing any draft, a temperature efficiency of 83% would be enough to heat an outdoor temperature of minus 23ºC. This would in most cases make it possible to not have a heating coil in the HVAC unit, which leads to a reduction in the pressure loss and energy consumption. The radiators in the room must be designed to heat up the 15ºC supply air to the room temperature set point. Equation (29) was used to calculate the graph in Fig. 16.

3.9 Optimization of static pressure after fan

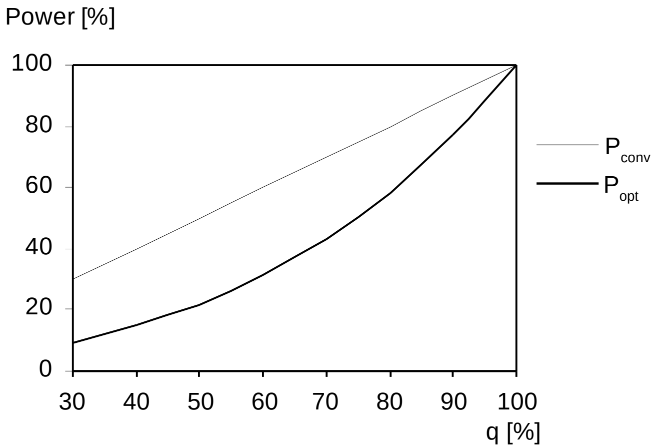

Fig. 17. The power required to transport the air through the duct system, i.e. immediately after the HVAC unit to the room. P_{opt} is the power used with optimized static pressure set point and P_{conv} is without the optimization.

Figure 17 shows the power used to transport the air from the HVAC unit to the room at symmetric load and air distribution. Equations (30)-(33) were used to calculate the graphs. The charts in Fig. 8 are based on a system with 10 branches, a maximum air velocity of 6 m/s and a constant static pressure of 40 Pa. The pressure loss in the main duct at maximum air flow (10 m^3/s) is 100 Pa. The power the fan has to deliver to force the flow through the duct system to the room is 50% of maximum power at 50% flow for the conventional system. The system with optimization of the static pressure set point only uses approximately 21% of the power used at maximum air flow.

4. Discussion

Higher supply air flow have the potential of increasing office workers performance and decreasing short-term sick leave. To increase the air flow, a cooling function is added to the supply air. Many premises have high internal heat loads, are exposed to solar radiation and are well insulated, which results in a need for cooling during the majority of the year. Using a 100% outdoor air system for cooling will increase the user satisfaction because of an improved perceived indoor air quality. Having a minimum air flow of 20 l/s when the room is occupied will result in a carbon dioxide concentration of 580 ppm. If the room is entered by an extra person this will result in an extra 100 W of heat load which will increase the flow rate by 8 l/s and the carbon dioxide concentration will still not exceed 700 ppm.

As mentioned before, there is not only a cooling need during summer conditions, this makes it possible to use the lower outdoor temperature to cool the building without using any cooling energy. The only energy used during these conditions is the fan energy. The fan energy is reduced by designing a system that only cools and ventilates where and when it is needed combined with an optimization of the static pressure set point. The controllers keeping the pressure constant in the branches send digital messages telling their damper positions. The static pressure after the fan is then controlled on the damper that is opened the most.

Occupancy sensors should be used to save energy and reduce sound generated in the HVAC system. When the room is unoccupied, the air flow is just enough to remove the building pollutants. When occupied, the air flow is increased to remove pollutants from both the building and the occupants. After that, any increased air flow is due to a cooling need. By using digital controllers or intelligent sensors it is possible to not only use the occupancy sensors to reduce the flow but also use the same sensor to control the lighting and for burglary alarm.

When the building is unoccupied, a low air flow is supplied. During such conditions the energy use is very low due to low air flow and pressure optimization and therefore there is very little to gain by shutting the HVAC system off and by that risk the “Monday morning sickness”. This also eliminates the use for time controlling the HVAC unit.

When the air is used for cooling the air flow through the HVAC unit during heating season is just the air flow to remove pollutants from the building and the occupants. The rotating wheels in today’s HVAC units are dimensioned for about 70-80% of maximum air flow. A lower air flow will increase the temperature efficiency. So either can the heat exchanger manages a lower outdoor temperature or it can be shorter and by that achieving a lower pressure loss. In regions where the outside air temperature never is below 15ºC, the rotating wheel and supply air diffusers that do not cause drafts makes the heating coil unnecessary. When the outdoor temperature is higher than the exhaust temperature, the wheel can also be used to cool the supply air before entering the cooling coil.

The system described only needs one pressure sensor in each branch duct to vary the supply air flow to several zones. A parameter, *s*, depending on branch diameter, length and velocity can be used to decide whether the static pressure will increase or decrease in the branch (Fig. 10). When it comes to practical applications the pressure will probably increase (s<1). Knowing *s* and the pressure loss at the diffusers the relative flow difference on the branch can be calculated with Eq. (17). The placement of the pressure sensor also depends on *s* and Fig. 11 should be used to minimize the air flow difference. To decrease space requirements the branch duct diameter can be reduced at a position showed in Fig. 12. If the transition is placed according to Fig. 12, the air flow difference will not increase. There is a possibility to decrease the air flow span when using several transitions. The equations based on a continiously decreasing velocity (Fig. 5) correlates well with the calculations made with PFS. The relatively high differece (6.5%) between measured and calculated air flow at diffuser 3 in Fig. 7 can depend on the accuracy in Eq. (3) and on differences in fittings, ducts and diffusers. More accurate measurments have to be carried out to determine this. In future work the continuously decreasing velocity should be applied on exhaust systems.

Diffusers that supplies constant air flow can be combined with variable air flow diffusers without any constant air flow devices when the static pressure is controlled at the branch. Diffusers can be added or removed without any need for commissioning the whole system. This makes the system very flexible and user friendly for the maintenance organization.

To assure long term stability the sensors should be static sensors, which are superior to the dynamic sensors that are often used. Using digital controllers that can communicate makes it possible to inspect the system with a graphic interface.

The most economical way to balance the system is to supply the air in every room and then let the air flow through the corridor to be exhausted at just one or two locations. The exhaust should be located at places where the pollution production is high, for example in printing and copying rooms. The supply air flow is measured at the beginning of the branch and then this value is received by a slave unit on the exhaust branch. This will result in one extra pressure sensor in the supply branch and one in the exhaust branch but will save ducts and exhaust air devices. One must keep in mind that smoking must not be allowed in the office.

In most cases, the duct system will require more space than, for instance, a conventional system with constant air flow and chilled beams; but on the other hand it will reduce a lot of plumbing and also the risk of having a liquid medium like water. The use of occupancy sensors makes it possible to decrease the dimensions of the system depending on the expected activity. The space used for shaft can be reduced by the fact that every zone will not require maximum air flow at the same time and when the static pressure set point is optimized the noise generation and energy use will still be low at most loads. In Sweden, for example, the room heights in cell offices tends to increase and the corridors are kept with a lower room height which makes it possible to have larger ducts in the corridors.

With the diffusers described, it is possible to vary the supply air temperature in a wide range without causing any risk of draft. To improve the total system energy efficiency and the possibility to satisfy as many occupants as possible, the supply air temperature should be optimized.

Acknowledgements
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Paper V

Optimal Supply Air Temperature with Respect to Energy Use in a Variable Air Volume System

V

By

Fredrik Engdahl and Dennis Johansson

Submitted for publication in Energy and Buildings

Optimal Supply Air Temperature with Respect to Energy Use in a Variable Air Volume System

Abstract

In a variable air volume system (VAV) with 100% outdoor air, the cooling need in the building is satisfied with a certain air flow at a certain supply air temperature. To minimize the system energy use, an optimal supply air temperature can be set dependent on the load, specific fan power, chiller coefficient of performance, outdoor temperature and the outdoor relative humidity. The theory for an optimal supply air temperature is presented and the heating, ventilation and air-conditioning (HVAC) energy use is calculated depending on supply air temperature control strategy, average U-value of the building envelope and two outdoor climates. The analyses show that controlling the supply air temperature optimally results in a significantly lower HVAC energy use than with a constant supply air temperature. The optimal average U-value of the building envelope is in practise mostly zero.

1. Introduction

The main reason for using heating, ventilation and air conditioning (HVAC) systems is to satisfy users when it comes to health, indoor air quality (IAQ) and thermal comfort. A variable air volume system (VAV) satisfies the health criterion and IAQ by supplying a minimum amount of air flow based on national regulations and standards. When there is a cooling need, the thermal comfort is satisfied by increasing the air flow and supplying enough air colder than the room temperature. When the heat load increases in a zone controlled by a VAV system, the flow increases. A room controller controls the air flow to the room by measuring the room air temperature and the supply air flow. The supply air flow depends on the load and temperature difference between the zone and the supply air. A low temperature of the supply air requires a lower air flow than a high supply air temperature does. The supply air temperature is controlled in the HVAC unit.

There are a number of reasons for using VAV systems for indoor climate control. Hung et al [1] studied the performance of flow controllers in VAV systems. By simulations and field measurements, they found that the flow controllers were able to provide a stable zone air temperature. They also found that furniture and the zone interior surface stabilizes the zone air temperature dynamics. Inoue and Matsumoto [2] have made energy analyses of the VAV system and compared it with other systems such as dual duct constant air volume (CAV) and 2-pipe induction unit. With meteorological data from

Tokyo, the VAV system was found to have the lowest cooling coil load and lowest annual fan energy use.

Many VAV systems supply a constant air temperature and returns a part of the extracted air to the HVAC unit and then to the supply air system (return air). The reason for that is to decrease the power requirement and the energy use when the outdoor temperature is higher than the exhaust air temperature. VAV systems that use 100% outdoor air [3, 4] are installed in order to increase indoor air quality when there is a cooling need and to decrease energy use by only supplying air when needed. Most research has been made on the VAV system that uses return air. Most often, focus is on the proportions between return air and outdoor air [5] in order to make it work in practice and to reduce the energy use. In a 100% outdoor air VAV system there is no return air and therefore this is not a problem.

In many offices and premises in northern Europe, there is a need of cooling the building during a major part of the year, because the buildings have internal heat loads, are exposed to solar radiation and are insulated. The outdoor temperature is usually lower than the zone temperature and therefore returning the exhaust air will not reduce the energy use. Hittle [6] pointed out that most VAV systems do not include any heating function in the main air-handling unit. Hittle probably refered to the USA and in northern Europe, heating of the supply air is more common. With a 100% outdoor air system in the northern climates, heating of the supply air is a necessity. When the outdoor temperature is low, a heat recovery unit should be used to considerbly reduce the energy use.

By cooling the building structure during nighttime [7, 8], the energy use can be decreased. The supply air flow is increased during nighttime when the outdoor temperature is lower than the zone temperature. This is called night cooling.

Depending on the supply air temperature, the power used in the HVAC unit to produce the cooling or heating and to run the fan will differ. The total energy use will depend on the efficiencies of the components such as the specific fan power (SFP) value, temperature efficiency of the heat recovery unit and the chiller coefficient of performance (COP). Other factors affecting the energy use are the internal heat load, the temperature set points in the zones of the building, the outdoor air temperature and the average U-value of the building envelope. Decreasing the U-value by increasing the insulation without changing the solar gains will increase the need for cooling when the outdoor temperature is lower than the indoor temperature. A decreased U-value also decreases the need for heating at lower outdoor temperatures and decreases the cooling at higher outdoor temperatures.

Zaheer-Uddin and Zheng [9] have shown that there is an optimal supply air temperature in a climate where the relative humidity is high and return air is used. The study did not include heating of outdoor air. In a case study [10] they saved 20% of the energy use by an increase of outdoor air during specific conditions in a system that used return air. Most often, the relative humidity in northern Europe is below 70% when the outdoor temperature is higher than 20°C. In northern Europe, low relative humidity might be a problem during the winter period if a high outdoor air flow is supplied. Norford et al. [11] simulated with DOE-2 an office building in New Jersey. The energy use was calculated for different constant supply air temperatures and a supply air temperature decreasing with increased outdoor temperature. By changing the supply air temperature, the energy use was reduced by 10% in winter time and between 11% and 21% in summer time. Ke et al. [12] simulated eight ventilation control strategies in VAV systems and three of the strategies included a change in supply air temperature. The climate data was from south central Pennsylvania, USA. Their conclusion was that the supply air temperature and supply air flow rate were the two proper optimizable parameters on the air side of the HVAC system. Ke and Mumma [13] simulated the effect on ventilation when changing the supply air temperature in a fan powered VAV system (FPVAV) that uses return air. The climate data was from Harrisburg, PA, USA. Mathews et al. [14] showed other ways, such as air-bypass control on cooling coils and system start-stop times, to reduce the HVAC energy use.

There is a lack of general knowledge and theoretical approach regarding the influence from supply air temperature in VAV-systems using only outside air. The objectives of this paper are to show the theory of an optimal supply air temperature in regards to energy use and to analyze the energy savings potential when applying the optimal temperature to a 100% outside air VAV system in a northern European climate. The optimal U-value is also studied with regards to energy use depending on outdoor climate and internal heat loads.

The supply air temperature is usually controlled to be constant all the year round or to decrease when the outdoor temperature increases (called decreasing strategy). Energy calculations are made for 12, 14 and 16°C constant supply air temperature, one example of the decreasing strategy and the optimal supply air temperature. The energy use is divided into three parts, fan electrical energy, heating energy and cooling electrical energy. The heating energy is either used in the HVAC unit or in the zones of the building (the office cells) by radiators.

In order to find a practically applicable theory and a general comparison of the energy use, a number of assumptions and limitations are made. The energy use is calculated from hourly temperature and humidity data from two different climates in Sweden. The theory is based on steady state calculations. No night cooling is considered. Heat and electricity are treated as equal and no economical aspects are taken. For example, heat and electricity could in practice have different costs or environmental impacts.

The result shows that there is a major potential in controlling the supply air temperature optimally to reduce the HVAC energy use if the internal heat loads are of importance. Only when a building is used 24-hours a day and the internal heat load is over 15 W/m^2 floor area, the optimal U-value is higher than zero. When the building is only used at day time (06-18), the internal loads have to be higher than 133 W/m^2 floor area.

Nomenclature

a	Constant in regression analysis ($g_{H2O}/(m^3 \cdot °C)$)
A_{facade}	Facade area (m^2)
A_{floor}	Floor area (m^2)
b	Constant in regression analysis (g_{H2O}/m^3)
COP	Coefficient of performance (-)
c_p	Specific heat at constant pressure (J/(kg·ºC))
h	Condensation enthalpy, 2300 (J/g_{H2O})
i	Exponent in relationship between fan power and air flow (-)
k_1	Constant (°C^2)
k_2	Constant (W)
k_3	Constant (W·°C^2)
k_4	Constant (W)
k_5	Constant (°C^2)
k_6	Constant (°C^3)
n	Number of zones (-)
P_{boil}	Power input to boiler (W)
P_{CM}	Power input to chiller (W)
P_{cond}	Power requirement to condense supply air in chiller (latent heat) (W)
$P_{cooling}$	Supplied cooling power to zone (W)
P_{fan}	Power input to fan (W)
P_{HR}	Power saved by heat recovery (W)
P_{HVAC}	Power input to HVAC unit (W)
P_{HVAC_1}	Power input to HVAC unit in case 1 ($t_{sat}<t_{SA}<t_{out}$) (W)
P_{HVAC_2}	Power input to HVAC unit in case 2 ($t_{fan}=t_{SA}$) (W))
P_{HVAC_3}	Power input to HVAC unit in case 3 ($t_{sat}<t_{SA}<t_{out}$) (W)
P_{HVAC_4}	Power input to HVAC unit in case 4 ($t_{SA}<t_{sat}$) (W)
$P_{H\&C}$	Power input to HVAC unit when zones are in different modes (heating and cooling)
$P_{internal}$	Internal heat load in zone (W)
P_{load}	Total load in zone (W)
P_{load_j}	Total load in zone j (W)
$P_{load_{total}}$	Sum of total load in all zones (W)
$P_{load_{cool}}$	Sum of total load in zones with $q>q_{min}$ (W)

P_{rad}	Power input to radiators in zone (W)
P_{sun}	Solar heat gains (W)
q	Supply air flow (m^3/s)
q_{min}	Minimum supply air flow (m^3/s)
q_{SFP}	Air flow where SFP value is determined for one zone (m^3/s)
$q_{SFPtotal}$	Air flow where SFP value is determined the whole system (m^3/s)
RH	Outdoor relative humidity (-)
SFP	Specific fan power ($W/(m^3/s)$)
t_{EX}	Exhaust air temperature (°C)
t_{fan}	Air temperature after fan (°C)
t_{HC}	Air temperature after heating coil (°C)
t_{HR}	Air temperature after heat recovery (°C)
t_{out}	Outdoor air temperature (°C)
t_{zone}	Zone temperature (°C)
t_{SA}	Supply air temperature (°C)
t_{SAhigh}	Upper limit for supply air temperature (°C)
t_{SAlow}	Lower limit for supply air temperature (°C)
t_{sat}	Saturation temperature (°C)
U	U-value ($W/(m^2 \cdot °C)$)
v_{sat}	Moisture content in outdoor air (g_{H2O}/m^3)
v_{satSA}	Moisture content in supply air (g_{H2O}/m^3)
W	Total annual energy use per floor area ($W_{heat}+W_{fan}+W_{CM}$) ($kWh/(m^2 \cdot year)$)
W_{heat}	Annual heat energy use per floor area including the HVAC unit and the radiator system ($kWh/(m^2 \cdot year)$)
W_{fan}	Annual fan energy use per floor area ($kWh/(m^2 \cdot year)$)
W_{CM}	Annual chiller energy use per floor area ($kWh/(m^2 \cdot year)$)

Greek letters

Δv	Difference in moisture content (g_{H2O}/m^3)
ρ	Air density (kg/m^3)
η_b	Efficiency of boiler (-)
η_t	Temperature efficiency of heat recovery unit (-)

2. Method

This chapter firstly describes the theory for calculating the power requirement of the different parts of the HVAC unit and the zone in Fig. 1. In this first part, the building is considered as one zone with one temperature set point and one air flow. Then, theory is divided into four different operating cases depending on outdoor conditions and what parts of the HVAC unit that are operating (using energy). The supply air temperature is optimized in each case regarding power requirement. After that, a multi zone approach is presented. Finally, climate data used in the energy calculations and other control strategies for the supply air temperature are presented.

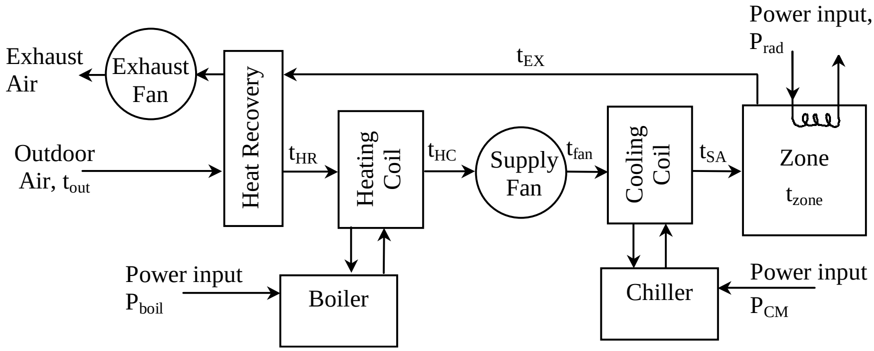

Fig. 1. Schematic figure of HVAC unit and system.

2.1 Assumptions and limitations

- It is assumed that there is no thermal storage nor in the zone or in the walls surrounding the zone. When the outdoor temperature changes, this results in an overestimation in changes in cooling or heating need in the zone. This will not affect the equations for optimization but in reality with thermal storage, the absolute energy use would be different. The internal walls are supposed to have infinite insulation. Otherwise, there would be heat exchange between the zones. The equations will still be valid though the energy exchange can be treated as a change in internal load. The same argument is valid for a heavy weight building with thermal storage when for example night cooling is being used. Steady state conditions are assumed.
- In the energy use calculations, there have been no limits for the maximum air flow. If an upper limit were used, this would result in a difference in zone comfort depending on the control strategy. In addition, it would not be possible to compare the control strategies but the power use would decrease.
- The infiltration is assumed to be zero. If infiltration were included, this would affect the cooling or heating need in the zone. Therefore, the calculated annual energy use for the described control strategies would be affected in the same direction.
- The ducts are assumed to be tight. If there were air leaking from the ducts, this would result in a higher SFP-value than the value given by the fan manufacturer. The correct SFP value can be measured when the system has been taken into operation.
- It is assumed that the exhausted air temperature is equal to the zone

temperature. In reality there is a temperature gradient in the zone resulting in an exhaust air temperature higher than the zone set point temperature. Then, the recovered heat energy is underestimated in the energy use calculations.

- The heat recovery unit is only used to heat the outdoor air and not to cool the outdoor air when the exhaust air temperature is lower then the outdoor temperature.
- In practise, the supply fan is often located after the pre-conditioning units. Here, it is located before the cooling coil to simplify the calculations. If the supply fan were located after the cooling coil, the temperature after the cooling coil had to be lower due to the heating from the supply fan. In some cases, this would lead to increased energy use for condensation.
- In the energy calculations no solar radiation is included (P_{sun} =0). If solar radiation were present, the absolute calculated HVAC-energy use would be affected. In the comparison between the control strategies, the affection would be in the same direction and therefore the difference would be small. Solar radiation will not affect the supply air temperature optimization because from the system perspective there is no difference in solar gain or internal heat load. Therefore, solar radiation can be treated as a part of the internal heat load.
- The efficiency of the boiler, η_b and the radiator system is set to be 1.
- The temperature efficiency, η_t of the heat recovery unit is assumed to be constant. The supply air flow and the exhaust air flow are equal.
- The coefficient of performance (COP) of the chiller is assumed to be constant. The specific heat, c_p, (1000 J/(kg·°C)), of air and the air density, ρ, (1.2 kg/m^3), are assumed to be constant. The density affects the fan power which would, in this case, vary approximately ±1% if the density was treated as temperature dependent.
- To simplify the model, the water pump energy used in the boiler and the heat recovery unit is assumed to be zero. Fan electricity, cooling electricity and heating energy are treated as equal.

2.2 Supply air temperature

The supply air temperature, t_{SA}, is limited by an upper temperature, t_{SAhigh}, due to mixing ventilation and ventilation effectiveness, and a lower temperature, t_{SAlow}, due to thermal comfort.

$$t_{SAlow} \le t_{SA} \le t_{SAhigh} \qquad\qquad t_{SAhigh} < t_{zone}$$

2.3 Heat balance of the zone

Equation (1) describes the total load, P_{load}, in a single zone that has to be cooled or heated by the system. Heating when the load is negative and cooling when the load is positive.

$$P_{load} = P_{internal} + P_{sun} - U \cdot A \cdot (t_{zone} - t_{out}) \qquad [W] \qquad (1)$$

Equation (2) describes the cooling power, $P_{cooling}$, provided by the supply air. The cooling power must be positive, meaning the supply air temperature, t_{SA}, must be lower than the zone temperature, t_{zone}.

$$P_{cooling} = q \cdot \rho \cdot c_p \cdot (t_{zone} - t_{SA}) \qquad [W] \qquad (2)$$

Equation (3) is valid when the zone temperature set point and steady state condition are reached.

$$P_{load} = P_{cooling} \qquad [W] \qquad (3)$$

2.4 HVAC unit

To meet the load in the zone, the HVAC unit must produce an air flow at a certain temperature. The radiator power, P_{rad}, is included in the HVAC unit power. The power, P_{HVAC}, used to produce this is described in Eq. (4).

$$P_{HVAC} = P_{CM} + P_{fan} + P_{boil} + P_{rad} \qquad [W] \qquad (4)$$

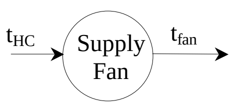

Fig. 2. Supply air fan.

The theoretical relationship between fan power, P_{fan}, and air flow, q, is cubic (i=3). However, in practice there are losses in the frequency converter and motor, and the fan efficiency is not constant. Therefore, a squared approach (i=2) is more appropriate.

$$P_{fan} = q^i \cdot \frac{SFP}{q_{SFP}^{i-1}} \qquad [W] \qquad (5)$$

Equation (6) describes the air temperature after the supply fan. P_{fan} is the sum of supply and exhaust fan power and it is assumed that the supply fan electricity ($1/2 \cdot P_{fan}$) converts into a rise in temperature of the supply air.

$$t_{fan} = t_{HC} + \frac{P_{fan}}{2 \cdot q \cdot \rho \cdot c_p} \qquad [°C] \qquad (6)$$

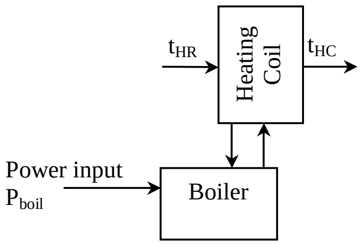

Fig. 3. Boiler and heating coil.

The boiler and the heating coil are used to increase the supply air temperature after the heat recovery unit. As far as the supply air temperature is higher than the maximum air temperature after heat recovery, it does not matter in energy perspective whether radiator or boiler is used to heat the zone (η_b=1).

$$P_{boil} = \frac{q \cdot \rho \cdot c_p \cdot (t_{HC} - t_{HR})}{\eta_b} \qquad [W] \qquad (7)$$

The power saved by the heat recovery unit, P_{HR}, is described in Eq. (8).

$$P_{HR} = q \cdot \rho \cdot c_p \cdot (t_{HR} - t_{out}) \qquad [W] \qquad (8)$$

The temperature efficiency of the heat recovery unit, η_t, is expressed in Eq. (9). It is assumed that it is not air flow dependent.

$$\eta_t = \frac{t_{HR} - t_{out}}{t_{EX} - t_{out}} \qquad [-] \qquad (9)$$

Perfectly mixed air is assumed in the zone, that results in Eq. (10).

$$t_{EX} = t_{zone} \qquad [°C] \qquad (10)$$

If the calculated supply air temperature is higher than the highest supply air temperature, then Eq. (11) is used to calculate the radiator power input (P_{rad}>0).

$$P_{rad} = q_{min} \cdot \rho \cdot c_p \cdot \left(t_{zone} - t_{SAhigh}\right) - P_{load} \qquad [W] \qquad (11)$$

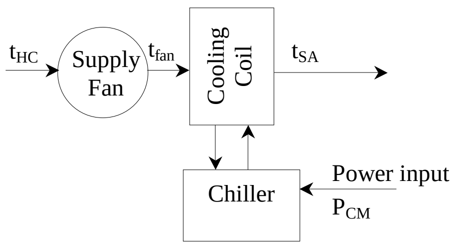

Fig. 4. Chiller and cooling coil.

The power input to the chiller, P_{CM}, is described in Eq. (12).

$$P_{CM} = q \cdot \rho \cdot c_p \cdot \frac{\left(t_{fan} - t_{SA}\right)}{COP} + P_{cond}\left(q, t_{sat}, t_{SA}\right) \qquad [W] \qquad (12)$$

When the supply air is cooled below the dew point temperature, there will be an extra energy loss in the chiller for condensation. Here, it is assumed that the condensed mass is equal to the outdoor moisture content minus the saturation moisture content at the supply air temperature. The power used to decrease the temperature of the condensed water is relatively small and therefore neglected.

To find a manageable expression for the condensed power, a linear regression analysis of the moisture content dependent on the saturation temperature is done. The analyzed range was between 12 and 26°C. The regression analysis results in Eq. (13).

$$v_{sat} = a \cdot t_{sat} + b \qquad [g_{H2O}/m^3] \qquad (13)$$

The constant, a is 0.972 $g_{H2O}/(m^3 \cdot °C)$ and the correlation coefficient is 0.995. The condensed mass when decreasing the temperature from t_{sat} to t_{SA} is expressed in Eq. (14) where b is reduced.

$$\Delta v = v_{sat} - v_{satSA} \qquad [g_{H2O}/m^3] \qquad (14)$$

The cooling power input caused by condensation is expressed in Eq. (15).

$$P_{cond} = \frac{q \cdot h \cdot \Delta v}{COP} = \frac{q \cdot h \cdot a \cdot (t_{sat} - t_{SA})}{COP} \quad \text{[W]} \quad (15)$$

Based on the equations given, the operation of the HVAC unit can be devided into four different cases.

2.5 Case 1 ($t_{HC} > t_{out}$)

In this case, the outdoor air is heated first by the heat recovery unit and, if needed, also with the boiler. To minimize the power input in this sequence, the supply air temperature should be set at a temperature where the cooling need can be satisfied with the minimum air flow, q_{min}. If the calculated temperature is lower than t_{SAlow}, then minimum air flow can not be used and t_{SAlow} should be used as supply air temperature.

The heat recovery unit is controlled in a way that t_{HR} reaches t_{SA} when possible. If t_{HR} is lower than t_{SA} then the boiler is used. If the zone is in heating mode ($P_{load} < 0$), the supply air temperature should be as high as possible (t_{SAhigh}).

When the HVAC unit is used to heat the outdoor air to reach the supply air temperature ($P_{CM} = 0$), then Eq. (4) is reduced to Eq. (16).

$$P_{HVAC_1} = P_{fan} + P_{boil} + P_{rad} \quad \text{[W]} \quad (16)$$

If the zone needs cooling (P_{load}>0), then P_{rad} equals 0 and

$$q = q_{\min} \quad \text{[m}^3\text{/s]} \quad (17)$$

Using Eqs. (2), (3) and (17) results in Eq. (18). This equation is used to determine what supply air temperature should be used if minimum air flow is supplied.

$$t_{SA} = t_{zone} - \frac{P_{load}}{q_{\min} \cdot \rho \cdot c_p} \quad [°\text{C}] \quad (18)$$

If t_{SA} according to Eq. 18 is higher than t_{HR}, and t_{HR} is higher than or equal to t_{SAlow}, then it is possible to supply t_{SA}=t_{HR} and heat the air in the zone with the radiator system. This will not affect the HVAC energy use.

2.6 Case 2 (t_{SA}=t_{fan})

In this case the outdoor air is neither heated nor cooled by the HVAC unit (free cooling). This results in Eq. (19). To satisfy the cooling need, the air flow is the parameter that is changing and therefore the only active part of the HVAC unit is the fan.

$$t_{SA} = t_{fan} \qquad [°C] \qquad (19)$$

Equation (4) is then reduced to Eq. (20).

$$P_{HVAC_2} = P_{fan} \qquad [W] \qquad (20)$$

Using Eq. (20) with Eqs. (2), (3), (5) and (6) result in Eq. (21).

$$t_{fan} = \frac{t_{out} + t_{zone}}{2} - \frac{1}{2} \cdot \sqrt{(t_{out} - t_{zone})^2 - k_1} \qquad [°C] \qquad (21)$$

Where:

$$k_1 = \frac{2 \cdot P_{load} \cdot SFP}{c_P^2 \cdot q_{SFP} \cdot \rho^2} \qquad [°C^2] \qquad (22)$$

Equation (21) is not valid for $(t_{out} - t_{zone})^2 - k_1 < 0$, that is when the temperature after the fan is higher then the zone temperature and free cooling is not possible.

2.7 Case 3 ($t_{sat} < t_{SA} < t_{fan}$)

In this case, the outdoor air is cooled to reach the supply air temperature but not below the saturation temperature, t_{sat}, of the supply air. As no heating of the air is needed, Eq. (4) is reduced to Eq. (23).

$$P_{HVAC_3} = P_{CM} + P_{fan} \qquad [W] \qquad (23)$$

In this case, there is no condensation (t_{SA}>t_{sat}), which results in Eq. (24).

$$P_{cond} = 0 \qquad [W] \qquad (24)$$

Solving q from Eq. (2) and then using Eqs. (3)-(6), (12), (23) and (24) results in Eq. (25).

$$P_{HVAC_3} = k_2 \cdot \frac{t_{out} - t_{SA}}{t_{zone} - t_{SA}} + k_3 \cdot \frac{1}{(t_{zone} - t_{SA})^2} \qquad \text{[W]} \qquad (25)$$

Where:

$$k_2 = \frac{P_{load}}{COP} \qquad \text{[W]} \qquad (26)$$

$$k_3 = \left(1 + \frac{1}{2 \cdot COP}\right) \cdot \frac{P_{load}{}^2 \cdot SFP}{\rho^2 \cdot c_p^2 \cdot q_{SFP}} \qquad [\mathrm{W \cdot {}^\circ C^2}] \qquad (27)$$

Equation (25) is only valid for the case when $t_{SA}<t_{zone}$, otherwise there would be no cooling of the zone.

To find the optimal supply air temperature the HVAC power, P_{HVAC}, in Eq. (25) is derived with respect to the supply air temperature.

$$\frac{dP_{HVAC_3}}{dt_{SA}} = -k_2 \cdot \frac{t_{zone} - t_{out}}{(t_{zone} - t_{SA})^2} + k_3 \cdot \frac{2}{(t_{zone} - t_{SA})^3} \qquad [\mathrm{W/{}^\circ C}] \qquad (28)$$

Optimization occurs when Eq. (28) equals zero.

$$\frac{dP_{HVAC_3}}{dt_{SA}} = 0 \qquad [\mathrm{W/{}^\circ C}] \qquad (29)$$

Equations (28) and (29) result in Eq. (30), which is the optimal supply air temperature when the chiller is running and there is no condensation.

$$t_{SA} = t_{zone} - \frac{k_3}{k_2} \cdot \frac{2}{t_{zone} - t_{out}} \qquad [\mathrm{{}^\circ C}] \qquad (30)$$

Equation (30) is valid for the case when $t_{out}<t_{zone}$. When $t_{out}>t_{zone}$, Eq. (30) results in a t_{SA} greater than t_{zone} which is out of the range for Eq. (2). When studying Eq. (25), the factor multiplied with k_2 is continuously decreasing with decreasing t_{SA} for the case when $t_{out}>t_{zone}$. Therefore t_{SA} should be as low as possible, that is t_{SAlow}. The breakpoint when free cooling (Case 2) should not be used is when the result of Eq. (30) is higher than the temperature after the fan

(t_{SA}>t_{fan}). The second derivative of Eq. (25) with Eq. (30) as t_{SA} is positive which indicates a minimum.

2.8 Case 4 (t_{SA}<t_{sat})

This case is equal to Case 3 except that the air is now cooled below the dew point temperature (t_{sat}) of the supply air. Using Eqs. (2)-(6), (15) and (23) result in Eq. (31).

$$P_{HVAC_4} = k_2 \cdot \frac{t_{out} - t_{SA}}{t_{zone} - t_{SA}} + k_3 \cdot \frac{1}{(t_{zone} - t_{SA})^2} + k_4 \cdot \frac{t_{sat} - t_{SA}}{t_{zone} - t_{SA}} \quad \text{[W]} \quad (31)$$

Where:

$$k_4 = \frac{P_{load} \cdot a \cdot h}{\rho \cdot c_p \cdot COP} \quad \text{[W]} \quad (32)$$

To find the optimal supply air temperature, the HVAC power in Eq. (31) is derived with respect to the supply air temperature.

$$\frac{dP_{HVAC_4}}{dt_{SA}} = -k_2 \cdot \frac{t_{zone} - t_{out}}{(t_{zone} - t_{SA})^2} + k_3 \cdot \frac{2}{(t_{zone} - t_{SA})^3} + k_4 \cdot \frac{t_{sat} - t_{zone}}{(t_{zone} - t_{SA})^2} \quad \text{[W/°C]} \quad (33)$$

Optimum occurs when Eq. (33) equals zero.

$$\frac{dP_{HVAC_4}}{dt_{SA}} = 0 \quad \text{[W/°C]} \quad (34)$$

Equations (33) and (34) result in Eq. (35), that describes the optimal, supply air temperature in this case.

$$t_{SA} = \frac{k_2 \cdot t_{zone} \cdot (t_{out} - t_{zone}) + 2 \cdot k_3 + k_4 \cdot t_{zone} \cdot (t_{sat} - t_{zone})}{k_2 \cdot (t_{out} - t_{zone}) + k_4 \cdot (t_{sat} - t_{zone})} \quad \text{[°C]} \quad (35)$$

The second derivative of Eq. (31) with Eq. (35) as t_{SA} is positive which indicates a minimum.

2.9 Multi zone model

When more than one zone is presented, combinations of the described cases can occur. If the heat balance for all zones can be provided without the flow reaching either minimum or maximum, the optimal supply air temperature, t_{SA}, can be calculated from the single zone equations. Then, a total load has to be used as well as a corresponding air flow, $q_{SFP_{total}}$, valid for all zones together. The temperature set points, t_{zone}, must be equal in all zones.

$$P_{load_{total}} = \sum_{j=1}^{n} P_{load_j} \qquad \text{[W]} \qquad (36)$$

If n number of zones are in heating mode, that is when the calculated flow is lower than q_{min}, and the rest are in cooling mode with just free cooling available, the case might occur when the heat recovery unit is not fully used due to the zones in cooling mode. If the supply air temperature is lower than the maximum temperature after the heat recovery, there will be an extra heating power to heat up the air in those zones that are in heating mode. The equation is valid for $t_{SA}>t_{out}$ and $t_{SA}<t_{HRmax}$.

$$P_{H\&C} = \left(n \cdot q_{min} + \frac{P_{load_{cool}}}{(t_{zone} - t_{SA}) \cdot \rho \cdot c_p} \right)^2 \cdot \frac{SFP}{q_{SFP_{total}}} + \qquad \text{[W]} \qquad (37)$$
$$+ n \cdot q_{min} \cdot \rho \cdot c_p \cdot \left(t_{HR_{max}} - t_{SA} \right)$$

To find the optimal supply air temperature, the HVAC power in Eq. (37) is derived with respect to the supply air temperature. Then t_{SA} is solved resulting in Eq. (38).

$$t_{SA} = t_{zone} - \sqrt[3]{\frac{-k_6}{2} + \sqrt{\left(\frac{k_5}{3}\right)^3 - \left(\frac{k_6}{2}\right)^2}} + \sqrt[3]{\frac{-k_6}{2} - \sqrt{\left(\frac{k_5}{3}\right)^3 - \left(\frac{k_6}{2}\right)^2}} \qquad [^{\circ}\text{C}] \qquad (38)$$

Where:

$$k_5 = \frac{-2 \cdot P_{load} \cdot SFP}{\rho^2 \cdot c_p^2 \cdot q_{SFP}} \qquad [^{\circ}\text{C}^2] \qquad (39)$$

$$k_6 = \frac{-2 \cdot P_{load}^2 \cdot SFP}{n \cdot q_{\min} \cdot \rho^3 \cdot c_p^3} \qquad [^{\circ}\text{C}^3] \qquad (40)$$

If Eq. (38) results in $t_{SA}>t_{HRmax}$, the equation is not valid. Then the supply air temperature should be t_{HRmax} to decrease the risk of increased air flow to zones in cooling mode. The risk of having one zone in heating mode and one zone in cooling mode is low when the outdoor temperature is so high that the chiller has to be used.

2.10 Non optimal control strategies

To be able to compare the energy use when an optimal supply air temperature is used, the annual energy use is also calculated for other existing strategies. Two other stategies are considered, constant supply air temperature and decreasing supply air temperature according to Fig. 5.

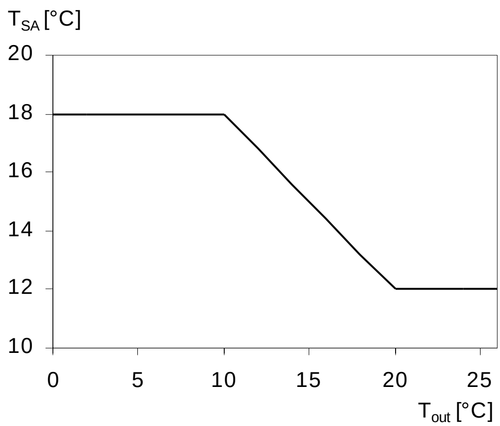

Fig. 5. The supply air temperature, t_{SA}, as a function of the outdoor temperature, t_{out}, with the control strategy called "decreasing" and the temperature coordinates (10, 18) and (20, 12).

Figure 5 shows the supply air temperature that was used in the one-year energy use calculations for Luleå and Sturup. The supply air temperature depends only on the outdoor air temperature, t_{out}. The temperature coordinates (10, 18) and (20, 12) in Fig. 5 are an example of the decreasing strategy and these coordinates were used in the energy calculations. For the other control strategy, constant supply air temperature, calculations were made for 12, 14 and 16°C respectively.

2.11 Climates

The climate data [15] used for one-year energy use calculations are one-year measurements in Luleå (north of Sweden, latitude: 65°33’N, longitude: 22°08’E) and Sturup (south of Sweden, latitude: 55°33’N, longitude: 13°22’E) for the year 1977. The data are hourly average values of the outdoor temperature and relative humidity. The year 1977 was the most representative year for the period 1973-1990 in Sturup according to the average temperature. The average temperature in Sturup 1977 was 7.1°C and in Luleå 1.2°C (1.5°C in average 1961-1990). Figures 6 and 8 show the temperature frequency for the whole year and for the daytime (06.00-18.00) of the year for Sturup and Luleå respectively. Figures 7 and 9 show the outdoor temperature at different relative humidity for Sturup and Luleå respectively. Each dot corresponds to the temperature and relative humidity for one hour. When the outdoor temperature was above 20°C in 1977, the relative humidity was above 70% during 32 hours in Sturup and 2 hours in Luleå.

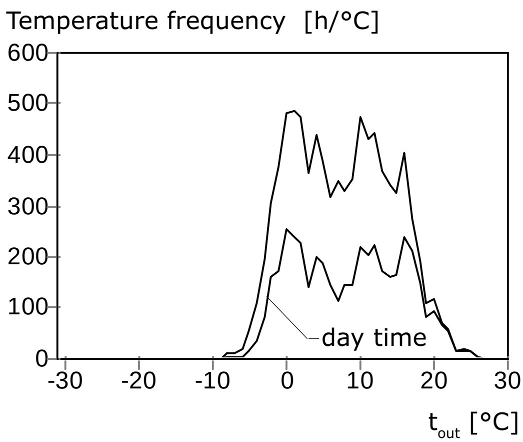

Fig. 6. One year temperature frequency for Sturup (1977) with and without the night hours (18.00-06.00).

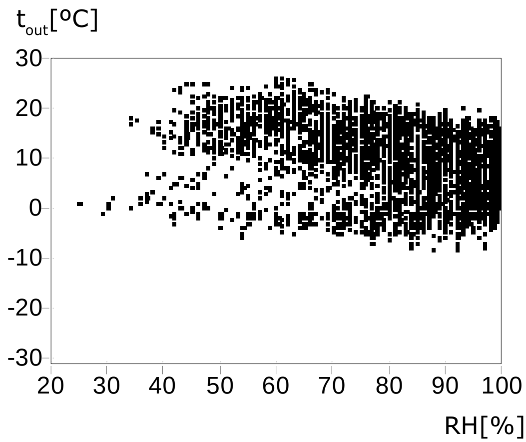

Fig. 7. Outdoor relative humidity and outdoor temperature plotted for each hour for Sturup 1977.

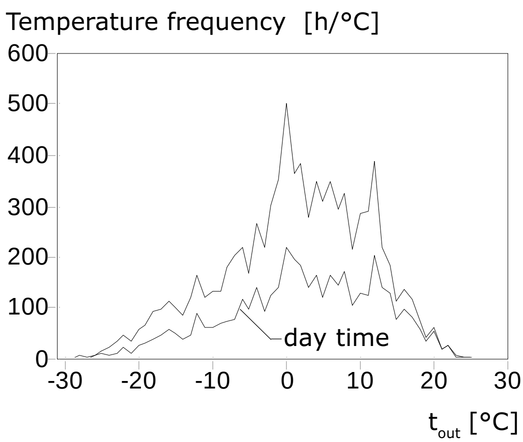

Fig. 8. One year temperature frequency for Luleå 1977 with and without the night hours (18.00-06.00).

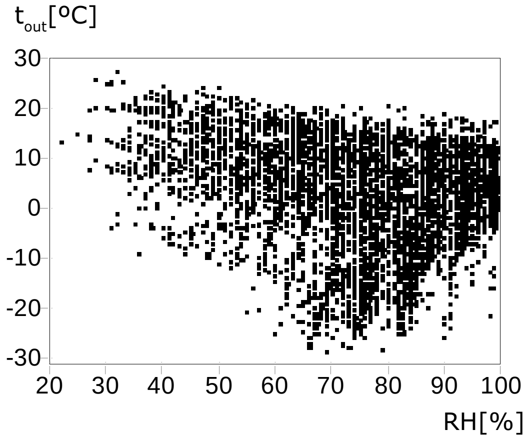

Fig. 9. Relative humidity and outdoor temperature plotted for each hour for Luleå 1977.

3. Results

Calculations were made to show examples of the optimal supply air temperature with respect to the HVAC energy use. The HVAC power depending on supply air temperature was calculated for the cases defined in the methods section. Calculations were also made to show the energy savings potential and the practical influence of an optimized supply air temperature.

3.1 Data used in the calculations

The input data used are typical for an HVAC system and one zone in an office building. Although, in many of the examples where the optimal supply air temperature is calculated, the input data are modified to be able to show different cases of events. The following data were used in the calculations if nothing else is declared.

SFP=2000 W/(m^3/s), COP=3, t_{zone} = 23°C, U=1.5 W/(m$^{2}\cdot$°C), A_{facade}=9 m^2, A_{floor}=13.5 m^2, η_t=0.6, RH=60 %, q_{SFP}=0.05 m^3/s, q_{min}=10 l/s, t_{SAhigh}=18°C , t_{SAlow}=12°C, P_{sun}=0.

3.2 Optimal supply air temperature

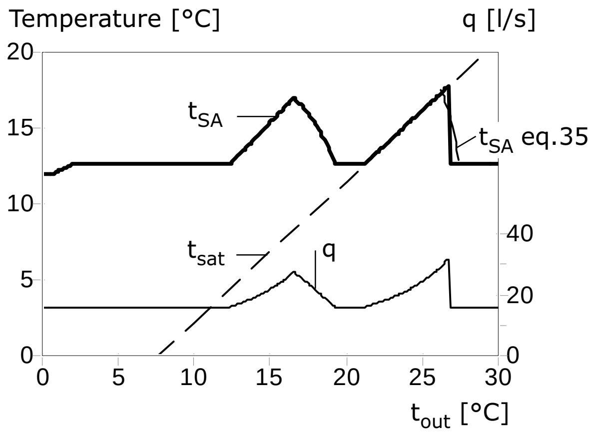

Fig. 10. Optimal supply air temperature (t_{SA}) and supply air flow, q, at different outdoor temperatures (t_{out}) and with a constant load, P_{load}=$P_{internal}$= 200 W (U=0) corresponding to 15 W/m^2 floor area. η_t =0.5, q_{min}=16 l/s. The dashed line is the saturation temperature (t_{sat}).

The equations described in Section 2.2-2.8 with a constant load (P_{load}= 200 W) result in the optimal supply air temperatures shown in Fig. 10. To show the variation in supply air temperature independent of a load caused by the outdoor temperature, there is no energy flow through the facade (U=0). The temperature efficiency, η_t, was 0.5 to show the breaking point where the heat recovery does not handle the heating that is needed. Below 2ºC, the heat recovery unit can not provide 12.6ºC. At this outdoor temperature, the heating is prefered to come from the radiators according to Case 1 in the method section. This choice is made to decrease the risk of having zones not receiving sufficient cooling power with a minimum air flow in a multi zone case. When the supply air temperature is lower than the saturation temperature, there are two lines describing the optimal supply air temperature in Figs. 10 and 11. The thin line is when Eq. (35) (linear moister content) is used to calculate the condensation energy.

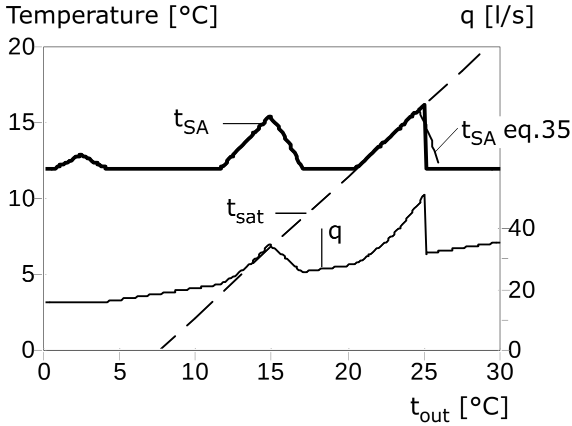

Fig. 11. Optimal supply air temperature (t_{SA}) and supply air flow, q, at different outdoor temperatures (t_{out}). The load, P_{load} increases with an increasing outdoor temperature ($P_{internal}$=400 W corresponding to 30 W/m^2 floor area, U=1,67 W/(m^2·°C)). η_t =0.5, q_{min} =16 l/s. The dashed line is the saturation temperature (t_{sat}).

The outdoor temperature regions for the different cases can be identified in Fig. 11 where the outdoor temperature influences the load. Also here, the η_t is 0.5. Case 1, when the outdoor air is heated in the HVAC unit, is between zero and twelve degrees outdoor temperature. Case 2, when the outdoor air is neither heated nor cooled in the HVAC unit is between 12 and 15.2°C (called free cooling). Case 3, when the outdoor air is cooled without reaching condensation is between 15.2 and 24.8°C. Case 4, when the outdoor air is cooled below saturation temperature is between 24.8 and 30°C.

3.3 The HVAC power use depending on supply air temperature

The HVAC power, P_{HVAC}, is shown in Figs. 12-16 for the different cases and for different supply air temperature set points. This is to illustrate the sensitivity of P_{HVAC}, depending on the supply air temperature. The input data were the same as in Fig. 11 to make it possible to compare Figs. 12-15 with Fig. 11.

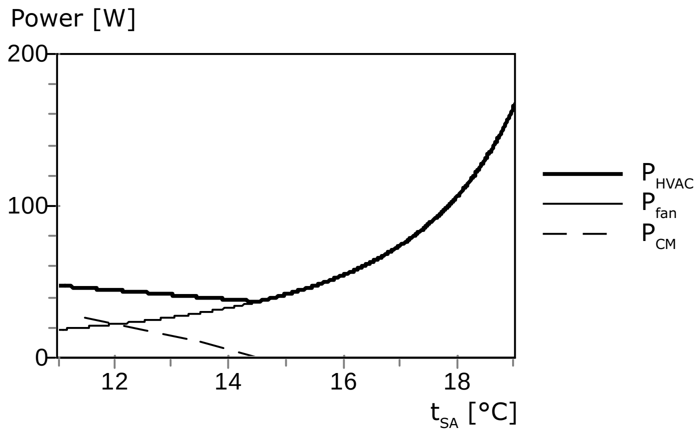

Fig. 12. The power requirement (P_{HVAC}) depending on the supply air temperature in Case 3 at 14°C outdoor temperature. Input data according to Fig. 11.

A minimum power requirement can be identified in Fig. 12 at a supply air temperature of 14.4°C. The minimum occurs in Case 2 when only free cooling is used. This can also be identified in Fig. 11 at 14°C outdoor temperature.

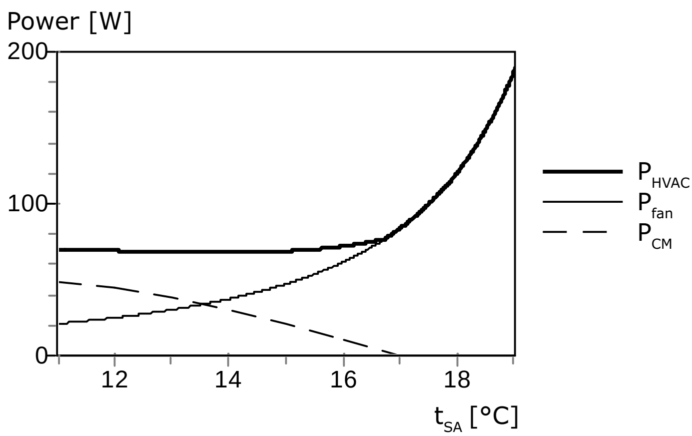

Fig. 13. The power requirement (P_{HVAC}) depending on supply air temperature in Case 3 at 16°C outdoor temperature. Input data according to Fig. 11.

A minimum power requirement can be identified in Fig. 13 for Case 3 at a supply air temperature of 13°C. In this case the outdoor temperature is 16°C and at this minimum both free cooling and the chiller are being used.

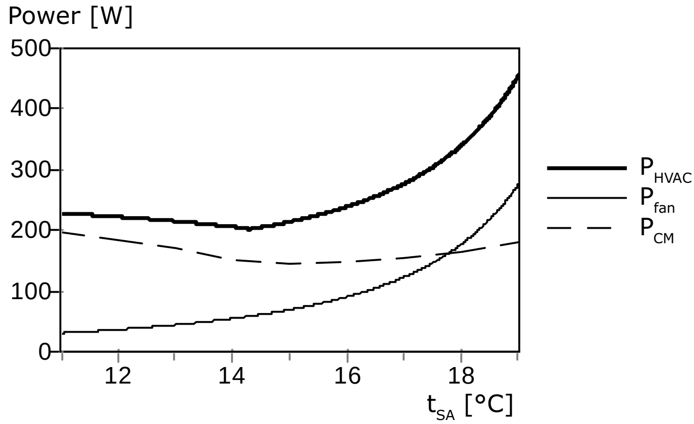

Fig. 14. The power requirement (P_{HVAC}) depending on supply air temperature in Case 3 at 23°C outdoor temperature. Input data according to Fig. 11.

A minimum power requirement can be identified in Fig. 14 for Case 3. In this case the outdoor temperature is 23°C and the minimum is just before condensation.

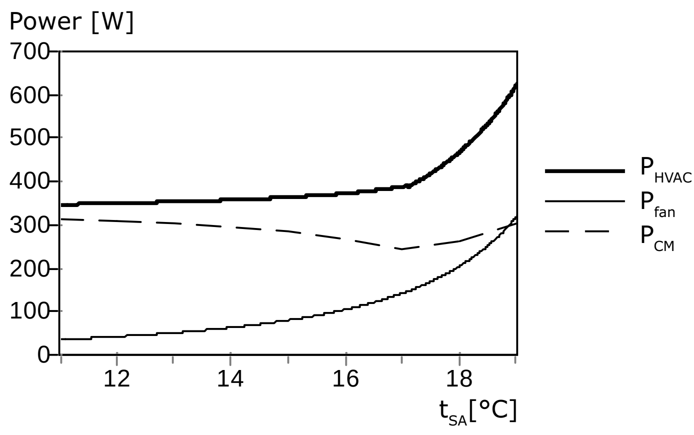

Fig. 15. The power requirement (P_{HVAC}) depending on supply air temperature in Case 4 at 26°C outdoor temperature. Input data according to Fig. 11.

A minimum power requirement can be identified in Fig. 15 for Case 4, which is when the supply air temperature is as low as possible. In this case the outdoor temperature is 26°C and the minimum is below the saturation temperature.

3.4 Optimal supply air temperature depending on fan flow exponent (i)

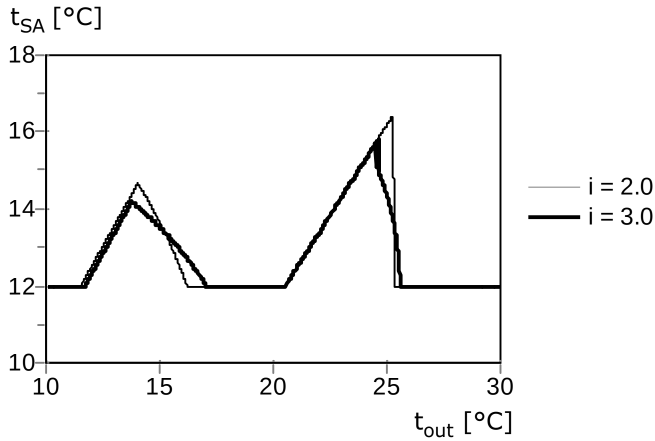

Fig. 16. Optimal supply air temperature with different exponents (i) in Eq. (5). $P_{internal}$=400 W (corresponding to 30 W/m^2 floor area) and U=0.

In the equations it is assumed that the exponent, i, in Eq. (5) is 2. Figure 16 shows what the optimal temperature would be if the exponent differs from 2. The difference in supply air temperature is relatively small.

3.5 Different temperature set point in different zones

All equations in the method section are based on a uniform zone temperature, t_{zone}. In reality, this is not the case. Different zones have diffrent zone temperatures. The power, P_{HVAC}, calculated from the equations, using the mixed exhaust air temperature from several zones as input for t_{zone}, was compared to the lowest possible power calculated iteratively (lower curve). The result is shown in Fig. 17. The calculations were made for a system with two zones. The temperature in zone 1 was 20°C and 25°C in zone 2 and the internal loads were 23 W/m^2 floor area in each zone.

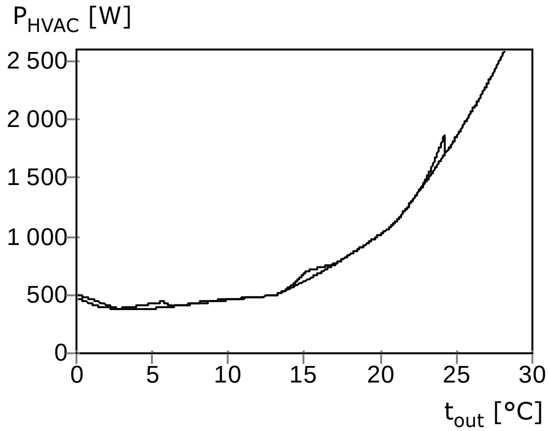

Fig. 17. Power requirement depending on outdoor temperature for two zones with different temperature set points. Upper curve when equations for single zone are used with the mixed exhaust air temperature as t_{zone} in the equations. Lower curve is when the sum of the power for the two zones is optimized iteratively.

3.6 Energy use depending on supply air temperature in two different climates

Decreasing the supply air temperature when the outdoor air temperature is increasing, as described in Fig. 5, is a control strategy that is commonly used in Sweden. Another way of controlling the supply air temperature is to keep the temperature constant all the year round. Figures 18-21 show the daytime, 06.00-18.00, energy use for one year (1977) in Sturup and Luleå at different internal loads and supply air temperature control strategies (described in 2.10). The energy use was calculated for one zone with a floor area (A_{floor}) of 13.5 m^2 and a facade area (A_{facade}) of 9 m^2 with a U-value of 1.5 W/(m^2·ºC). The energy to create the internal load is not included in the annual energy use.

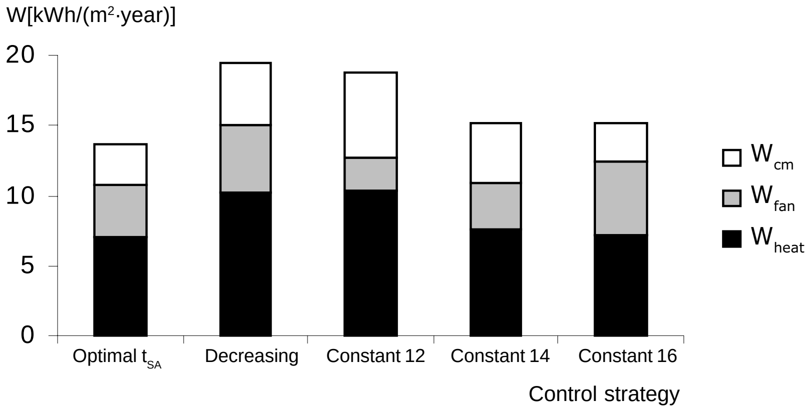

Fig. 18. Day time energy use per m^2 floor area for one zone with a facade area of 9 m^2 and floor area of 13.5 m^2 during one year in Sturup with an internal load of 350 W (corresponding to 26 W/m^2 floor area).

The smallest difference in energy use between the optimal t_{SA} and the other control strategies in Fig. 18 is 11%, and that is compared to a constant supply air temperature at 14°C (Constant 14).

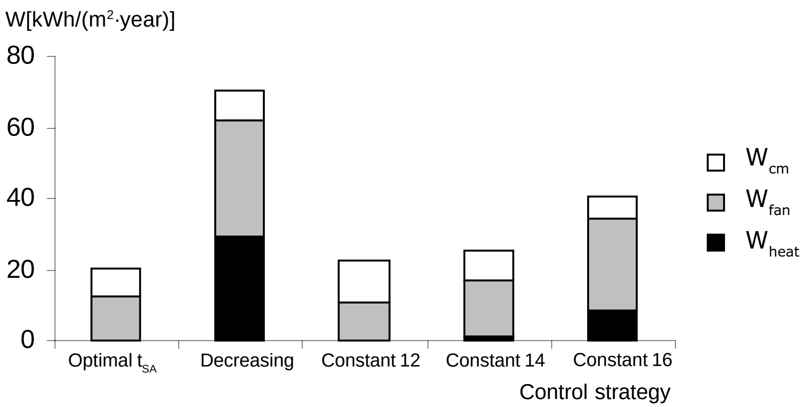

Fig. 19. Day time energy use for one zone with a facade area of 9 m^2 during one year in Sturup with an internal load of 600 W (corresponding to 44 W/m^2 floor area).

If the supply air is controlled optimally in Sturup and the internal load is 44 W/m^2 floor area (Fig. 19), there is never a need for heating the air with the boiler. The needed heating is done in the heat recovery unit. The difference in energy use between the optimal supply air temperature and a constant supply air temperature at 12°C is 8% in Sturup at 44 W/m^2 floor area internal load.

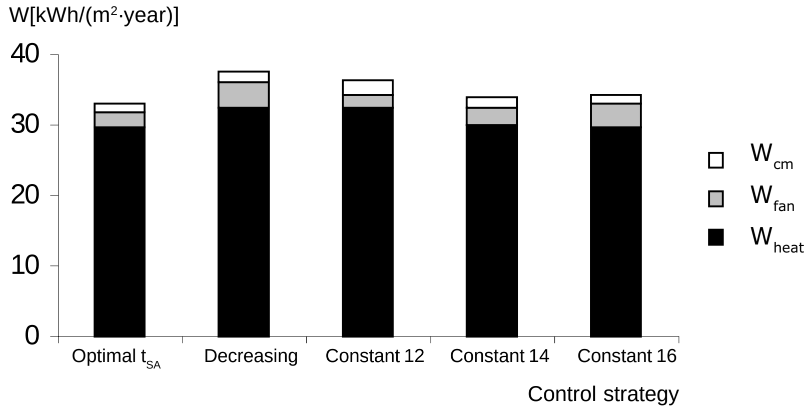

Fig. 20. Day time energy use per m^2 floor area for one zone with a facade area of 9 m^2 during one year in daytime in Luleå with an internal load of 350 W (corresponding to 26 W/m^2 floor area).

At 26 W/m^2 floor area internal load in Luleå, the sum of the fan and the chiller energy use are between 10 and 14% of the total HVAC energy use dependingon control strategy (Fig. 20).

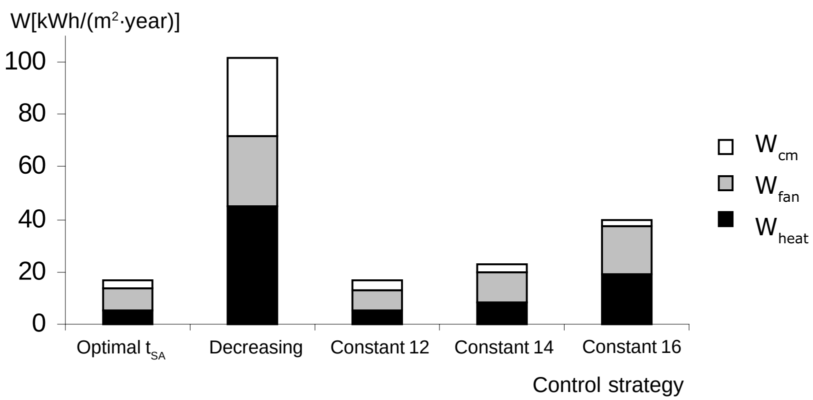

Fig. 21. Day time energy use per m^2 floor area for one zone with a facade area of 9 m^2 during one year in Luleå with an internal load of 600 W (corresponding to 44 W/m^2 floor area).

At 44 W/m^2 floor area internal load in Luleå the decreasing strategy uses six times more HVAC energy during day time than the optimal strategy (Fig. 21). The HVAC energy use is lower when the supply air temperature is controlled optimally and the internal load is 44 W/m^2 floor area than when the internal load is 26 W/m^2 floor area. This is because the HVAC energy use does not include the energy used to generate internal loads. The difference in internal load is 18.5 W/m² floor area and this results in an increased energy need of 81 kWh/(year·m²) for the internal load.

3.7 Time distribution of cases

Figures 22 and 23 show how often a certain case (see the methods section) occurs during daytime (06.00-18.00) 1977 when the supply air temperature was optimized. The optimal supply air temperature in Luleå was never below the saturation temperature (Case 4).

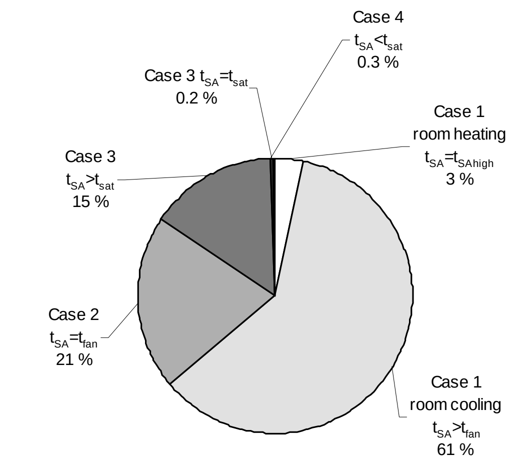

Fig. 22. Time distribution of cases for Sturup 1977 with 350 W internal load (corresponding to 26 W/m² floor area) and optimized supply air temperature.

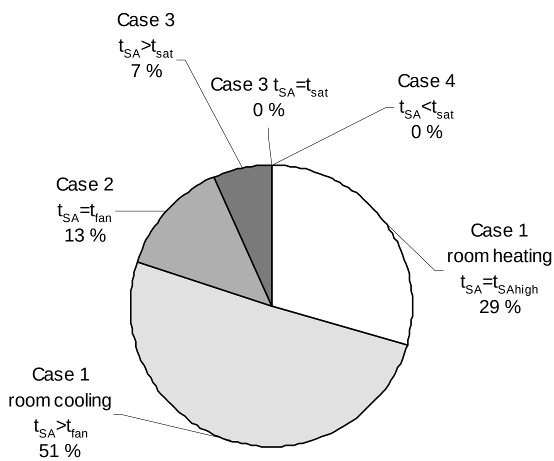

Fig. 23. Time distribution of cases for Luleå during 1977 with 350 W of internal load (corresponding to 26 W/m² floor area) and optimized supply air temperature.

3.8 Optimal U-value with optimal supply air temperature

Figures 24 and 25 show the annual energy use as a function of the average U-value of the facade in Sturup and Luleå respectively. The zone temperature set point was 20°C during nighttime when not occupied ($P_{internal}$=0) and 23°C during daytime. When not occupied, the minimum supply air flow (q_{min}) was 5 l/s corresponding to the Swedish building code. Only when the internal load is active more than 12 hours a day, an optimal U-value above zero can be found.

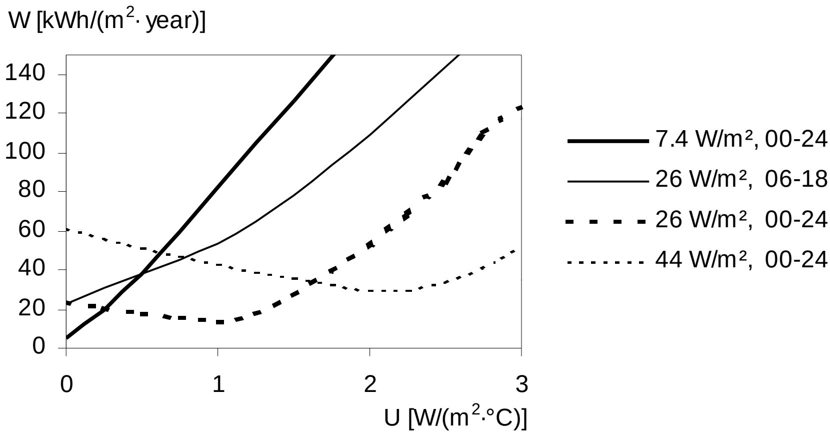

Fig. 24. One year energy use per m² floor area in Sturup for different U-values, occupied hours and internal loads ($P_{internal}/A_{floor}$).

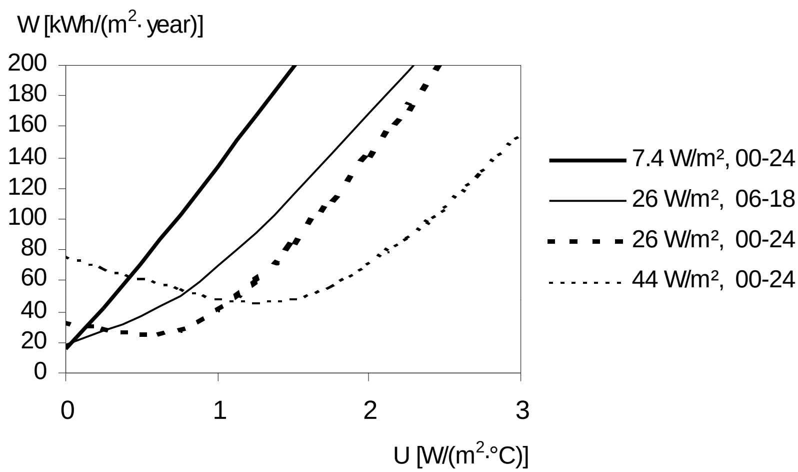

Fig. 25. One year energy use per m² floor area in Luleå for different U-values, occupied hours and internal loads ($P_{internal}/A_{floor}$).

An optimal U-value depending on climate, occupied hours and internal heat load is calculated and the result is shown in Fig. 26. If the building is only used during day-time hours (06-18), the internal load has to be above 135 W/m^2 in Sturup and above 175 W/m^2 in Luleå in order to get an energy optimal U-value higher than zero. U-values less than zero were excluded because they are not applicable in practise.

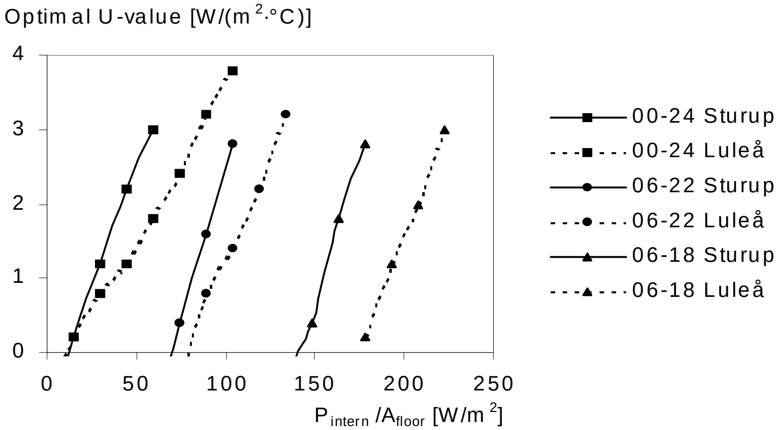

Fig. 26. Optimal U-value depending on internal load, occupied hours and outdoor climate.

4. Discussion

There is a major potential in controlling the supply air temperature optimally to reduce the HVAC energy use. A comparison of the energy use between a constant supply air temperature at 12°C and the optimal strategy shows a difference of only 8% in Sturup with an internal load in a zone of 44 W/m² floor area (Fig. 19). This is a rather small difference, though this is only true if the internal loads are constant and that is not the case in practice. If the internal load would change from 44 W/m² floor area to 26 W/m² floor area, still comparing the same strategies, it would result in a difference of 27% (Fig. 18).

Considering the Figs. 12-15 showing the power input, the risk of a high increase in power requirement when having a lower supply air temperature than the optimal is relatively small, but a higher temperature could result in a relatively high increase of power requirement.

The HVAC energy use in Luleå with 44 W/m² floor area internal load (Fig. 21) is lower than when the internal load is 26 W/m² floor area (Fig. 20). If the difference in load is electrical equipment and not people, 81 kWh/m^2 must be added because of the extra 18.5 W/m² floor area internal load. Therefore, decreasing the internal loads leads to lower sum of HVAC energy use and electrical equipment energy use.

In the optimization there are no economical aspects and all energies are treated as equal. If economical aspects are taken into account, the constants in the equations must be revalued. Installed power and the unit sizes influence on the HVAC system performance have also not been taken into consideration.

The heat recovery unit can also be used in reverse, by cooling the supply air with the exhaust air. Especially in warm climates this can reduce the energy use additionally. In the studied climates there are very few hours when this would be possible and therefore the energy saving potential for this is negligible.

When the outdoor temperature is high, the relative humidity is most often low in the studied climates. Therefore, an extremely high relative humidity will not occur unless the internal moisture production is exceptionally high.

Supplying air at 12°C to a room controlled on occupancy level instead of temperature can result in a room temperature that is too low. Installing an air pre-heater in the duct system can solve this problem but will affect the energy calculations. Also supplying air at a lower temperature than 18°C at low velocity and at floor level (displacement ventilation) can cause draught problems. Placing an ejector before the displacement air terminal can solve this problem.

If the supply air temperature is optimally controlled, Figs. 22 and 23 show that the supply air temperature will almost never (0.3% in Sturup) be below the saturation temperature. To make the control strategy easier to implement, the supply air temperature can be controlled to never be below saturation temperature without any risk of a significant increase in energy use.

Using an exponent (i) for the fan equations close to the theoretical value, 3, would, a s Fig. 16 shows, not result in any significant difference between the calculated set point and the optimal set point. When Figs. 12-15 are considered, the effect on the energy use would be even less.

When a building consists of several zones with different zone temperature set points and one t_{zone} is used in the equations for a single zone, the result will not be the optimal supply air temperature. However, Fig. 17 shows that it is possible to use the exhaust air temperature as a zone temperature set point in the equations for a single zone with a negligible increase in energy use.

All energies are treated as equal but the COP for the chiller can be considered as the cost relation between 1 kWh fan electricity and 1 kWh cooling energy. In the examples shown in the result section 1 kWh fan electricity costs three times more than 1 kWh cooling energy. The chiller COP can vary depending on supply air temperature and outdoor temperature. The level of variation is dependent on what kind of cooling system that is used. If the variations in COP are significant, the equations for the optimization should be adjusted.

As can be seen in Figs. 24-26, the optimal U-value regarding energy use is only above zero when the internal loads are very high or when the loads are active more than 12 hours a day. The energy use might be reduced at a higher U-value but the penalty might be a poorer thermal comfort and an increase in energy use if the use of the building is changed.

To be able to control the temperature in an optimal way, data about the fan, chiller, heat recovery unit, internal loads, zone mode, zone temperature set point and the outdoor temperature are needed. All these parameters are known in a modern HVAC system with individual control. By implementing the equations into a control strategy, the energy use in a 100% outdoor VAV system can be reduced. When a system has been taken into operation, it is almost impossible to control the efficiencies of the products, but the supply air temperature can be controlled and optimized to decrease the energy use. Therefore, the equations can be efficient to implement in existing 100% outdoor VAV systems if the parameters mentioned above are known.

The optimization of the supply air temperature regarding energy use can be combined with other alternatives to reduce the energy use and also with a broader life cycle perspective. This is not stressed in this paper but can imply further research.

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