THESIS FOR THE DEGREE OF LICENTIATE OF ENGINEERING

Demand-based control of indoor climate in office buildings

Design of local and central feed-forward control systems for high comfort, low energy use and low peak power

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Building Services Engineering
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Abstract

In this work, a number of novel strategies for controlling the heating, ventilation and air-conditioning (HVAC) systems in office buildings have been designed and evaluated. The purpose of the control strategies is to improve the match between what is supplied by the HVAC-systems and what is required to achieve a desirable indoor climate in the building. This purpose is based on the conjecture that when this match is perfect, both the energy usage and the required peak power of the HVAC-components are minimized.

Most of the novel control strategies are based on feed-forward (FF) control, i.e. to utilize information about measurable disturbances by the control system, and the strategies are designed to manage a number of different tasks in the HVAC-systems. When it comes to local control, the work has focused on methods for both thermal climate and indoor air quality control. On a central level, the work has focused on the control of supply air temperature as well as night-mode strategies. These task specific strategies were primary evaluated separately but the most favourable ones were also combined in a consecutive study. Practical issues regarding implementation were also identified experimentally and their impact on the performance of the novel strategies was determined.

The studies have been conducted in a simulated environment but the aspect of implementation has always been kept in mind. For that reason, both ideal strategies as well as more practical implementable ones have been evaluated in parallel.

The most important finding is that there is a large potential of controlling the ventilation system using the methods that have been designed in this work. However, when it comes to thermal climate control, the difference in performance when the novel and conventional strategies are compared is rather small.

In summary, a couple of conditions that are affecting the potential of FF-control have been identified, such as the thermal mass of the building, ambient climate etc. For all possible combinations of these conditions, the most feasible FF-designs resulted in that the control tasks could be performed using substantially less energy compared to when conventional controllers were used. Further, the required peak power of the HVAC-components could at the same time be reduced.

Keywords: Building services engineering, energy efficiency, central control, local control, demand based control, feed-forward, disturbance rejection, HVAC, office buildings, IAQ, thermal climate

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I owe enormous gratitude to my parents and brother for their lifelong support and encouragement. Thanks for making me the person I am today. Most of all, my greatest appreciation is reserved for my wife and son. Thanks for all the love and happiness you have given me. You are my everything.

Göteborg, August 2012

Mattias Gruber

Contents

Symbols, abbreviations and definitions

The system of designations which is used in this work is well established in the physics of heat, power and energy technology. The system is used, with minor variations, in the major part of international literature on thermodynamics, fluid dynamics and heat transfer. The system is based on the following main rules:

- *Upper case letters* designate absolute quantities.
- *Lower case letters* designate specific quantities.
- *A letter capped by a dot*, i.e. a time derivative, designates a flow quantity.
- *The same letter is used for one quantity*, whether it is in its absolute, specific or flow form.

Symbols

This section provides a comprehensive list of all symbols and designations used in this work. The section lists symbols for physical quantities divided into Latin letters and Greek letters, subscripts and some fundamental dimensionless numbers. As far as possible, abbreviations, definitions and designations according to international or Swedish standards have been used.

Latin letters

A	area	[m^2]
c	concentration	[PPM]
c_p	specific heat capacity at constant pressure (fluids)	
d	distance (thickness of a layer)	[m]
d, D	diameter	[m]
d_h	hydraulic diameter	[m]
E	total energy, weighted sum of energy types	[W]
f	coefficient of friction pressure loss	[-]
G	dynamical transfer function	
H	height	[m]
h	opening	[-]
L	length	[m]
m	mass	[kg]
$\dot{M}$	mass flow rate	[kg/s]
$\dot{m}$	specific mass flow rate (mass velocity)	[kg/(s·m^2]
n	number	
n	rotational speed	[revs/s]
p	pressure (pressure difference is designated Δp, see Δ)	[Pa]
Q	thermal energy	[J]
q	specific thermal energy	[J/kg]
$\dot{Q}$	thermal capacity	[W]
$\dot{q}$	specific heat load	[W/m^2]

R	convection to radiation coefficient	[-]
r	radius	[m]
SFP	Specific Fan Power	[kW/(m^3/s)]
T	thermodynamic (absolute) temperature	[K]
t	celsius temperature	[°C]
U	thermal transmittance (total coefficient of heat transfer)	[W/(m^2·K)]
u	general input signal	
v	velocity	[m/s]
V	volume	[m^3]
$\dot{V}$	volume flow rate	[m^3/s]
W	work (mechanical or electric)	[J]
$\dot{W}$	power (mechanical or electric)	[W]
x	length coordinate	[m]
y	general output signal	
y	length coordinate	[m]

Greek letters

α	heat transfer coefficient for convection	[W/(m^2·K)]
β	authority	[-]
Δp	pressure difference	[Pa or kPa]
ΔT	temperature change; see figure A1.1	[K or °C]
λ	thermal conductivity	[W/(m·K)]
η	efficiency	[-]
η_t	temperature efficiency	[-]
ρ	density	[kg/m^3]
τ	time	[s]

Subscripts

Subscripts are usually self explanatory and written with lower case letters. The order is: medium and/or component + position, e.g. for temperatures $t_{a,i}$ (air inlet), $t_{a,e}$ (air, exhaust) or t_{ah1} (air-heater inlet). If there is only one medium under consideration, then that particular index may be excluded. Some frequently used abbreviations are listed below:

Medium

a	air
w	water

Function

t	temperature (efficiency)
p	constant pressure
v	constant volume

Position

i	Inlet
o	Outlet

s	Supply
e	exhaust (from room)
x	extract (from building)
o	outdoor
r	Room

Abbreviations

AHU	Air handling unit
CAV	Constant air volume
DOAS	Dedicated outdoor air system
FB	Feed-back
FCU	Fan-coil unit
FF	Feed-forward
HRX	Heat recovery exchanger
HVAC	Heating, ventilation and air-conditioning
IAQ	Indoor air quality
LHS	Left hand side
MPC	Model predictive control
N	North
OF	Occupancy factor
PI	Proportional and integrating
PMV	Predicted mean vote
RHC	Receding horizon controller
RHS	Right hand side
S	South
SAT	Supply air temperature
SFP	Specific Fan Power

Definitions

In this work, the following definitions apply:

Dimensionless numbers

Re Reynolds number; $\mathrm{Re} = \dfrac{u \cdot l}{\nu}$

Conditioned space

Air-conditioning: Modifying the thermal state or composition of air by heating, cooling, humidification, dehumidification, removal of particles and removal or addition of gases.
Conditioned: See air-conditioning.
Conditioned space: Enclosed space, conditioned to control the thermal climate and air quality according to specified requirements.
Room: Room is taken in a general sense and may be a residential room, an office, a hospital ward, a supermarket, a cold storage facility, a process space, the passenger compartment of cars, buses, trains, air-planes etc.

Ventilation

Ventilation (ASHRAE): Process of supplying or removing air by natural or mechanical means to or from a space. Such air may or may not have been conditioned (= air treated to control its temperature, relative humidity, purity, pressure, and movement).
Comment: Ventilation may be controlled or uncontrolled (infiltration/exfiltration). In this context only controlled ventilation will be discussed.
Ventilation system: Technical system designed to supply or remove air to or from a space. This may include conditioning of the air by supply or removal of heat, moisture, particles, gases, etc. The system may be designed for a constant air volume flow rate, CAV, or a variable air volume flow rate, VAV. The flow of a VAV system may vary according to a predetermined pattern or it may be determined by actual demand, i.e. demand controlled ventilation, DCV.

Air-streams

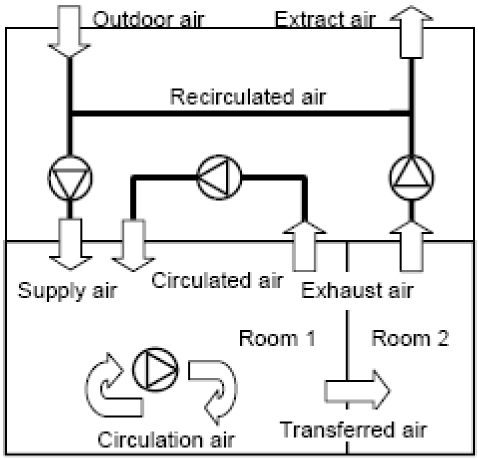


Figure 1 Different types of air flows inside, into and out of the room

According to figure 1:
Circulated air: Air that is circulated inside a room or exhaust air returned to the same room (c.f. recirculated air). Circulated air that is mixed with supply air is sometimes known as secondary air.
Extract air: Exhaust air delivered to the open air outside a building (c.f. recirculated air).
Exhaust air: Air that is removed from a room (c.f. exit air, supply air, recirculated air).
Outdoor air (not fresh air!): Air in or from the open air outside a building.
Recirculated air (return air): Exhaust air that is returned to a group of rooms (c.f. circulation air, supply air). Recirculated air may be a mix of circulation air and transferred air.
Supply air: Air that is supplied to a room. Supply air may be outdoor air, circulation air, return air or transferred air.
Transferred air: Air that is transferred from one room to another room.
Infiltration: Air entering a building through leaks in the building envelope (unintentional ventilation).
Exfiltration: Air leaving a building through leaks in the building envelope (unintentional ventilation).

1 Introduction

The purpose of a heating, ventilation and air-conditioning (HVAC) -system is to provide the services needed to maintain the thermal climate and the indoor air quality (IAQ) within certain limits. In turn, these limits, often specified by standards and guidelines, are set to ensure that comfort, health, productivity and functions of processes are promoted by the indoor climate. These services require energy and that energy has to be generated[7]. In the European Union, the building sector accounts for approximately 40 %[6] of the total energy usage whereof 76 %[51] is used by the HVAC-system for comfort control.

The Swedish parliament decided in 2006 on a national energy plan which states that the specific energy use in residential and commercial buildings, which consists of 35 %[2] of the national energy use, should be halved until year 2050 compared to the 1995 level. An intermediate target was set to a 20 % reduction until the year 2020[2]. These goals also stress the aspect of energy efficiency when it comes to HVAC applications.

1.1 Background

The energy efficiency aspect is most often viewed from an HVAC-system <u>or</u> from a building point of view. An energy efficiency analysis of the HVAC-system can either focus on the efficiency of sub components or on how the system performs in total. For example, the focus can be to increase the electrical efficiency of fans and pumps or decide on a plan how to allocate the overall work to the single components. If instead the building is targeted, the purpose can for example be to minimize thermal losses or to mitigate the effect of solar heat gain.

In these types of analyses it is assumed that the prerequisites of one of the system are known and the purpose is then to make the other system fulfilling these requirements using as less energy as possible. Hence, the systemic[1] aspects are left out which means that the risk of focusing on measures with an overall small impact is always imminent.

Another type of analysis is to extend the control volume around the entire system, i.e. the HVAC-system and the building. A large control volume requires a systemic analysis which on one hand can account for various system effects but on the other can be quite complex.

In this work, the focus is on the compatibleness between the HVAC-system and the building. This work is performed by analysing the requirements of the building (or rather by the people inside) and how these can be achieved by the HVAC-system. Hence, the energy efficiency measures cannot be allocated to the building or to the HVAC-system separately but rather to the interface between them.

[1] **Systemic** refers to something that is spread throughout, system-wide, affecting a group or system. In this context, systemic denotes a system induced effect, i.e. an effect only visible when the entire system is taken into account

In this work, the services provided by an HVAC-system, and in turn the associated energy use, are distinguished by what's demanded and what's supplied. The term "supplied" refers to the quantities (such as thermal power or fresh air supply) actually delivered at room level and the demand refers to the quantities required to achieve the desired indoor climate. Demand is decided by the building itself, the requirements on indoor climate, user pattern as well as internal and external disturbances. The energy use of the HVAC-system is determined by the energy required to provide the services associated with the supply and <u>for any given system, the energy use is minimized if the supply exactly matches the demand</u> [26].

1.1.1 HVAC-control system aspects

The operations of an HVAC-system are managed by a control system. Its purpose is to control the individual components so that they together provide the required services. Naturally, the control system plays a large role when it comes to the energy usage since the supply is determined by the signals of the control system. This means that the control signals preferable should be closely connected to the demand which for varies reasons is hard to achieve using conventional methods.

The supply generated by conventional controllers is based on measurements which in many cases is a vague representation of the actual demand. For example, there is a need for supplied heat or cooling which in conventional system is determined indirectly by a temperature deviation. The result is a connection that consists of a number of simplifications that increases the distance between supply and demand [25].

Another problem is that the measurements are associated with delays on both short and long time scales. The short scale is related to the measurement system and is influenced by sensor time-constants and transport-delays in the system. The long scale is related to the combined inertia of the HVAC-system and building. The indoor climate is a result of the combined effect of the building itself, the activities in the building, the outdoor climate and the services of the HVAC-system. This means that the demand will vary instantaneously when some of these factors are varied. However, the full effect of the new conditions will not be seen in the measurements until long after[5].

As indicated, in order for a control system to reach high performance, the control actions must reflect the behaviour and features of the process. Only then can the supply generated by the HVAC-system match the demand. This requirement is not fulfilled by conventional controllers in any sense. For example, conventional controllers are linear but are set out to control systems that in normal cases are non-linear. In the best case scenario, the characteristics of the control system will match the characteristics of the process in certain operational-points. However, if the operation evolves away from these points, the match between the supply and the demand is aggravated. Further, conventional P-controllers, which are common in room temperature control, are static but are set out to control dynamic systems. They are operated under the assumption of equilibrium conditions which hardly can be reached in reality.[44]

1.1.2 Installed capacity of the HVAC-components

Naturally, also the peak power required by the HVAC-components is minimized when the match between supply and demand is perfect. The design of conventional controllers results in that the generated supply is based on the control error. This means that the controller will not act until the effect of the disturbance is detected by the measurements. Due to the inertia of the process, the effect of the disturbance is dampened which means that the supply increases or decreases slowly. However, the effect of the disturbances is accumulated in the process which means that the controller has to increase the supply substantially to be able to turn the increasing trend when the deviation from the setpoint is further increased. Hence, over the duration of a disturbance, the supply of a conventional controller is typically too small in the beginning and too large in the end. Of course, the required peak power is determined by the largest supply. This means that the installed capacity of HVAC-components must be larger than the actual demand for power in order for the conventional controllers to perform their task.[26]

Reduced installed capacity is desirable from an economic point of view. However, another problem is that the full capacity of large units is very seldom used which is indicated by a low load factor. According to the German standard VDI2067:20, the match between supply and demand is typically aggravated in units with a low load factor. The reason is that the relation between input and output (i.e. characteristic) is often steeper at low loads as exemplified in figure 1.1 for an air-coil. Hence, a small input will result in a large output which means that the control task becomes more difficult.[26]

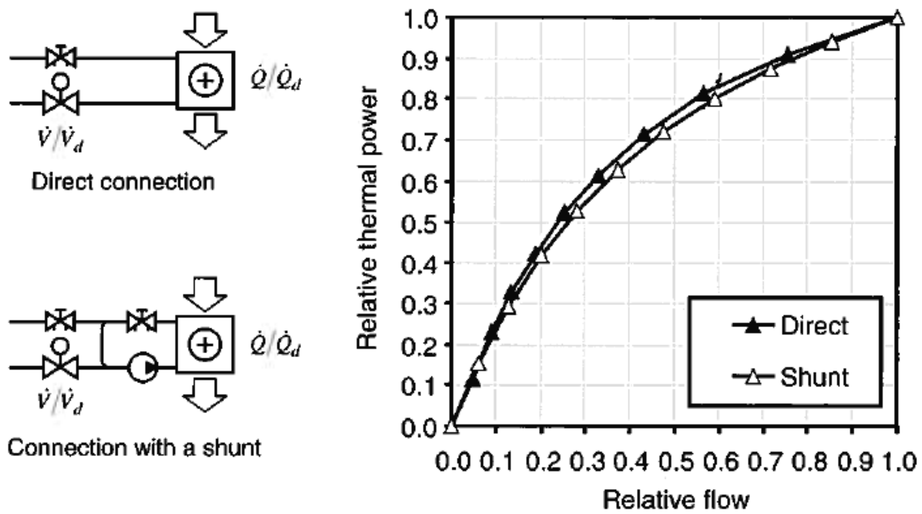


Figure 1.1 Example of flow characteristics of the same air-coil, for both varied (direct) and constant (shunt) coil flow[66].

1.2 Purpose

The aim of this work is to improve the match between the supply from the HVAC-system and the actual demand in office buildings by mainly using control related measures. The overall goal is to supply the correct quantities, at the right place at the right time. The work focuses on control of thermal climate regarding room air temperature and the IAQ regarding the room CO_2 concentrations. Their corresponding levels are controlled by the means of thermal energy and fresh air supply. Hence, in this work the term “supply” refers to the amount of fresh air as well as the cooling and heating energy actually delivered to a room. The term “demand” refers to the same parameters but instead to the quantities required at room level to maintain the desired levels. Compared to conventional control systems, the expected outcome is decreased energy usage regarding electricity, heating and cooling as well as reduced peak powers of the different components related to the HVAC-system. At the same time, the thermal comfort as well as the IAQ should either be improved or at least maintained.

This work focuses on the design of local and central control strategies. The most important alternative control method that is developed and tested is feed-forward (FF) control. The main characteristic feature of FF-control is that the control signals are partly generated by including information about measurable disturbances during operation. FF is a big part of this work and the most important questions to be answered are; what kind of information can be available and how is this information best utilized in the control system. In more direct terms; what is the gain of including disturbances in the control, how can these be measured and how should these measurement signals be processed and used in the control system.

1.3 Methodology

The work covers local control of thermal climate and IAQ, central control of supply air temperature (SAT) as well as night-mode strategies for utilization of free-cooling. Related to these control tasks, alternative control methods based on knowledge of the disturbances on the system have been designed and evaluated. Since the concept of energy efficiency is a relative measurement and can therefore only be regarded as relevant in the context of a comparative study, the evaluation was primary carried out by comparing alternative and conventional control strategies in pairs.[7]

The design and evaluation of alternative control strategies is primarily based on a set of simulation studies carried out in steps. Each step is focusing on a certain control task and the aim is to identify and select the most preferable alternative strategies for further testing. The studies aim to be broad and are carried out to take the most relevant effects into account. The layout of the studies depends on which type of control task considered and the conditions relevant for the case.

An experimental study has also been conducted to identify practical limitations related to implementation of the alternative strategies. The impact of these limitations on the control performance was tested in a sensitivity analysis presented in chapter 10.

1.3.1 The simulation platform in short

In short, the simulation platforms consist of a building part and an HVAC-part. The building part consists of either single rooms or part of buildings in different configurations and designs. These are subjected to various disturbance profiles and were simulated using both conventional and alternative control strategies.

The same type of HVAC-system has been used throughout the simulation studies. The layout and configuration is discussed in detail in chapter 3 and a brief overview is only given in this part of the work. To maintain a desirable indoor air quality, each zone is supplied with hygienic ventilation. The supply air is conditioned in a central air handling unit (AHU) primary consisting of one heat recovery unit (HRX), one heating coil, one cooling coil as well as two central fans; one for supply air and one for exhaust air. The ventilation air is distributed to the rooms via ducts and roof mounted air-diffusers.

The thermal control of the zones is primary handled by local fan coil units (FCU) consisting of one heating coil, one cooling coil and an integrated fan. Room air is circulated through the unit and is either heated or cooled[66].

The HVAC energy usage is characterized as either thermal energy or electric energy. The thermal energy is used for central air-conditioning or locally by the FCUs. The electrical energy consists of drive power to the central fans, the fans integrated in the FCUs as well as pumps for distribution of hot or cold water to the air-coils.

1.3.2 Criterions for comparability

The performance of the different control strategies has been compared on the basis of two criterions of the control system;

- one is related to the room temperature
- and the other is related to the room CO_2 concentration

When two control systems are compared, the criterions that are taken into consideration must be fulfilled. Then, it is regarded that both of the control systems have fulfilled the control task with the same satisfaction. These two control systems are then comparable and are referred to as corresponding.

The two criterions are primary based on the prevailing standards and guidelines related to IAQ and thermal climate in Swedish office buildings.

Regarding CO_2 concentration, the task of the controller is to avoid exceeding a threshold limit value, set to 1000 PPM according to Swedish guidelines.[21] Two control systems are regarded as corresponding from and IAQ point of view if their resulting maximum CO_2 peaks correspond to 1000 PPM. This criterion is managed by tuning the setpoint of the controllers so that the CO_2 peaks are coinciding.

The criterion on thermal climate is instead based on two boundaries, one upper and one lower, of the room temperature. The area between the two boundaries is denoted as the dead-band. The boundaries of the dead-band are referring to the maximum and minimum room air temperatures that are allowed to be maintained for a longer time, i.e. the boundaries are allowed to be breached momentary. The criterion is instead related to the number of degree-hours outside the dead-band[10], i.e. the area limited by the room temperature and the two boundaries in a temperature-time plot. Hence, short overshoots with large amplitudes can fulfil the criterion just as long overshoots with low amplitudes can. Two control systems are regarded as corresponding from a temperature point if the resulting degree-hours above as well as below the temperature dead-band are the same. This is illustrated in figure 1.2 for four room temperature evolutions that are corresponding to each other.

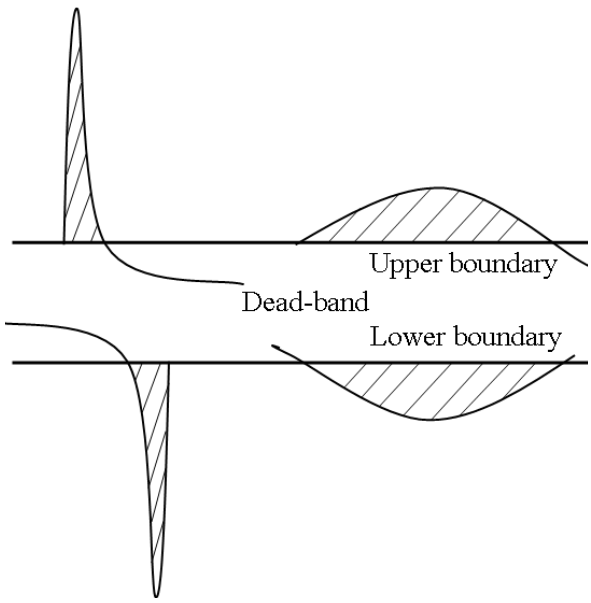


Figure 1.2 Example of four exceedings of the temperature dead-band that are corresponding according to the criteria for thermal climate. The areas outside the boundaries, i.e. the degree-hours outside the dead-band, are the same in all four cases which means that the corresponding control methods are comparable from a thermal climate point of view

1.3.3 Evaluation method

The performance of the alternative control strategies is evaluated relatively to conventional control systems. Hence, the performance is given as an improvement or aggravation compared to using conventional control systems. <u>The evaluation is performed using a set of performance indicators related to energy usage and required peak power of the most essential HVAC-components</u>. The indicators are calculated using equation 1.1, in which *x* denotes the quantity the indicator is focused on.

$$1-\left(\frac{x_{FF}}{x_{FB}}\right) \text{ [-]} \qquad \text{(eq. 1.1)}$$

Equation 1.1 can be interpreted as the percental savings in energy usage or peak power induced by the alternative control system. For example, an energy indicator of 0.5 correspond to that half of the energy usage is saved by using alternative strategies instead of conventional. A value of zero or less indicates that the alternative control system does not add to any improvements or to an aggravation of the operation.

The energy usage performance is divided into three separate indicators representing the relative usage of thermal, electrical and total energy. The thermal part is either refereeing to cooling or heating energy and consists of the combined thermal energy for central air-conditioning and thermal energy supplied by the FCUs. The electrical energy consists of the combined driving energy of the central fans and the fans integrated in the FCU. Hence, the electrical driving energy for pumps is neglected due to their insignificant contribution; this part represents about 1 % of the total electricity usage. The total energy usage is calculated according to equation 1.2. It is a combination of the thermal and electrical part in which the electricity is weighted with a factor of 2.5 according to the Energy Efficiency Directive 2006/32/EG[1].

$$E_{tot} = Q_{thermal} + \left(W_{electrical} \cdot 2.5\right) \text{ [kWh]} \qquad \text{(eq. 1.2)}$$

The peak power indicator are referring to the most essential HVAC-components; the central supply air fan, the FCU and the AHU. The purpose is to compare the differences in required installed design capacity of these components when conventional and alternative control strategies are used. The indicator denoted as "electrical power savings" are referring to the driving power of the central fan. The FCU and the AHU power indicators are referring to the thermal powers required by these components. The required dimension of the fan integrated in the largest FCU is omitted in this work since it is assumed to be reflected by the thermal power requirement of the FCU.

Special cases
In some studies, the evaluation is performed for both summer and winter ambient conditions and is extending over a number of days and nights. To maintain the temperature setpoints, both heating and cooling might be required in those cases. Then, one indicator is dedicated for heating energy and one for cooling. The total energy usage is calculated according to equation 1.2 in which heating and cooling is then treated as equal. The corresponding peak powers of the HVAC-components refer to the maximum design capacity indicated in the corresponding study. Hence, the peak power during winter refers to required heating power and during the summer it is the cooling power that is referred to. In the cases were many rooms are simulated together, the peak power of the FCU refers to the room with the largest demand for heating or cooling.

1.4 Structure

In the following chapter, control theory with focus on controller design is reviewed. This part brings up the different controller principle used in this work, such as feed-back, feed-forward and model predictive control, and the purpose is to facilitate for the reader in the following chapters.

In chapter 3, the simulation platform, including building and HVAC-system is reviewed. This chapter include descriptions of the models, model parameters, systems as well as a model validation.

In chapter 4 are the fundamental elements of the FF-controller designs that are used in the rest presented. This chapter focuses on the designs of the two fundamental components of a FF-controller; the FF-sensor and the FF-filter.

From chapter 5 to chapter 8 different FF-control strategies for local and central control are proposed and evaluated for different control tasks. In each chapter, FF-control system with the best performance according to the indicators in section 1.3.3 is selected.

- Chapter 5 is dedicated for FF-controllers on a local level
- Chapter 6 focuses on which parts of the local HVAC-system that preferable are controlled by FF methods
- Chapter 7 is dedicated for central night-mode control strategies
- Chapter 8 is dedicated for central supply air temperature control

The selected strategies are combined in chapter 9 and the combined effect is evaluated. In chapter 10, the sensitivity of the less robust selected strategy is evaluated.

In chapter 11 and 12, the work is summarized, discussed and further work is proposed.

2 Theory

A control system typically consists of three components;

- A sensor which functions as an interface between the controlled system and the control system
- A controller used to generate control signals based on the sensor output
- An actuator used to transform the control signal into a supply quantity of some kind

In some sense, the overall purpose of a control system is to optimize the operation of the system with respect to some criteria. In HVAC-applications, common criterions are for example comfort, energy usage and economy. Symbolically, the control system acts as "glue", joining the separate subsystems into one functioning unit. Hence, the system should have a complete view of the system in order to optimize the overall function.[25]. In the text below, the controllers used in this work are reviewed both from a general and an HVAC point of view.

Designations
In the following text some designations are introduced that requires a further explanation.

- An input signal to a system is denoted as either *u* or *v* and an output as *y*. The symbol *u* refers to a controlled input generated by an external system or a subsystem and *v* to an uncontrolled input signal acting as a disturbance. An output signal is defined as a measurable response of the process to the input signals.
- What happens in between the input and output signal is determined by a function denoted as *G*. This function determines the dynamic and static transformation between input and output signal and refers typically to the Laplace transform of an ordinary differential equation.
- The symbol *r* refers to the reference signal to the controller and *e* as the control error, i.e. the difference between the output and the reference.

Since the controllers in this section in many times are studied from an general point of view, the term "state" is used to denote a value of an arbitrary controlled variable and the term "states" to denote the values of all controlled variables.

2.1 Feed-back (FB) controllers

The term "conventional controllers", which is used extensively in this work, are referring to controllers of feed-back type. The term denotes a group of controllers with the common feature that the control signals are generated by comparing the measured (or actual) value of the controlled variable to a reference value. The difference between them, also referred to as the control error and denoted as *e*, is the input to the controller.

In figure 2.1, the principal structure of a process controlled by a FB controller is presented. The properties of the process is divided into three parts; the influence of the actuator denoted as G_u, the influence of disturbances denoted as G_v and the influence of the sensors used in the control denoted as G_y. The controller acts on the system by generating a control signal based on the control error which in turn is send to the actuator.

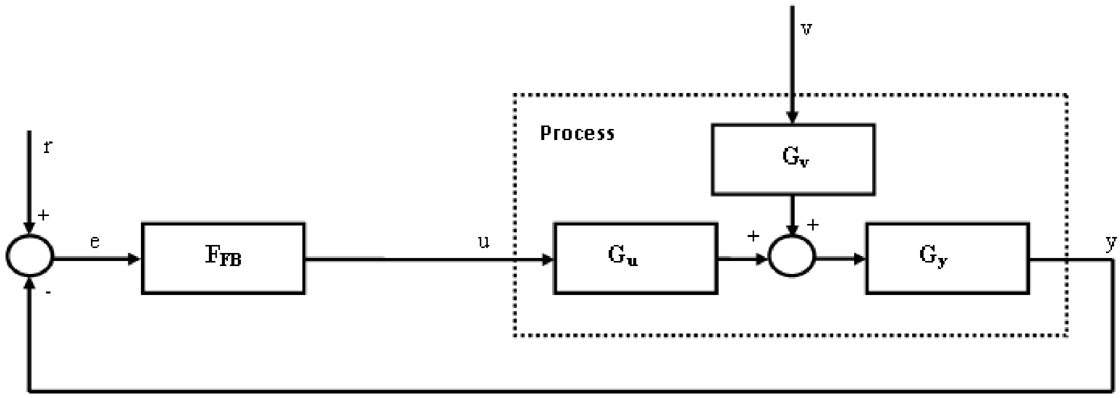


Figure 2.1 Schematic picture of a FB-controller and the controlled process

The purpose of a FB-controller can be divided in two tasks; reference signal tracking and disturbance compensation. The first named is also referred to as the servo problem and, in this case, the purpose of the controller is to control the process according to a trajectory given by a reference signal. In this work, the purpose of the FB-controller is to perform disturbance compensating measures. This means that the reference signal essentially is a constant and is referred to as a setpoint. In this case, the purpose of the FB-controller is to keep the process as close as possible to the setpoint by minimizing the impact of disturbances. This is done by measuring the output of the process and to send counter-acting control signals to the actuator.[41]

2.1.1 Classes

The classes within the FB controller group are distinguished by how the control error signal is processed by the controller. Typically, the signal processing part is designed based on three different modules. These are referred to as the P-, I- and D-action and can either be combined or used separately. The most common designs are P, PI and PID.

In figure 2.2, the principle architecture of a FB-controller is presented schematically. In this case, the controller has P, I and D actions which means the controller is of a PID type. Other classes are formed by removing either the D-module or/and the I-module without changing the overall architecture.

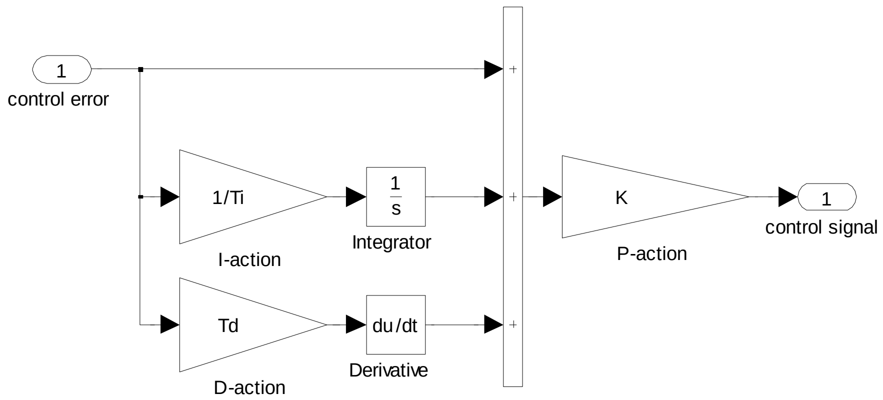


Figure 2.2 Schematic picture of a FB-controller

The corresponding mathematical interpretation to figure 2.2 is given in equation 2.1. As can be seen, the control signal u is generated by summing up the contributions of the modules.

$$u(\tau) = K \cdot e(\tau) + \frac{K}{T_i} \cdot \int_0^{\tau} e(\tau)\, d\tau + K \cdot T_d \cdot \frac{de(\tau)}{d\tau} \qquad \text{(eq. 2.1)}$$

The influence of the different modules on the final control signal is determined by parameters associated to each module. These parameters are denoted as K, T_i and T_d in equation 2.1 and are connected to the P, I and D-modules respectively. In the following text they are referred to as the static gain, I- and D-time.

When designing a FB-controller, the parameters are tuned together. This can be done in different ways but the overall goal is to maximize the performance of the controller by finding a parameter combination which optimizes a trade-off between speed and stability.[41] A low control signal activity as a response to a certain control error results in a stable but slow control. A large response results in a fast control but might also lead to instabilities. That is, the controller does not find a stationary operational point since any control signal is larger than what is required to get back to the setpoint. Hence, each control signal results in an overshoot of the setpoint.

The purpose of the figures 2.3, 2.4 and 2.5 presented below is to visualize the response of the three main modules to a step in control error of an arbitrary quantity a. The figures should be interpreted as the resulting behaviour if the control error is registered by the controller but the control signals do not reach the process. Hence, the process is uncontrolled.

The P-module is short for proportional control and corresponds to the first term on the RHS of equation 2.1. The control signal is simply generated by multiplying the control error with the static gain denoted as K. In figure 2.3, the control signal response of a P-module to a step change in error signal is presented. As can be see, the P-module acts fast but only take the present error into account when generating the control signal. If the control error is zero, the contribution of the P-action is hence also zero. This means that the P-action needs an error to generate a control signal. To obtain a control with no remaining errors, an infinite static gain is required which would lead to unstable control. Hence, a P-controller is always associated with a remaining control error.

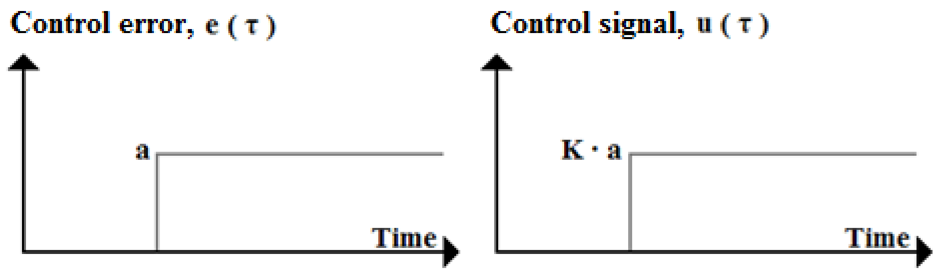


Figure 2.3 Response of the P-module (right) to a step change in control error signal of quantity *a* (left)

The I-module is short for integrating control and corresponds to the second term on the RHS of equation 2.1. The I-action is generated by accumulating past control errors, i.e. by integrating the control error over time, and its overall contribution to the total control signals is determined by I-time denoted as T_i. This parameter is interpreted as the time until the contribution of I-action equals the contribution of the P-module for a given static error. In figure 2.4, the control signal response of the I-module to a step change in error signal is presented. As can be seen, the I-action is slow but is growing for non-zero control errors. This means that the remaining error of the P-module can be removed by also including I-action. However, to avoid instability, the corresponding static gain must be decreased when also I-action is used which can result in an overall slower control.

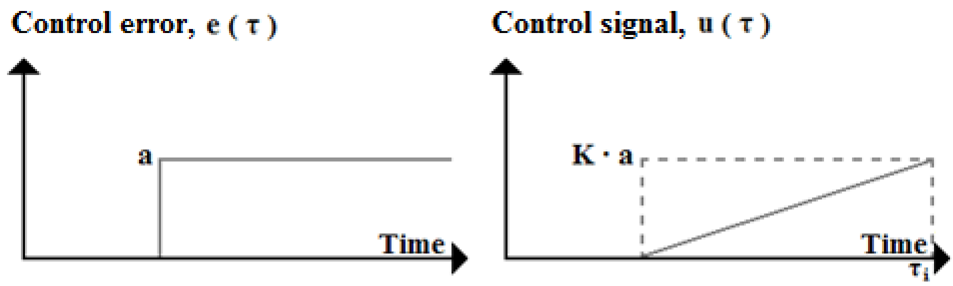


Figure 2.4 Response of the I-module (right) to a step change in control error signal of quantity *a* (left)

The D-module is short for derivation control and corresponds to the third term on the RHS of equation 2.1. The D-action is generated by multiplying the derivative of the control error with the D-time denoted as T_d. In figure 2.5, the control signal response of the D-module to a step change in the control error signal is presented. The response of the D-module to changes in control errors is rapid, but its contribution is declining for stationary conditions. The parameter T_d is interpreted as the time from that a step in control error occurs until the contribution of D-action has declined to 37 % of the contribution of the P-module. The purpose of the D-action is to compensate for future errors by taking the rate of change of the control error into account. To avoid instability, the static gain must be decreased also when D-action is included but due to its fast response, the D-action is typically increasing the speed of the controller.

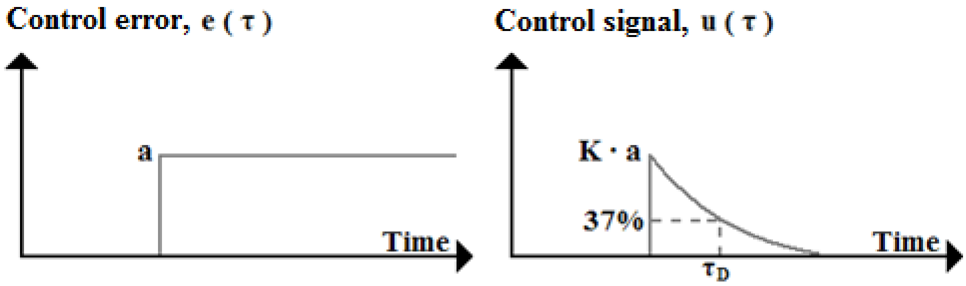


Figure 2.5 Response of the D-module (right) to a step change in control error signal of quantity *a* (left)

2.2 Feed-forward (FF) controllers

The main feature of FF-control is rejection of measurable disturbances. In this context, disturbance rejection refers to reducing the impact of a disturbance on the controlled variable. Ideally, when perfect FF-control is applied, the effect of the disturbance is not visible in the output of the controlled system. Then, the controller achieves a perfect disturbance rejection.

The basic idea of FF is to use information about the disturbances to anticipate the influence on the system and to generate a counteracting control signal.[77] In figure 2.6, (c.f. figure 2.1) a schematic picture of a FF-controller is presented. The FF-control is managed by the two main component shown in the figure; sensors for disturbance measurements denoted as F_V and a filter for transforming these signals to appropriate control signals denoted as F_{FF}. The FF-filter consists of a process model that calculates the influence from the measured disturbance, denoted as *v*, on the system. The counteracting control signal, denoted as *u*, is sent to the process via the actuators G_u.

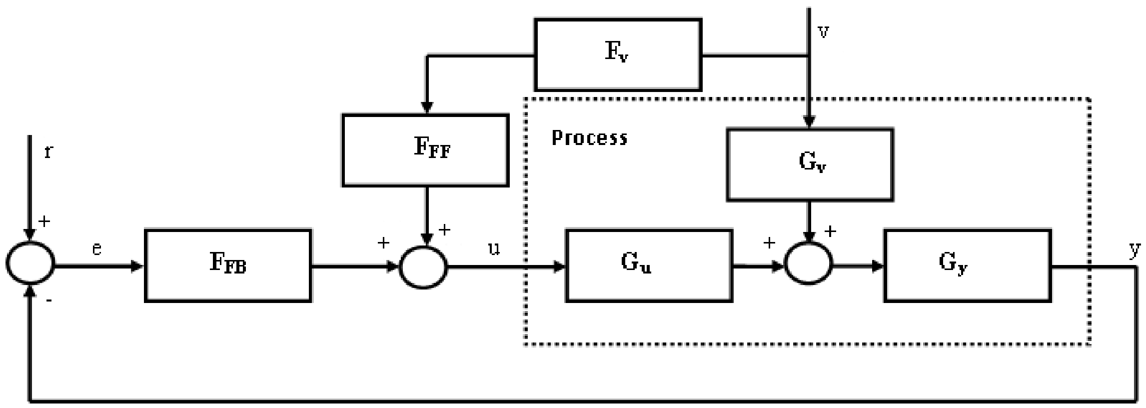


Figure 2.6 Schematic picture of a FF-controller

2.2.1 Design aspects

In the ideal case, the process model of the FF-filter satisfies equation 2.2[77]. The corresponding ideal control signals is generated by mimic the influence that the disturbance has on the process ($-G_V$). This means that the foundation of a FF-controller is a calculation process to determine the inverted dynamic response of the controlled process. The influence of the actuators is compensated by including an inverse of its dynamic (G_u^{-1}). Hence, the time-constants of the actuators are included as corresponding D-actions in the FF-controller. The result is that the time-delays in the actuators are eliminated by boosting the control signal[77].

$$F_{FF} = -G_v \cdot G_u^{-1} \qquad \text{(eq. 2.2)}$$

Since the FF-control is an open-loop compensation, the FF-filter model has to be accurate, and in many cases this is a large challenge. Due to the complexity of many systems, the FF-filter is usually associated with some kind of model simplifications or errors. Hence, it is rare that a FF-controller alone can perform the control task with satisfaction. This is solved by integrating a FB-controller, denoted as F_B, in parallel with the FF-part. In many applications the combined effect is a fast but coarse response by the FF-controller and a delayed but accurate fine adjustment of the FB-part. For example, an approximation to the ideal FF-filter described in equation 2.2, is to utilize a static FF-filter in the FF-controller combined with a dynamic FB-controller, i.e. with D, and/or I-action. This type of control system is relatively common simplification used in industrial applications.[41]

Due to the FB contribution, the demand for accuracy of the FF-filter becomes less strict and even though a relative poor model is used, the gain of FF can still be positive. For example, knowing the features of the system and actuators only within ±20 % can be enough to substantially increase the performance of the control operation compared to separate FB-controller.[23] However, there is always a limit of how low the accuracy can become and when passed, there is no longer any point of using the FF-part. In some cases the contribution of an inaccurate model can actually aggravate the performance compared to a common FB-controller.[41]

2.2.2 Fields of application examples

To many building services engineers, the term “FF control systems” seems surprisingly unknown even though this is a very common method used for temperature control in buildings. As an example, consider a hydronic heating system consisting of distributed radiators and a central boiler to produce hot water. At room level, the submitted heating power is controlled by thermostatic valves used to vary the hot water flow rate. On a central level, the hot water temperature from the boiler is determined by the outdoor air temperature (OAT) using a “supply temperature function”[62]. As an analogy to figure 2.6, the thermostatic valves are the FB-controllers (F_{FB}), the disturbance (v) is the OAT, the FF-sensor (F_v) is an OAT sensor, the FF-filter (F_{FF}) is the supply temperature function and the process is the building.

Another example is that the supply air temperature of all-air systems is in many cases is determined in the exact same manner as the supply water temperature in the previous example.

2.3 Model predictive controllers (MPC)

The designation model predictive controller (MPC) does not refer to a single strategy but to a class of control methods with some common features. Overall, the control signals of an MPC controller is generated by minimizing an objective function subjected to constraints. Hence, the control signals are a product of an optimization procedure. This is in many cases done in an iterative manner by predicting the response of the process to control signals and measurable disturbances using a dynamic process model. The result consists of an optimal future process trajectory, typically associated by the shortest route between two states, and the corresponding control signals over a finite optimization horizon divided in *N* steps.[56]

2.3.1 Principle of optimality

There are different ways to solve an optimal control problem. One of the most transparent one is to apply the principle of optimality. This principle states that an optimal trajectory between two boundary points also constitutes the optimal trajectory from any intermediate point to one of the boundary points. This principle can be applied by initiating the calculation at time N, in the point that the controller should steer the process to, denoted as the final state. The process model is then used to go backwards to the present time and solving the optimal control signals along the way. According to the principle, the optimal trajectory between the final state and the state at time N-1 is optimal independent on how the state at time N-1 was reached. By going backward to time 0 the optimal control action can be determine for a given initial condition of the process.[77]

2.3.2 Objective function

Equation 2.3 below illustrates a common objective function used in MPC controllers, along with the solution consisting of future optimal control signals denoted as *U*. The quadratic form of the objective function is suitable for optimization purposes since the so called least-square method can be used[13]. According to the previously used designations, the variable *y* is used to denote the controlled variables of the process and *u* the control signals from the controller.

As mentioned, the objective function is typically subjected to constraints which restrict the solution regarding both the values of the controlled variables, i.e. the allowed states, as well as the values of the control signals. The constraints can for example be used to include the operational interval of actuators or floating setpoints of the controlled variable such as a temperature dead-band.

$$J = \min_{U_0} y_N' P y_N + \sum_{k=0}^{N-1} y_k' Q x_k + u_k' R u_k \qquad \text{(eq. 2.3)}$$
$$U = \left[u_0', \ldots, u_{N-1}'\right]$$

By considering the first row in equation 2.3, the essence of the optimization process can be better understood. The objective is to minimize the left hand side of the objective function *J*. This is done by minimizing the sum of all the terms on the RHS consisting of variables associated with the control task along with their respective weight factor denoted as P, Q and R. The weights are design parameters and can very well be compared to the parameters associated with the FB-controller presented in section 2.1. By adjusting the weight the user can decide on the most important features of the controller and hence the primary goal of control task. The decision variables, i.e. the variables whose values can be adjusted to affect the value of the cost function, are elements in *U*, i.e. the future control signals from time 0 to *N*.

The first term in the objective function of equation 2.3 penalizes the final state by the weight *P*. Hence, if the controlled variables at time *N* ends up in a state which differs from the desired one (denoted as y_N) this term will grow. The second term includes all other state values, from time 0 to time *N-1*. In each time-step, deviations from the optimal trajectory, determined by the principle of optimality and by back-casting the process model, is penalizes in by the weight *Q*. The third and last term penalizes the control signals by the weight *R*. The purpose of this term is to avoid large control signal activities but also to enable the ability to favour some control signals and suppress others.

2.3.3 Receding horizon controller

An MPC controller can either perform the control task in an open- or feed-back loop manner. The only input to an open-loop MPC control system is the initial state. That is, information about where the process starts from. The solution is a complete sequence of control signals over the entire horizon. Since the solution is an open-loop controller the process model has to describe the system perfectly within the current range of operation. The controller will otherwise lead the system along a non-optimal trajectory.

A certain type of feed-back MPC controller is the so called receding horizon controller (RHC). This type was used in chapter 10 of this work and its main features are presented in figure 2.7[13]. By measuring the present state, the optimization problem is solved over a long but finite horizon. However, only the first step of the resulting control signal sequence is implemented. The horizon is then moved forward and the procedure is repeated again for the next time step and the corresponding state, hence the name receding horizon. This procedure is repeated until the system has converged to the final state at time N. This means that the solution of a RHC is an open-loop control signal sequence but since the open-loop calculation is performed in each time step for the present state, the overall behaviour corresponds to an MPC with feed-back function. [13]

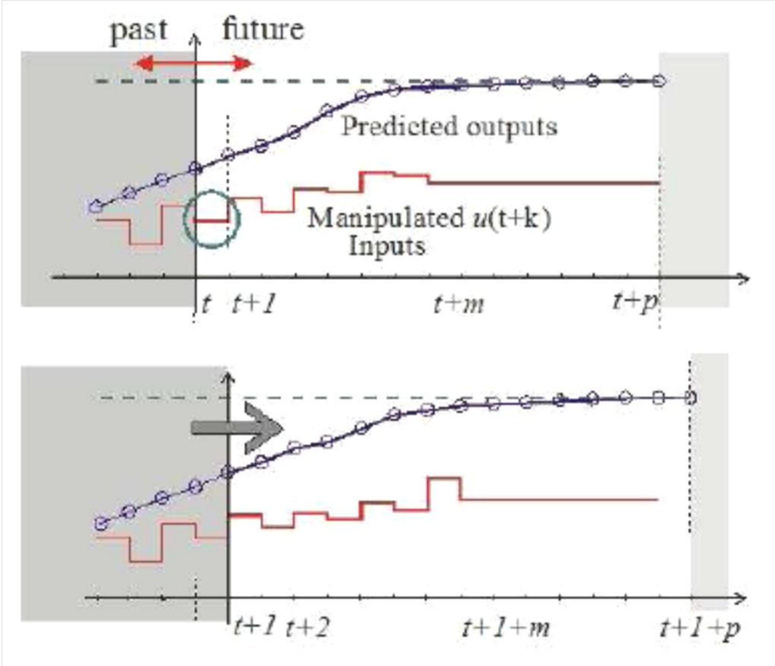


Figure 2.7 Main feature of a RHC controller[13]

A RHC controller has some nice feature. It is more robust to disturbances and model error due to the utilization of state measurements in each time step. The stability of the controller can be guaranteed if the weights in the objective function fulfil some criterions. Further, persistent feasibility, i.e. that the control task can be fulfilled without violating the constraints, can also be guaranteed in the design process. These features are further discussed in appendix F.

One of the main issue of the RHC design process is to determine N so that these desirable features are inherited. For computational reasons, one should keep N small, but not to small since then the process cannot converge to the final state. Also, the RHC can be compared to an optimal P-controller and typically lacks of I-action. This means that remaining control errors can only be eliminated if the process model is perfect. [56]

3 System description

The simulation platform consists essentially of a building part, an HVAC-part and a control system. All of these subsystems were designed so that the results from the studies are reflecting the effect of using alternative control systems as far as possible. The overall goal of the current designs was to minimize the effects which just as well can be achieved by choosing another design of the HVAC-system, the building and/or by improving the conventional control system. However, the designs are always based on what is practically achievable; either by taking practical limitations into account or by studying their effect in the sensitivity analysis of chapter 10.

The most important design features are introduced below and are more thoroughly discussed in the following chapter.

HVAC-system
The HVAC-system was designed to promote the best possible match between demand and supply as possible. The current design was chosen so that the operation determined by the control system can be fulfilled without any restrictions. Three main features of the design reflect this;

- Separate systems for IAQ and thermal climate control are used so that both the demand of fresh air and thermal power can be fulfilled without compromising each other
- Both the IAQ and thermal control are based on measurements used to indicate the demand
- The air distributions system is flow-controlled
- The minimum supply air flow rate to a zone is zero

Building
The building was designed so that the impacts of disturbances are mitigated as far as possible before they affect the rooms. That is, it was regarded that the purpose of the control system is not to compensate for effects which just as well can be avoided by an improved building design. Three main features of the design reflect this;

- The building envelope is tight
- The building envelope is well insulated
- Outside solar radiation shades are used

Control system
Both the FB and the FF control systems in the studies are designed with the highest performance as possible. The purpose was to compare the FF-controllers to the best available FB-system to isolate the effect of utilising alternative control. That is, to avoid effects that just as well could be achieved by retro-fitted conventional control systems.

3.1 HVAC-system

As mentioned in section 1.3.1, the same type of HVAC-system has been used throughout the simulation studies. It is referred to as the reference system and a schematic picture of the main components on both room and central level are presented in figure 3.1. The reference system includes the HVAC-units, various sensors and the distribution system, but not a control system. In the following studies, different strategies to control this system and its subcomponent are evaluated.

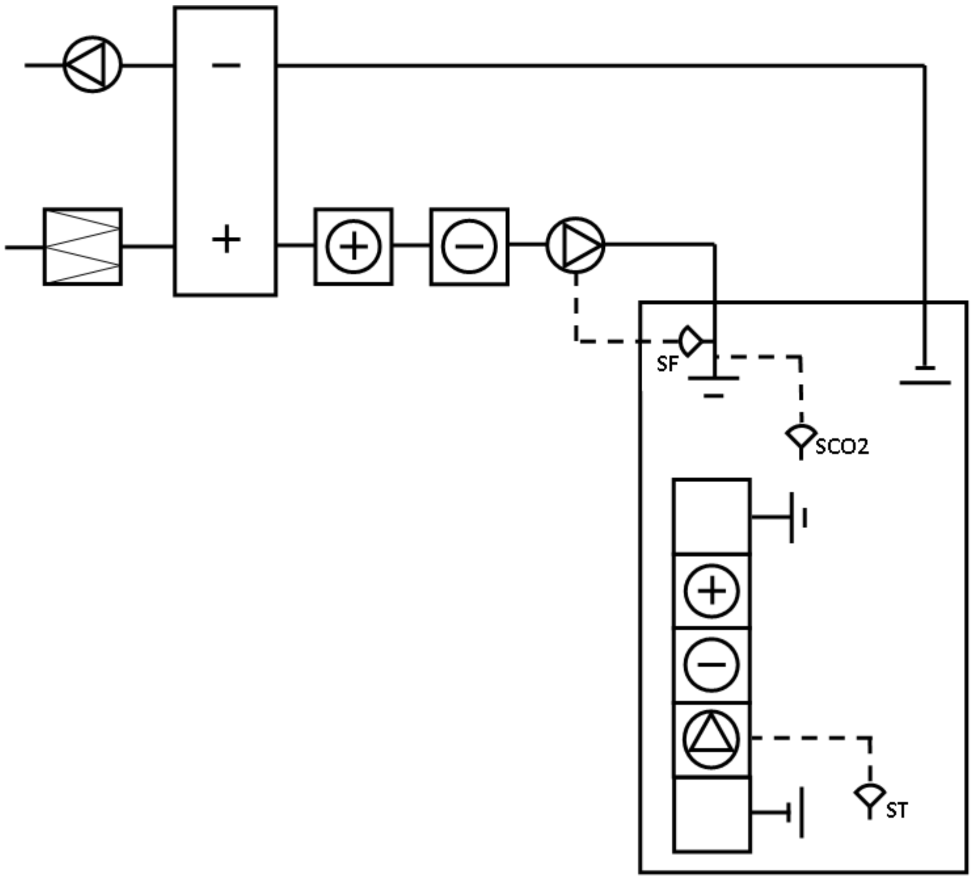


Figure 3.1 Schematic picture of the HVAC reference system

3.1.1 Ventilation system

The IAQ in a zone is represented by the corresponding level of CO_2. Even though CO_2 is not hazardous until in very large concentrations, the approach to use it as an IAQ indicator is common in demand control ventilation applications since CO_2 is a reliable surrogate for bioeffluents from occupancy[72]. In this work, the level is controlled by the ventilation system by the means of variable fresh air supply. The air is transported via ducts to the rooms where the distribution is managed by roof-mounted diffusers with variable opening determined by the current CO_2 concentration within the zone.

Central components
The supply air is conditioned on a central level, before distributed to the rooms. The AHU consists of;

- A heat-recovery unit between the supply and the extract air
- A heating coil

- A cooling coil
- Two flow controlled fans for supply and extract air

The ventilation system is a so called Dedicated Outdoor Air System (DOAS) which means that the entire supply consists of outdoor air[48]. However, a large part of the energy in the exhaust ducts can be recovered in the supply air by the heat-recovery unit which consists of a rotary heat-exchanger. Both the heating and cooling coil are water-to-air heat-exchanger. That is, the primary side consists of a water circuit with hot or cold water and the secondary side consists of the air circuit.

Supply air flow rate levels
The design flow rate of the ventilation system, i.e. the maximum available fresh air supply rate, to a zone was set to correspond to a guideline commonly used in Sweden for office building applications. The guideline consists of the sum of an occupancy dependent term, 7 l/(s·person), and a floor area dependent term, 0.35 $l/(s \cdot m^2)$ [21]. Hence, the design flow rate is dependent on the number of people the room is designed for as well as the size of the room.

In most office building application, the minimum fresh air supply to a zone is limited to a value above zero. The purpose is to account for emissions that does not derive from humans, such as emissions from building material and furnitures, and is normally maintained for low activity periods such as nights and weekends. In this work, the lower value is set to zero which means that if the CO_2-level in a zone is below the set point, the corresponding supply air flow rate will evolve to zero. This approach was chosen to make the work more transparent and is motivated by that the focus is on demand based control from a CO_2 point of view. All recommended values of minimum supply air flow rates in guidelines are rough estimates, since all buildings are different. Hence, they can’t be considered as either demand based or general and would therefore be ungrounded in this work.

3.1.2 Local system for thermal control

In this work, the thermal climate is represented by the room air temperature and, as mentioned, the largest part of the thermal control is handled by room integrated FCUs. A FCU consists of a small fan, a heating and a cooling coil. The coils are designed in the same manner as in the central AHU with water circuits on the primary side. The fan is used to circulate room air through the unit and the coils are used to either heat or cool the air before supplied to the room. Hence, the FCU can both be used to supply cooling or heating power to the room[66].

3.1.3 Properties of the reference system

The choice of reference system is based on that the two supply variables, i.e. fresh air and thermal power, are uncoupled. This means that the reference system has the possibility to maintain the desirable setpoints of both temperature and CO_2 without compromising each other. Hence, the potential of matching the supply and the demand is not limited by the system design and the focus can instead be on the potential of demand based control measures.

An alternative reference system would be an all-air based. However, this type of system was rejected do to the coupling of the controlled variables. One primal control signal on a local scale would limit the possibilities of a simultaneous match between supply and demand for both CO_2 and temperature. Hence, the potential of adding a demand based control system is limited by the HVAC-system.

3.2 Models

A large part of this work has been dedicated to develop the HVAC and building models used in the simulations. The approach was to use modular built systems, i.e. to model single HVAC components and rooms and to connect them into systems. The simulation software Simulink, which is the simulation platform of MatLab by Mathworks, was used.

The HVAC-components can be divided in three different classes;

- Components in the air-systems
- Components in the water-systems
- Interface components between the water- and air-system

The water system refers to the primary side of the heating and cooling coils and contains relatively few components in this work. The modelling work has instead been focused on the air system and the interface components. The air system refers to components for transportation and distribution of air while the interface components are used for heating or cooling; either on a central level in the ventilation system and on a local level in the FCUs.

In the text below, the focus is on the main features of the models and the parameter tuning procedure. In appendix G, the governing mathematical equations of the models are presented along with;

- The required input variables
- The required model parameters
- Values of model parameters that has been treated as constants throughout the work
- The resulting outputs

The original form of the equations that were not developed in this work can be found in the corresponding citations.

3.2.1 Room models

Thermal model

The thermal room model used in this work consists of a system of differential equations presented as equation G.1 in appendix G. The model was developed during this work although it is inspired by van Paassen[74]. The equation system requires a large number of model parameters and different sets of values have also been used to generate different simulation setups. Due to the comprehensiveness of the room model, section 3.4.1 is entirely dedicated for reviewing the model parameters used in this work. In section 3.4.1.3, the most frequently used parameter sets are validated regarding the static and dynamic behaviour of room air temperature.

The basis of the room model is a dynamical energy balance. The temperature of the room air, which is the main output, is represented by one node. Hence, the model solves the mean air temperature of the room. The temperature of the building structure is calculated in twelve nodes. Each building element, such as walls, roof and floor is represented by two nodes, one on each surface. This approach was chosen to be able to determine the operative temperature in a later stage. This also means that the temperature gradient inside the wall is not determined which on the other hand is not necessary since the only output is the room air temperature.

The total thermal inertia of the building derives from three parts; the room air, the building structure and the furnishing of the room. The parts from the room air and the furnishing are relatively small and the absolutely dominating part derives from the building structure.

CO_2 model
The CO_2 model of the room, presented as equation G.2 in appendix G, is based on a dynamic mass-balance [22]. The model assumes perfect mixing of room and ventilation air.

Air flow through door
This model is presented as equation G.3 in appendix G and described the temperature driven flow rate through an open door [27]. The admissible door-openings, h_{door}, used in this work are either 0 or 1, i.e. closed or open.

3.2.2 HVAC-components

The HVAC-component models are focused on pressure-flow relations and on thermal processes. In many components, but not in all, both a thermal and flow-pressure part is solved in parallel.

Flow control
As mentioned in the beginning of this chapter, the components related to the distribution of water and air has been modelled as if the distribution system is flow controlled. This was done both as a simplification and to increase potential match between supply and the demand in the HVAC-system.

This design primary applies to the fan/pumps and branches in the duct-work. In real branches, the inlet stream is divided between the outlets based on the corresponding absolute pressure levels of the outlets. If a higher flow rate is desirable in one of the rooms, the corresponding diffuser is opened which decreases the absolute pressure at the corresponding outlet of the branch. However, to solve the connection between flow and pressure in the distribution system is computational intensive since any change of pressure level in the system also affects the flow distribution in the rest. For larger systems, extremely short simulation step-sizes are required to avoid divergence.

Instead, the pressure dependence on the flow was treated indirect. This was handled by defining the flow levels in the system as inputs from the local controllers in the distribution systems of water and air. In the air case, when a change of supply air flow rate is desirable in a certain room, the respective controller sends a corresponding signal both to the diffuser and the branch. The diffuser opening is changed, but the actual diverted flow in the branch is determined directly by the signal from the controller. Hence, the diffuser is actually not controlling the flow rate but is acting in the same way as if this was the case. By synchronizing the diffuser opening rate and the diverted flow rate, the pressure drop over the diffuser corresponds to as if the diffuser is providing the changed flow rate to the room. Furthermore, the same signal is also send to the central fan, which sums the all signals local diffuser controllers of the rooms. Hence, the central fan supplies what is requested locally.

The water distribution system is managed in the same way but lacks of branches. Instead, the control signal to the control valve is also send to the pump. Hence, the control valve opening is changed but it is actually the signal to the pump that determines the flow rate.

3.2.2.1 Pressure/flow models

All components in the distribution system for air and water ads to a pressure drop. The pressure drop is determined by the flow rate through the component and the maximum pressure drop occurs at the design flow rate. The sizes of the HVAC-components are tuned so that the pressure drop at design flow rate corresponds to commonly used guidelines.

These guidelines are presented for components in the air distribution system in table 3.1[11] below. The design flow rate in the ventilation system is determined by the design hygienic flow rate of the rooms when all diffusers are fully opened. The design flow rate through the FCU on a local level is determined by the design thermal capacity.

Table 3.1 Guidelines for design pressure drops of different HVAC-components associated with the ventilation system

HVAC-component	Pressure drop
Heating coil	40 Pa
Cooling coil	60 Pa
Heat recovery unit	100 Pa
Filter	50 Pa
Grate	25 Pa
Duct	1 Pa/m
Branch and bend	10 Pa
Diffuser	30 Pa[42]
Balancing damper	Dependent on location

The filter and grate components in table 3.1 do not contribute to the function of the HVAC-system but are included since they are common in real HVAC-system. Hence, these are included to make the studies more realistic and, in this work, their purpose is to increase the total pressure drop of the system which in turn increases the electrical usage of the central fan.

The purpose of balancing dampers is to even out the pressure drop between the different branches in the air-system so that the design flow rate is achieved in the entire system when all diffusers are fully open. In this work, the openings of the balancing valves were tuned manually in an iterative way.

In the water distribution system, the design flow rate is determined by the design thermal capacity of the air-coils. The guidelines for the corresponding pressure drop that were used to tune the models are presented in the table below. The pressure drop of the control valve refers to fully opened.

Table 3.2 Guidelines for design pressure drops of different HVAC-components associated with the water system

HVAC-component	Pressure drop
Pipes	100 Pa/m
Control valve	10 kPa

General pressure drop
The pressure drop over some of the components was modelled in a general way using equation G.4 in appendix G. These models require experimentally determined nominal values which for example can be found in manufacturing data.

The nominal values consist of the design pressure drop presented in table 3.1 as well as the design flow rate. The characteristic of the component is determined by the so called turbulence exponent presented in table 3.3[63]. A value of one correspond to pressure drop characteristic for laminar flow and a value of two for pressure drop characteristic for turbulent flow.

Table 3.3 Turbulence exponents of components with general pressure drop models

HVAC-component	Turbulence exponent
Grate	1.5
Filter	1.4
Balancing damper	2.0
Heat recovery unit	1.3

Pipe and duct
The ducts are used for transportation of supply and exhaust air and the pipes to transport hot or cold water to the air-coils. The same model was used both for pipes and ducts and it describes the frictional pressure drop and the transport-delay of the media, i.e. the time it takes for a change at the inlet to be notable at the outlet. The transport-delay is affecting flow rate, temperature and CO_2 concentration as can be seen in equation G.5 in appendix G.

The pressure drop is described by equation G.6. The friction factor consists of two models, one for laminar and one for turbulent flow. The transition occurs at a Re-number of 2600 and both models are based on the Moody correlations[71].

The size of the ducts and pipes is tuned to correspond to the specific pressure drops presented in tables 3.1 and 3.2 above. The total length of duct system is determined by the distance between the AHU and the first room as well as the distances between the rooms. The latter are determined by the sizes of the room. The first-named, i.e. the distance between the AHU and the first room, was tuned so that the total pressure drop of the straight main duct (not including branches and bends) corresponded to 100 Pa at design flow rate.[11]

The length of the pipes is not of importance since the electricity usage of the pumps on the primary side of the air-coils was neglected.

Duct branch
Branches are used in the multi-zone platforms to divert air from the main duct to the rooms. The model is used to determine the pressure drop as well as the flow rate in the main duct after a branch.

The pressure drop is determined by the equation system presented as equation G.7 in appendix G. The pressure loss coefficients are modelled based on the experimental data presented by Idelchik.[33] The size of the branch is determined by the corresponding sizes of the ducts connected to it. This results in a pressure drop contribution of about 25 Pa to the main duct at design flow.

Bends
Bends are used in distribution system to change the direction of the flow. In this work, they are inserted where it is expected that they would be needed in real system. They are adding pressure drop which is modelled using equation G.8 in appendix G. The model allows any angle but only 90° bends are used in this work. The model as well as the values for bend radius is based on the work presented by Sorensen[63]. The diameter of the bend was modelled to fit the ducts and pipes connected to it.

Diffuser and control valves
The control valves and diffusers are actuators with the purpose to vary the flow rate in their respective branch. This is done by varying the opening and there by the flow resistance. As mentioned, in this work, their function is limited to adding pressure drops since the flow-pressure dependence is simplified.

The diffuser and the control valve are based on the same model presented as equation G.9 in appendix G. This model was also used in the work by Sorensen[63] and describes an exponential characteristic. The design pressure drop is based on the guidelines presented in table 3.1 and 3.2. By associating the model parameters denoted as maximum and minimum flow rate to the resulting pressure drops, the authority in equation 3.1 of the actuator can be incorporated in the model. For the diffusers, an authority of 0.3 was used based on manufacturing data[42]. For the control valves, an authority of 0.5 was used according to a guideline presented by R.Petitjean[55].

$$\beta = \frac{\Delta p_{fully\,open}}{\Delta p_{fully\,closed}} \text{ [-]} \qquad \text{(eq. 3.1)}$$

Pump and fan

As mentioned, the fan and pump models are provided with the total flow rate desired in the system as an input. The resulting system pressure drop is also provided as an input in the next time step. These, together with the efficiency, are used to calculate the electricity usage via equation G.10.

The efficiency parameter of the central fans are tuned so that the total specific fan power (SFP) value defined in equation 3.2 has a value of 2 kW/m^3 at design flow rate. This value corresponds to Swedish guidelines for energy efficiency of a ventilations system with both supply and exhaust air.[3] The efficiency of the integrated fans in the FCUs are based on manufacturing data[30].

$$SFP = \frac{\dot{W}}{\dot{V}} \text{ [kW/(m}^3\text{/s)]} \qquad \text{(eq. 3.2)}$$

3.2.2.2 Thermal models

Heat recovery unit

The heat recovery unit is modelled as a rotating heat exchanger that recovers heat from the exhaust air to the supply air. It is not utilized for cooling recovery and is non-hygroscopic.

The thermal part is entirely based on the model in Sorensen [63] which is presented as equation G.11 in appendix G. The parameters of the heat exchanging wheel are based on manufacturing data[29]. The maximum temperature efficiency, defined in equation 3.3 with indexes according to figure 3.2, was set to 0.75.

$$\eta_t = \frac{t_s - t_o}{t_e - t_o} \text{ [-]} \qquad \text{(eq. 3.3)}$$

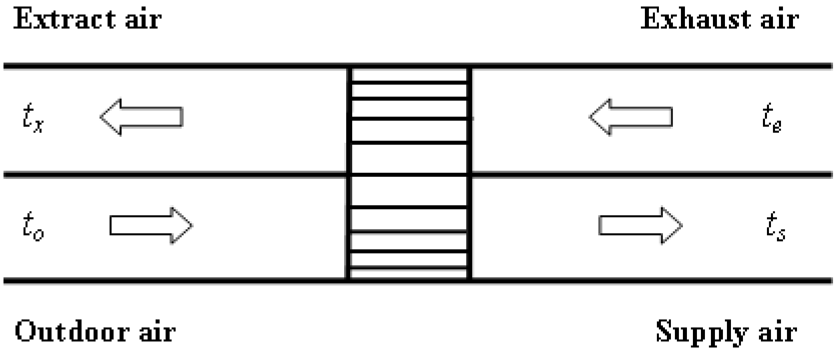


Figure 3.2 Schematic picture of the heat recovery unit

Air-coils for heating and dry cooling
The air-coils are modelled as finned-tube coils with air on the secondary side and water on the primary. Both the air-coils for heating and cooling in the central air conditioning system as well as in the FCU for local control are based on the same model. If they act as coolers or heaters depends on the supply water temperature. Only sensible heat is taken into account by the model which means that the energy usage in the cooler refers to dry cooling without condensation on the air-side.

The thermal model is based on the validated steady-state energy balance over the air-side presented as equation 6.12 in appendix G[63]. The dynamic part, presented as equation G.13 in appendix G, is based on an empirical model presented by M.Anderson[8]. This model is derived by step response tests and includes the time-constants and transport-delays between the different inputs and the output.

The overall heat transfer coefficient in equation G.12 is calculated by equation G.14 presented in appendix G. In this equation, the design overall heat transfer coefficient on the air-side is required. On a central level, this coefficient is tuned to manage the desired temperature lift/drop of the design flow rate with the lowest/highest outdoor air temperatures used in the simulations. For the coils integrated in the FCU, the design overall heat transfer coefficient corresponds to a design power supply of ±500 W in offices and ±1000 W in the meeting rooms.

The pressure drop on the air-side is calculated using equation G.15. The hydraulic diameter parameter of the air-coils are tuned so that the design air velocity through the coils corresponds to 2 m/s[17], and the flow lengths are tuned to achieve the design pressure drops as presented in table 3.1. The rest of the geometry as well as the pressure drop coefficient on the air-side is modelled based on the results presented by Wang et.alt[67]. The model of the friction factor is calculated using a regression model validated for Re-numbers between 200 and 10 000. The resulting pressure drop corresponds to a turbulence exponent of about 1.49.

The pressure drop on the water-side is calculated using equation G.16. The friction coefficient parameter is calculated using one out of two models; one for laminar and one for turbulent flow. Both are presented in the thesis work by Haglund-Stignor[32] and their valid ranges are below and above Re number of 2300.

3.2.2.3 Other models

Sensors
Sensors are modelled with a time-constant and a transport delay using equation G.17. The model parameters are based on manufacturing and experimental data and are presented in table 3.4 in section 3.3.3.

FB controllers
The FB-controllers are modelled with P-, and I-action. The models are based on equation 2.1 and the tuning of the parameter is described in section 3.3.3 below.

3.3 Control system

In this section the FB-control loops are described. The performance of these control systems is compare to the performance of the alternative control systems in the evaluations. In the following work, the alternative control systems are constructed by adding a FF-control part either to the local FB-loops or to one of the central FB-loops. When this is done, the setpoints of the FB-controllers are also changed; to the threshold in the IAQ case and to the temperature dead-band in the thermal case.

As mentioned, the overall control objective is divided into two parts; control of thermal climate regarding room air temperature level and control of IAQ regarding carbon dioxide level. These levels are primary controlled by local room integrated FCUs as well as central supply of fresh air distributed by ducts and air diffusers to the rooms.

3.3.1 Local FB-control loops

A schematic picture of the local FB control loops are presented in figure 3.3. The supply air flow rate is controlled by varying the opening of the air diffuser whilst the power supply of the FCU is managed by varying the rotational speed of the integrated fan. Additionally, an internal control loop is controlling the hot/cold water supply through the air coil to maintain a constant temperature of the circulated air through the FCU. This is managed by a direct connected control valve[66] (see figure 1.1 in section 1.1.2).

As mentioned, the threshold value of the CO_2 concentration was set to 1000 PPM according to Swedish guidelines[21] and the setpoints of the controllers are tuned so that the overshoot does not violate this.

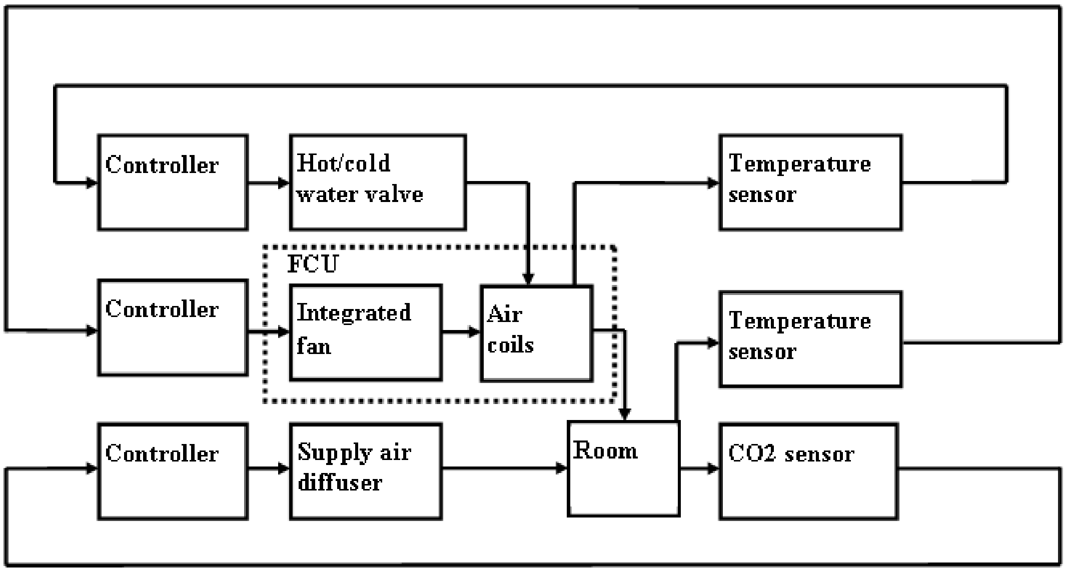


Figure 3.3 Schematic picture of local control loops

The magnitude of the room temperature dead-band is set to one degree, between 21 and 22 °C. This means that the heating mode is activated below 21 °C and that the cooling mode is activated above 22 °C. In between, the temperature control is inactive, thereof the name dead-band.

In contrast to the control objective of the CO_2, there is no corresponding guideline that states an upper and/or lower limit for the room temperature. Instead, the magnitude of the dead-band was set in relation to the size of the disturbances that the building is subjected to during the simulations. The purpose was to simulated conditions when the performance of the control system is challenged and its operation is crucial to maintain the setpoints.

3.3.2 Central FB-control loops

In figure 3.4, the central FB control loops are presented schematically. In most application the central fan is pressure controlled but as a simplification described in section 3.2.2, a flow controlled approach was chosen in this work. That is, the central fan supplies the same amount of air as required by the diffusers. Hence, it is assumed that the desired supply air flow rate is known on a local level and that this information can be distributed to a central level. This assumption is based on the technology provided by the TTD diffusers by LindInvent[42] with integrated flow rate sensors. This solution also contributes by improving the prerequisites for matching supply and demand from an HVAC-system point of view. As discussed in the beginning of this chapter, avoiding underlying limitations in the reference system is desirable to be able to isolate the impact of the control system and thereby make the evaluation more transparent.

The central SAT control system consists of a controller using sequence control of the heat recovery exchanger, heating and cooling coil. The controller prioritizes the HRX and if full load is not enough to reach the SAT setpoint, the heating coil is activated. The cooling coil can only be utilized if both the HRX and the heating coil are inactive.

The water-side configurations of the coils are identical to the corresponding local loop in figure 3.3, i.e. the actuator consists of a direct connected control valve. The HRX is of a rotating type and is controlled by a variable speed motor connected to the wheel.

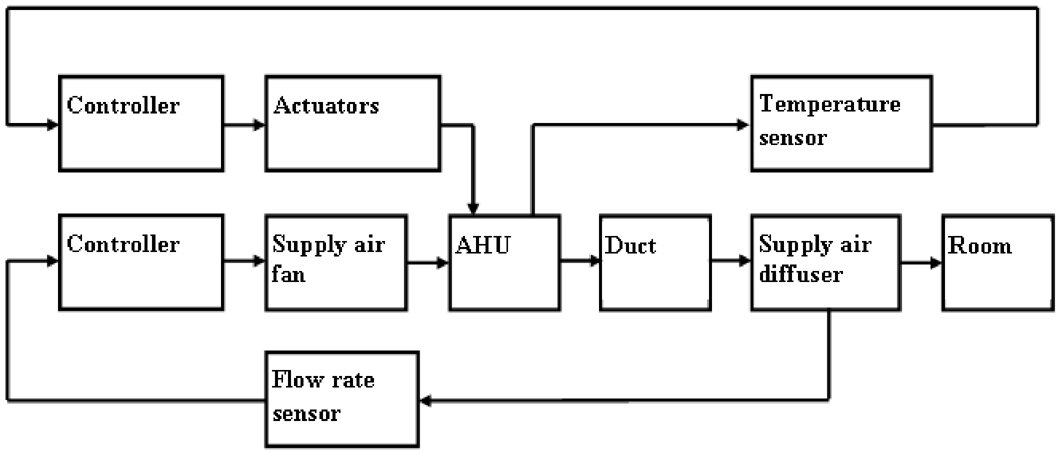


Figure 3.4 Schematic picture of central control loops

Although the ventilation system is designed for hygienic purposes, the temperature of the supply air is allowed between 15 and 18 °C and can hence support the local thermal control in cooling mode.

3.3.3 FB-control system design

The FB-control loops are designed with the highest performance as possible, but not higher than what can be regarded as realistic and implementable. The intention is to make the work as transparent as possible by comparing the FF-control methods with the best possible FB-control system.

All FB-controllers in this work are of PI type. This control design was chosen due do to their commonness in the HVAC sector as well as their documented satisfying performance in the same applications[62]. The main design focus has been on tuning, starting off with the Ziegler-Nichols oscillation method[53], and further improved by hands-on fine adjustments.

Also the sensors used in the FB-loop are modelled with the best practical performance as possible, since their functions are directly affecting the performance of the FB-control. The characteristic data of the sensors are presented in table 3.4. The data consists of time-constants deriving from manufacturer data[19] and experiments[45] as well as time-delays determined in connection to the experiment presented in chapter 10. Further, it is assumed that the sensors are 100 % accurate (the uncertainty is 0 %), i.e. and they measure the exact quantities of the system.

All sensors used in the FB-loop are ideally located. The temperature and CO_2 sensor on room level are modelled as located in the exhaust air duct. Hence, their function is not limited by slow air movements or protective casing that screens the environment in the CO_2 and temperature case. Nor is the function limited by disturbances from the building structure in the temperature case. This approach allows the short time-delays and small time-constants presented in table 3.4.

Table 3.4 Modelled characteristics of sensors used in FB-loops

	Delay-time [s]	Time-constant [s]	Uncertainty [%]
CO_2 sensor	198	120	0
Temperature sensor	198	60	0
Air flow rate sensor	10	3	0

3.3.4 FF-control system aspects

As can be seen in the figure 3.3, the local FB system utilizes measurements of room CO_2 and room temperature to control the actuators. The FF control systems are configured in the same way but as open-loop control. However, the control signals produced by the FF-controllers are describing the magnitude of the supply in terms of physical units. This means that the actuators must be able to transform these signals; from supply air flow rate to diffuser opening in the IAQ case and from thermal power to rotational speed of the integrated fan in the thermal case. In this work, the transformation is handled by the FF controller by incorporating actuator models. But in practise, distributed FB controllers, tuned to follow the output of the FF-controllers as reference values, in combination with the necessary sensors can instead be used.

In section 2.2, figure 2.6 indicated that the FF control system also includes a FB-controller used to compensate for unmeasurable disturbances and/or model simplifications and errors. In this work, the minimum output of these FB-controllers is set to zero. This means that the FB-controller can’t compensate for model errors that result in supplies larger than the actual demand. In practice, the FB signals can of course be smaller. However, the only purpose of negative signals is just to compensate for errors in the FF-part, while positive signals also contribute by supporting the disturbance rejection. Since this work is an evaluation of FF control, the features of the negative signals are unnecessary because they would only make the results regarding the potential of FF-control less transparent.

In practice, a FF-controller can be used to control virtually any type of supply system such as heating, cooling, air as well as its subcomponents such as radiator, air conditioning units, dampers etc. However, the main feature that is focused upon in this work is disturbance rejection and managing heat loads in office buildings. Therefore, the FF-control is limited to perform heat and emission loads contra-actions. This means that the FF part of the control system can’t increase the heat supply and its minimum contribution to the supply air flow rate is zero (i.e. FF cannot counter-act the contribution of the FB-part, only contribute).

3.4 Building

In this section, the building models used in this work are validated and described more in detail. In the evaluations, both single-zone and multi-zone platforms will be used.

The work is focused on office buildings and distinguishes between three different types of office-like room. These are denoted as offices, meeting rooms and corridors and all have a rectangular shape with a height of 2.7 m. The floor areas of the office and meeting rooms are 10 and 18 m^2 and the design occupant densities are 0.1 and 0.5 person/m^2 respectively[3]. In turn, the corridor is not designed for occupancy and has a floor area of 140 m^2.

3.4.1 Single-zone platforms

The single-zone platforms are presented schematically in figure 3.5 and consist either of an office or a meeting room. The purpose of the two single-zone designs is to accomplish two different kinds of load distributions with different relations between internal and external disturbances. This is further discussed later on.

In the office room case, one of the long sides is facing the ambience and 1 m^2 this area is covered by a window. The window is modelled as 2-pane with external solar shading. This corresponds to an overall heat transfer coefficient of 2 W/K and a solar reduction factor of 0.12.[12] In the meeting room case, all walls are facing adjacent rooms as can be seen in figure 3.5.

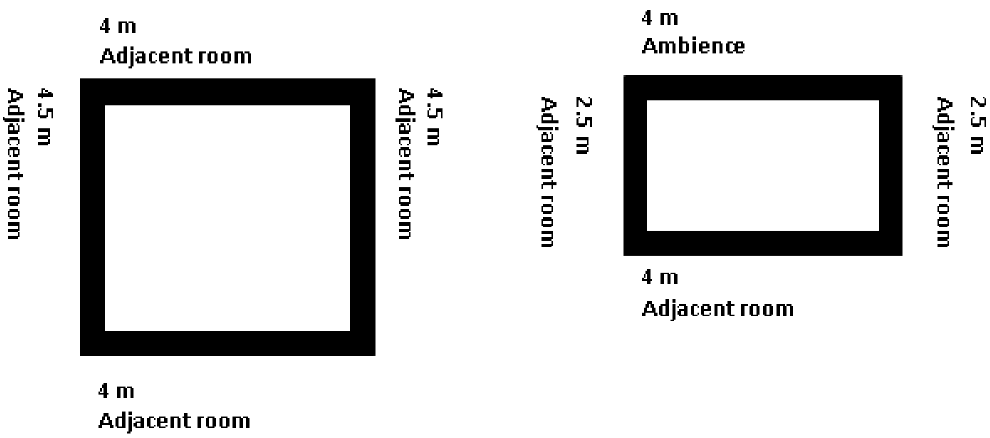


Figure 3.5 Size and alignment of walls in the meeting room and the office room respectively

The interior furnishing of both rooms are modelled by wooden furnitures with a mass of 20 kg/m^2. It was assumed that 10 % of this mass is thermally active.[63] Both rooms are also modelled with a door on the interior long-side walls. The models makes no difference between the thermally properties of a door or wall; the door is included to model the transferred air through the opening. The area of the door is set to 1.25 m^2 and the height to 2.5 m^2.

Thermal boundary conditions
In the single-zone simulations, the thermal boundary condition for walls facing adjacent rooms was set to isothermal. This can be interpreted as adjacent rooms subjected to small disturbances, which means that the corresponding control systems can maintain constant room temperatures. For roof and walls, the boundary conditions were set to adiabatic which can be interpreted as symmetrical building floors. Hence, what happens on one floor also happens on the floor below as well as above.[70]

3.4.1.1 Building structure density

The density of the building structure is included as a parameter in this work. In most cases, this is managed by distinguishing between heavy and light building structures. However, in the final study, also a medium kind of structure is included.

The heavy and the light buildings are designed as extremities; the heavy building is entirely based on concrete and the light is primarily based on gypsum and wood. The architectural relevance of these structures is not considered as the primary focus at this point. The purpose of including these structures is rather to determine the influence of building density on the potential of FF control. However, the medium building is included amongst the final results as a more relevant structure regarding common building design.

In the following tables the design of the three building structures are presented. The numbering of the layers starts with the layer closest to the room air and is increased further away. The thermal conductivity of the floor and roof are zero due to adiabatic boundary conditions.

The building designs are primarily based on the work performed by P-E Nilsson[50] but some inspiration comes from manufacturer's data[4]. The material data is collected from different handbooks such as[20, 54].

Heavy building
In the table below the design of the heavy building structure is presented. The walls facing adjacent rooms as well as the floor and roof are designed as single layer structures of concrete. The exterior wall is a three layer construction with an inner layer identical to the roof/floor and adjacent wall single layer constructions. This first concrete layer is followed by a layer of mineral wool and finally a layer of bricks.

Table 3.5 Design of heavy building structure

Building element	Layer	Material	d [m]	λ [W/(m·K)]	ρ [kg/m^3]	c_p [J/(kg·K)]
Interior wall	First	Concrete	0.15	1.5	2300	880
Exterior wall	First	Concrete	0.15	1.5	2300	880
	Second	Mineral wool	0.2	0.04	20	750
	Third	Brick	0.15	0.12	1500	800
Roof/floor	First	Concrete	0.15	0	2300	880

Light building
The corresponding design of the light structure building is presented in table 3.6. For all surfaces, the layer closest to the room consists of gypsum followed by a layer of mineral wool. The interior walls are symmetrical and the outer layer of the exterior walls consists of a thin sheet of aluminium. For the roof and floor the outer layer consists of wood.

Table 3.6 Design of light building structure

Building element	Layer	Material	d [m]	λ [W/(m·K)]	ρ [kg/m^3]	c_p [J/(kg·K)]
Interior wall	First	Gypsum	0.013	0.22	970	1090
	Second	Mineral wool	0.07	0.04	20	750
	Third	Gypsum	0.013	0.22	970	1090
Exterior wall	First	Gypsum	0.013	0.22	970	1090
	Second	Mineral wool	0.2	0.04	20	750
	Third	Metal sheet	0.002	272	2700	890
Roof/floor	First	Gypsum	0.013	0	970	1090
	Second	Mineral wool	0.07	0	20	750
	Third	Wood	0.044	0	500	2300

Medium building
The medium structure building is designed as a building with heavy bearing elements and light elements for the climate envelope. The light elements such as the wall structures are identical to the corresponding ones of the light structure

building presented in table 3.6. Also the heavy load bearing elements, such as floor and roof, in the medium structure are identical to the corresponding ones in the heavy building design. The parameters for the medium structure building design are presented in table 3.7.

Table 3.7 Design of medium building structure

Building element	Layer	Material	d [m]	λ [W/(m·K)]	ρ [kg/m^3]	c_p [J/(kg·K)]
Interior wall	First	Gypsum	0.026	0.22	970	1090
	Second	Gypsum	0.026	0.22	970	1090
Exterior wall	First	Gypsum	0.013	0.22	970	1090
	Second	Mineral wool	0.2	0.04	20	750
	Third	Metal sheet	0.002	272	2700	890
Roof/floor	First	Concrete	0.15	0	2300	880

3.4.1.2 Validation

Since the disturbances acting on a building is intermittent and the time-constant is large, the temperature of the building never reaches steady-state. This means that there will always be a temperature gradient in the building elements. If the disturbances are considered as periodical, the building elements will switch between cooling and heating the room air. That is, when the disturbances are large, the elements are heated and when the disturbances are low the elements submits the heat.

Validated parameters
An important design parameter in this context is the so called penetration depth, which refers to how deep a heat-wave is transmitted into the structure during the heating mode until a mode switch occurs. The penetration depth depends on the length of the period of the disturbance as well as the material properties of the building element[31].

Another important design parameter is the heat transfer coefficient for convection, denoted as α. This parameter determines the heat flux between the air and the building structure for a certain temperature difference. As a model simplification, the α-value is most of the time considered as a constant, even though it is dependent on the advection of the air. Hence, a constant α-value should represent many different cases with a satisfying accuracy which requires a carefully chosen value.

Both the penetration depth and the α-value used in the building models were tuned during a validation process in which the simulation software IDA ICE was used as a reference. IDA ICE is a software for energy and indoor climate simulations and has been validated in IEA Task 22 and BESTEST[64]. Hence, IDA ICE is considered as a relevant reference for validation.

In the validation process, the parameters for the heavy and light building structure were individually tuned. This was done for two different disturbances; both with an amplitude of 280 W but with different period lengths. The parameter values were adjusted to fit the IDA ICE result for both type of disturbances as far as possible. The penetration depth was tuned by fitting the dynamic response and the α-value by fitting the steady-state value of the temperature of the room air.

Heavy structure building

The room temperature responses when the disturbances are applied to the heavy structure building are illustrated in figure 3.5 and 3.6. The disturbance corresponding to figure 3.5 is a step, switching from 0 to 280 W instantaneously. In the same manner, the disturbance related to figure 3.6 is shifting between 0 and 280 W every 12th hour. Notice that the scales of the axes differ.

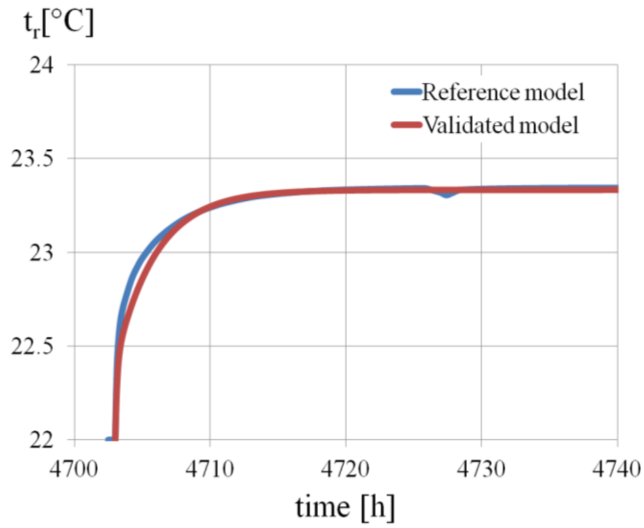


Figure 3.5 Step response of heavy building structure

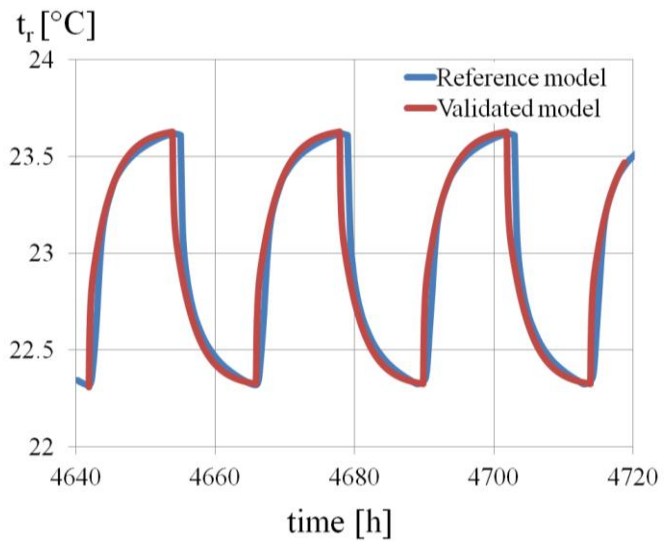


Figure 3.6 Pulse response of heavy building structure

Light structure building
The room temperature responses when the disturbances are applied to the light structure building are illustrated in figure 3.7 and 3.8. The disturbance corresponding to figure 3.7 is a step, switching from 0 to 280 W instantaneously. In the same manner, the disturbance related to figure 3.8 is shifting between 0 and 280 W every 12th hour. Notice that the scales of the axes differ.

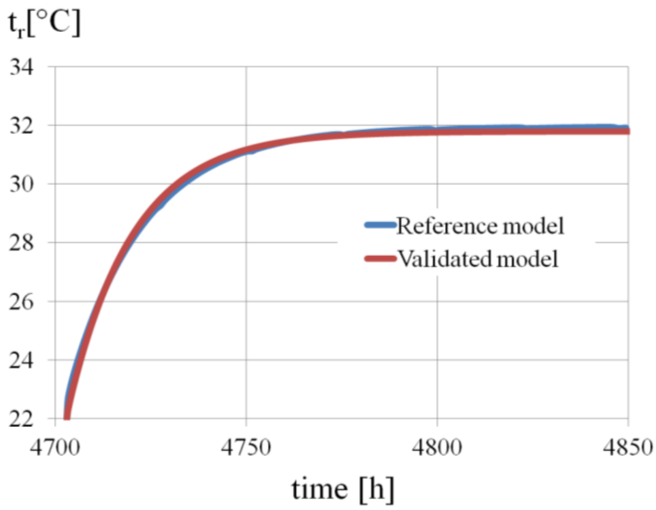


Figure 3.7 Step response of light building structure

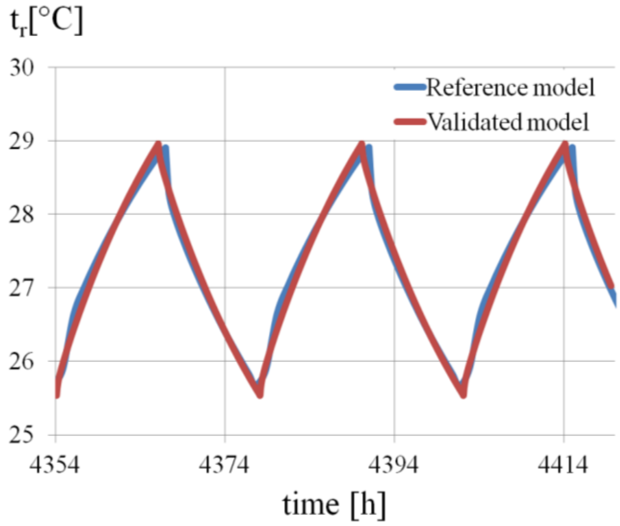


Figure 3.8 Pulse response of light building structure

Medium structure building
The medium building was not directly validated. However, it consists of a combination of elements found in the light and heavy structures which in turn is validated. By using the tuned parameters in the medium structure it is also considered as corresponding to an equivalent IDA ICE model.

3.4.2 Multi-zone platform

The multi-zone platform is constructed by thermally connect the single-zone platforms presented above. Figure 3.9 presents the layout of rooms and duct work used in the multi-zone platform. Additional components in the air distribution system compared to the single-zone system are bends, branches and balancing dampers.

The system is intended to represent part of an office building and by that bring the work closer to reality compared to single-zones platforms. It consists of nine office rooms, one meeting room and one corridor thermally connected to each other. As can be seen in the figure, the layout of the rooms prevents the use of thermal boundary conditions of the adjacent walls. However, the roof and floor are still described as adiabatic.

The building is located in an east-west direction and four of the offices are facing north (N) and five are facing south (S). The meeting room is embedded in the building and lacking exterior walls.

The corridor is not directly thermally controller and lacks of distinguished ventilation air supply. The supplies of fresh air and thermal power are instead handled indirectly by transferred air between the corridor and the rooms through open doors.

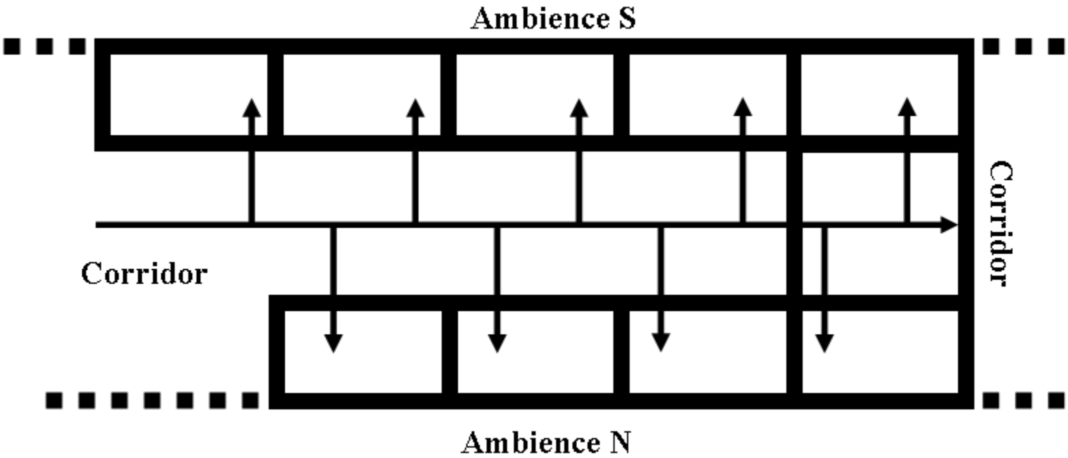


Figure 3.9 Configuration of multi-zone platform including duct work

4 Feed-forward control design

As mentioned, FF-controllers are primarily based on two parts; the FF-sensor and the FF-filter. The purpose of the sensor is to detect and/or measure a certain disturbance and to return a relevant and useful signal to the filter. The purpose of the filter is to process this signal into a control signal that directly can be sent to the actuator in question. Alternatively, the output of the filter can consists of a setpoint, of for example supply air flow rate, which in turn is sent and managed by a FB-controller connected to the corresponding actuator.

A large part of this work is dedicated to determine the most favourable FF-controllers for office building applications. The FF-sensors are included on the basis of existing measurement technologies. That is, the sensor technologies are more or less given. The main part of the FF design work aims instead to evaluate the potential of a certain FF-sensor output and to find the best way to process this signal in the FF-filter. This was performed in a design selection process presented in chapter 5. In that study, the best FF-filter for a given FF-sensor was identified by comparing the performance of different combinations. In the following chapters, the selected FF-control designs are evaluated; both for local and central control applications.

4.1 Disturbances characterization

The disturbances considered in this work are seen as either external or internal and are measured using the FF-sensors indicated in figure 4.1. The figure also indicates the connection between the controlled variables and the disturbances by presenting the final destination of the sensor output on a local level.

The disturbance characterization used in this work is a bit different from the conventional way. Here, an external disturbance is characterized by that there exists other than control related measures to reduce the impact on the controlled variable before the disturbances enter the building. As mentioned in the beginning of chapter 3, such measures are used to a large extent in this work; the impact of ambient climate is reduced by a tight and well insulated building envelope and the impact of solar radiation is reduced by external shading. The internal disturbances consist of occupancy, lighting, equipment and opening of the door.

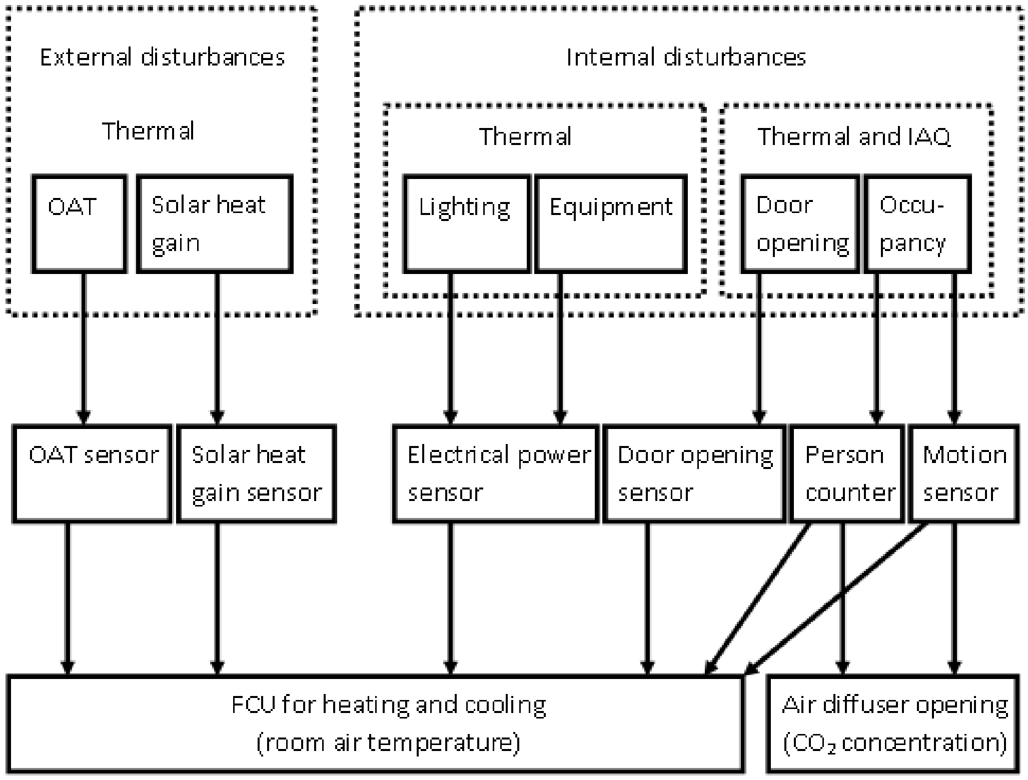


Figure 4.1 Disturbance categorization

Another common disturbance is the infiltration (or exfiltration) rate of outdoor air to the room. The rate is temperature and/or wind driven which means that a corresponding FF-sensor needs to measure the wind speed and the direction as well as the temperature difference between the room and the ambience. The infiltration rate is also dependent on the tightness of the building which is quite troublesome to determine. It is therefore questionable to assume that the infiltration rate is known and can for that reason not be included as an input to the FF-control system. To increase the transparency of the results, it was neither included as a disturbance in the simulations. However, the impact of not taking the infiltration rate into account is evaluated in the sensitivity analysis presented in chapter 10.

4.1.1 Impact of disturbances

As shown in figure 4.1, the IAQ is mainly affected by persons and in some sense also due to opening/closing of a door, depending on the state of the air on the other side.

On the other hand, all disturbances in this work have an impact on the thermal climate by inducing an uncontrolled increase or decrease of heat supply. Depending on the source, these disturbances affect the room differently dependent the heat transfer relation between convection and radiation. The part transferred by radiation is absorbed by the building structure and the room air is heated indirectly by the increased temperature of the surfaces. Due to the large thermal inertia of the building, the impact of the radiation on the room temperature is dampened. On the contrary, the convection part is directly transferred to the room air. Hence, in this case, it is the building structure that instead is heated indirectly by the increased air temperature.

In table 4.1[9], the convection/radiation relations used in this work are presented. For example, the solar radiation is entirely absorbed by the surfaces of the room while the main part of the heat supply from equipments is transferred directly to the room air.

Table 4.1 Part of heat transferred to the room by convection and radiation respectively for different disturbances

	Convection part [-]	Radiation part [-]
Lighting	0.33	0.67
Equipment	0.85	0.15
Occupancy	0.48	0.52
Door opening	1	0
OAT	1	0
Solar heat gain	0	1

4.2 FF-sensor technology

As mentioned, the FF-sensors are based on existing technologies and are more or less given. In the main part of the work, all FF sensors are treated as ideal; the accuracy is 100 % and the measurements are not delayed in any sense. However, chapter 10 contain a sensitivity analysis in which the effect of measurement uncertainty related to the selected FF-controller is evaluated.

4.2.1 Electrical power disturbances

Lighting in table 4.1 refers to office related illumination while equipment refers to computers and/or office related machinery such as beamers and over-head machines. Both of these disturbances are characterized as direct power supply to the room and both are measured by power sensors. This means that the output from the FF-sensor is proportional to the magnitude of the disturbance.

4.2.2 Occupancy

Two FF-sensors have been used to measure occupancy; person counters and movement sensors.

Motion sensor
The movement sensor output is limited to the values 1 and 0 which indicate if a room is occupied or not. Hence, the output is dependent on the magnitude of the

disturbance since the output is the same whether one or more persons are occupying the room.

Person counter
The output of a person counter corresponds to the number of people inside a room. Hence, the CO_2 emission rate and heat emission per capita, i.e. the activity level, is left as model parameters and the effect of this approach is also studied closer in the sensitivity analysis in chapter 10.

The person counter technology is assumed to be based on the response of a CO_2 sensor. This method has been used in a number of references in this work. For example, in the work performed by S.A.Mumma a CO_2 sensor and a dynamic mass-balance of the room is used[49]. A similar approach is also used in two separate articles by X.Xu[72] and X.B.Yang[73]. The results show that the prediction error of the number of persons can be limited to about 9 %. In a journal article by T.M.Lawrence, a black-box model is used instead and experiments shows that the estimation error is limited to 4-15%[40].

However, none of these references review the how long it takes from that the number of persons inside the room has changed until an accurate estimate can be determined. This is a crucial aspect of FF-control since the delay will affect the disturbance rejection features of the control system. To determine this, an experimental study has been conducted which is presented in connection to the sensitise analysis in chapter 10.

Implementation
As mentioned in section 3.4.1, the office rooms are designed for one person. Hence, each time a motion sensor response is indicated, also the number of people entering the room is known. This means that the output of a motion sensor and a person counter returns the same information in this case.

In the meeting rooms, on the other hand, a different approach is required. In this work two methods are proposed;

- Utilizing the motion sensor output even though the signal is independent on the number of people inside the room. Three novel methods are proposed further in this work.
- Combining the motion sensor with a person counter based on a CO_2 sensor response. The motion sensor is then used to indicate a disturbance and the CO_2 sensor is used to determine the size.

4.2.3 Door opening

The door opening sensor returns if the door is opened or closed as well as the temperature and CO_2 concentration difference over the door. The actual temperature driven flow rate between the rooms is left for the FF-model to handle.

4.2.4 OAT and solar heat gain

The OAT is measured by a simple temperature sensor and the solar radiation by a power sensor measuring the solar heat gain to the room.

4.3 FF-filter designs

As indicated, the main differentiation between disturbances in this work is not done based on the type of disturbance but with respect to the disturbance measuring method. This means that a certain FF-controller is not designed for a certain disturbance but for a certain FF-sensor or a combination of them. That is, disturbances which can be measured in the same way are also treated as a group. This differentiation is distinct for heat emitted by equipment and lighting. Even though these two disturbances affect the room differently, they are always treated as a group since they are measured in the same way, i.e. with a power sensor.

A certain FF-filter design is characterized by how the FF-sensor output is processed. In this section, a number of novel FF-filter designs for thermal and IAQ control are proposed.

4.3.1 Background

Most FF-filter designs found in literature are based on linear transfer functions between disturbance and FF control signal.[41, 62, 68] Such transfer functions can either be derived by physical modelling or by an identification process. The outcome from the latter example is a black-box model. A black-box model can be determined relatively easy, but is on the other hand purely mathematical. Hence, it lacks of physical correspondence which means that the model can’t be transferred from one system to another by parameter adjustments. Also, the validity of a black-box model is dependent on how it was produced. For example, a step response can be used to identify some basic control related characteristics such as delay time, static gain and dominant time constants. However, such model is essentially only valid for step-shaped input signals which mean that the errors can be large in many practical applications[43].

A general problem related to linear FF-filters is that the processes most often are non-linear. This is also the case for buildings with mechanical ventilation if the supply air flow rate is variable. In equation G.1 and G.2 in appendix G, the non-linear behaviour is indicated by that the supply air flow rate is multiplied with the room temperature and room concentration respectively. In a modelling process, this problem is normally solved by choosing an equilibrium-point for which a linear model is derived. However, the correspondence between the process and the model will decrease as the process deviates from this point.

In figure 4.2 and 4.3, the output of the room model used in this work and a corresponding linearized model is compared. Figure 4.2 presents the room CO_2 concentration and figure 4.3 the room air temperature. Both are presented as a function of the supply air flow rate when all other inputs are held constant.

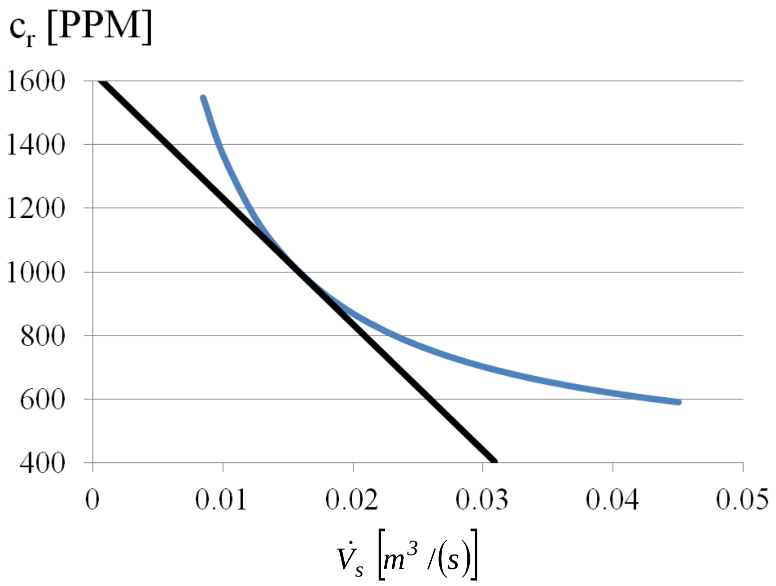


Figure 4.2 Room CO_2 concentration as a function of the supply air flow rate

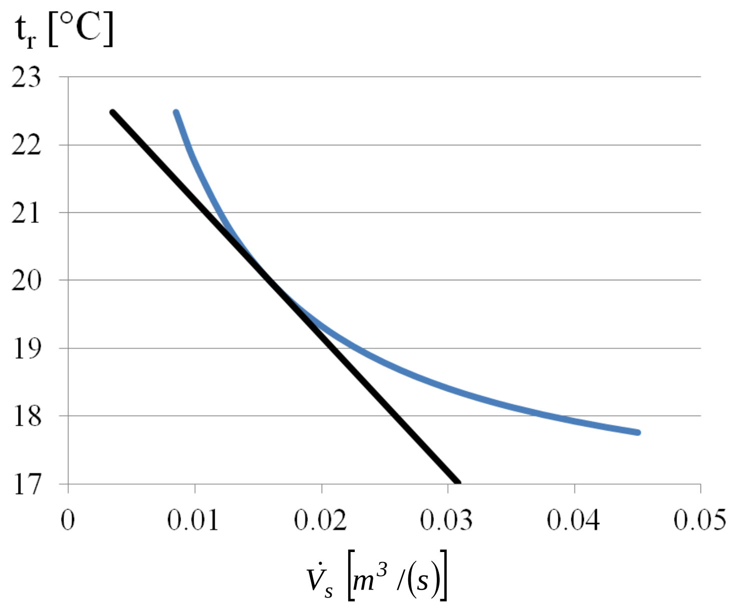


Figure 4.3 Room air temperature as a function of the supply air flow rate

As can be seen, the accuracy of the linear model is only acceptable in a narrow area of operation. Already for small deviations from the equilibrium-point, the magnitude of the estimation error can be a couple of degrees or a couple of 100 PPM. For that reason, FF-filters based on linear models were rejected in this work and the aim was instead to develop more accurate FF-filter designs.

4.3.2 Signal processing methods

In the following text the proposed FF-filter designs are presented. These designs are divided in three groups;

- Filters for thermal disturbances
- Filters for IAQ affecting disturbances
- Filters used to process the output of a motion sensor

The proposed signal processing methods in the filters for IAQ and thermal disturbances are similar while the methods for the motion sensor outputs are completely unrelated to the others.

All filters in this work are designed to manage the combined load of all disturbances related to the specific filter-group. Hence, two filters are required by a complete FF-controller; one for thermal disturbances and one for IAQ affecting disturbances.

4.3.2.1 FF-filters for thermal control

Most thermal disturbances included in this work are associated with heating loads. This means that the most important control actions related to disturbance rejection are associated by a fast increase of cooling supply or a fast decrease of heating supply. For that reason, a delimitation of this work was to limit the FF-filters outputs to control signals associated with an increased cooling supply. This signal is in turn send to the FCU. If the FCU is in heating mode, the heating supply is decreased by the corresponding amount and if the FCU is in cooling mode the supply is instead increased.

<u>The FF-filters for thermal control are not limited to perform control actions outside the temperature dead-band like the FB-controllers</u>. On the contrary, one of their main features is to start compensate for disturbance inside the dead-band to avoid exceeding the setpoints. However, there is no point of rejecting the disturbances entirely if the room temperature is located inside the dead-band. Then the aim of the FF-controllers is to apply control actions which eventually lead up to the upper setpoint. Hence, <u>in this work, disturbance rejection from a thermal point of view refers to limiting the breaching of the temperature dead-band bounded by the heater and cooler setpoints.</u>

In total, all FF-sensor outputs related to thermal disturbances are evaluated for four different types of FF-filters. These filters are divided in two groups depending on if the filter is characterised as so called model-based or so called parameter-based. Each group contains two filters; the model-based are denoted dynamic and static, and parameter-based as direct dynamic and direct.

4.3.2.1.1 Graphical illustration

The features of the four filters for thermal control are presented graphically in the section below. In figure 4.4, the features of the model-based are presented and in figure 4.5 the features of the parameter-based.

Both figures illustrate a scenario, in which a room is subjected to a step-shaped thermal disturbance. The magnitude of the disturbance is measured by a FF-sensor and the figure illustrates how the output of the different filters looks like. In the scenario, the output of the filter is blocked. This means that the response of the room presented in the figures is referring to the uncontrolled evolution.

The purpose of the figures is to present the fundamental features of the filters. To simplify the comparison between the filters, this is done in general terms by avoiding physical units in the illustration. To emphasise this, the response of the room air temperature (t_r) is described as the deviation of the setpoint ($t_{r,sp}$) on the y-axis. Hence, the setpoint is represented by the value zero. Also the filter outputs and the magnitude of the disturbance are presented in general terms

4.3.2.1.2 Model-based filters

A model-based FF-filter is defined as a filter containing an energy balance of the room. These filters are designed to control the room towards an assigned temperature setpoint and the energy balance is used to calculate the required cooling supply. Hence, the model-based filters are essentially basing their control on an estimation of the room temperature response to a disturbance.

The model-based filters can be compared to the filter design in equation 4.1 below, which is the same as equation 2.2 presented in section 2.2. The term G_u^{-1} corresponds to the energy balance of the room while the term $-G_v$ corresponds to a model of the FCU used to transform the estimated cooling supply into an increased speed of the integrated fan.

$$F_{FF} = -G_v \cdot G_u^{-1} \qquad \text{(eq. 4.1)}$$

Figure 4.4 illustrates the responses of the two model-based filters in the scenario discussed above. For transparency, the output of the filter is presented as the estimated room temperature response (t_{est}) which essentially is an intermediated step to determine the required cooling supply. To emphasize that the filter output actually corresponds to a cooling supply, the estimated room temperature is presented in negative quantities.

In the scenario illustrated in figure 4.4, the room temperature corresponds initially to the setpoint, i.e. the control error is zero. However, after 20 minutes, the disturbance of quantity 2 enters the room and the room temperature starts to deviate from the setpoint. At time 3 h, the temperature has reached steady-state again which corresponds to a control error of 1.

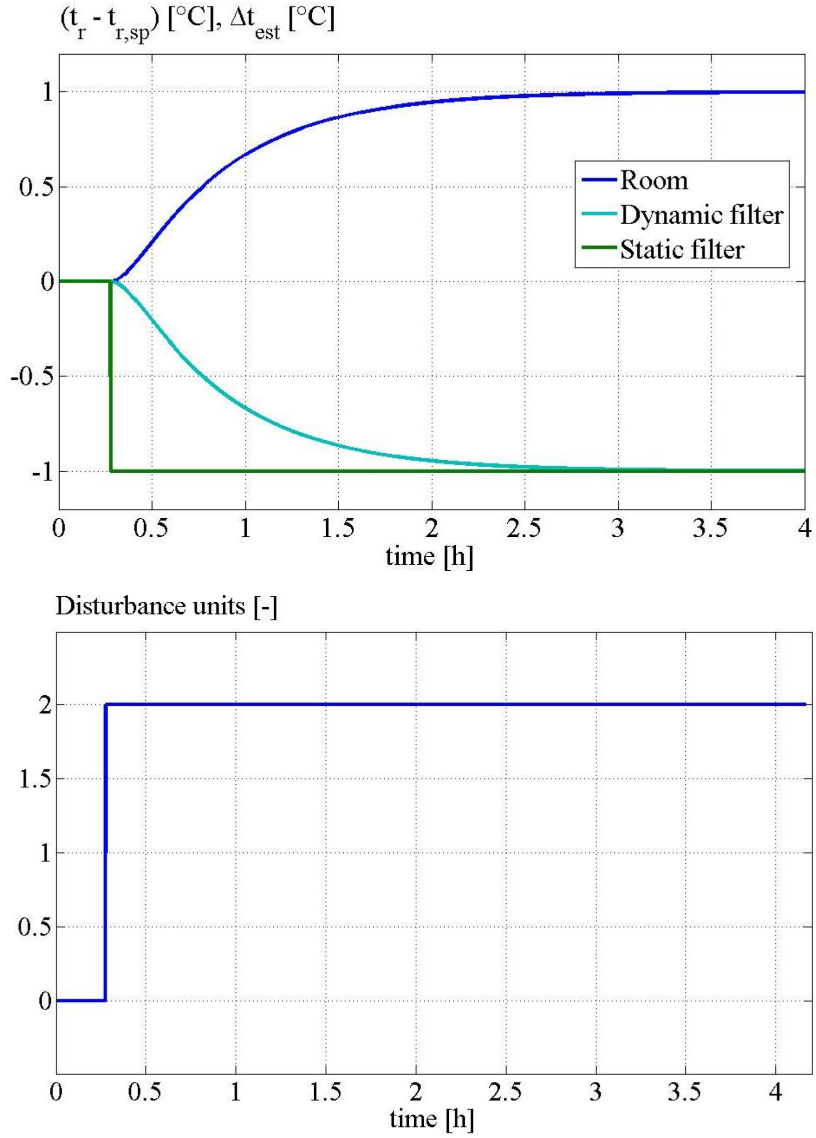


Figure 4.4 Response of uncontrolled room as well as dynamic and static model-based FF-filters (upper figure) to a step-shaped thermal disturbance (lower figure). The units are presented in general terms

The dynamic filter
The room temperature estimation of the dynamic filter matches the system perfectly both regarding dynamic and static behaviour. This is visible in figure 4.4, by that the output of the filter is a perfect inverse mirroring of the room response to the disturbance. As can be seen, the output is time dependent and reflects the behaviour of the room in each time-step.

When the dynamic filter output is represented as a cooling supply signal, it corresponds to the exact amount which is needed to maintain the setpoint at a certain time-step. This means that the dynamic filter is as close to ideal as possible.

The main disadvantage of the dynamic filter is that the disturbance duration is assumed as infinite which might result in an aggravated supply of energy since the disturbances actually are intermittent. Imagine that the room temperature is one degree away from the upper limit of the dead-band at the time when a thermal disturbance is detected. Also imagine that the magnitude of the disturbance is moderate, its duration is known to be 1 h and that the building structure is heavy. Then, due to the thermal inertia of the building, the total heating energy submitted by the disturbance is too small to reach the upper boundary of the dead-band in one hour. Hence, the best strategy would be to do nothing. But when the dynamic filter is used, the FF-controller starts to cool instantaneously, since if the disturbance duration was infinite, the upper boundary of the dead-band would eventually be reached.

The static filter
The difference between the dynamic and static filter was briefly touched upon in section 2.2. The static filter is a simplification which lacks of the dynamic properties of the process. This is indicated in figure 4.4 by that the output of the filter instantaneously attains the value that corresponds to the steady-state conditions of the room.
A static interpretation of a dynamic system can in many cases result in an acceptable control. However, the fruitfulness of this simplification depends on the magnitude of the system dynamic. Supposedly, the aggravation of the control system becomes unacceptable in systems with large time-constants like buildings.

Delimitation
One characteristic feature of the model-based filters is that the energy balance requires a lot of input data and model parameters. All energy streams, input and output, as well as information about building materials, structure and interior mass should preferable be included. This also means that the model-based filter takes the cooling effect of the ventilation system, temperature of adjacent rooms as well as the convection/radiation ratio of the disturbances into account.

In their current form, i.e. based on physical models, the models-based filters are complex and hard to derive. For that reason, they should be regarded as theoretical designs. They are primary included to show the theoretical potential of thermal FF-control but also to be compared to the much simpler and more implementable parameter-based filters discussed further on. However, a design based on black-box models would be much easier to derive and according to M.Soleimani-Mohseni, equal performance can also be expected[62].

4.3.2.1.3 Parameter-based filters

Instead of controlling the room towards a temperature setpoint and by that managing the entire control procedure like the model-based filter, the purpose of the parameter-based filters is to complement the FB-controller by increasing the speed and disturbance rejection properties of the control system.

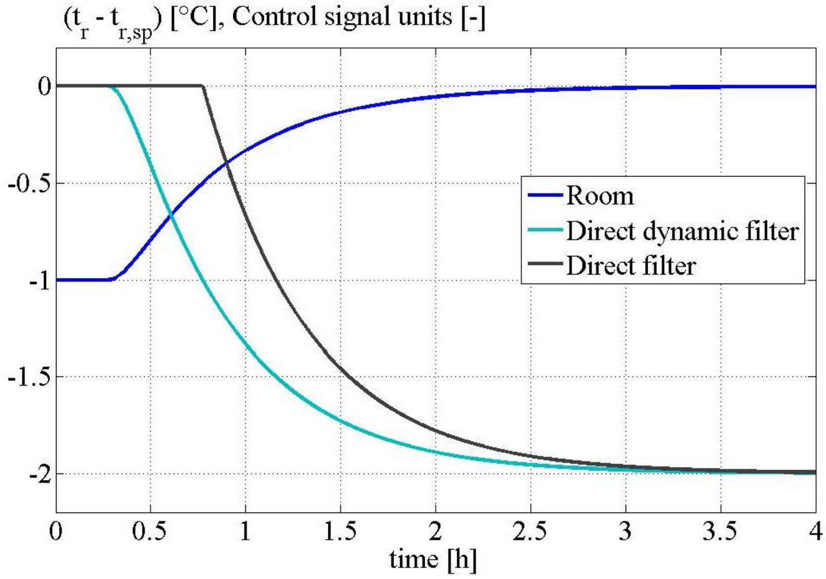
The parameter-based filters are not equipped with knowledge of the room like the model-based filters. The filter outputs are instead generated by directly modifying the output of the FF-sensor. These filters are not as accurate as the model-based but are included to represent an implementable alternative since they require relatively few input signals and model parameters.

The two parameter-based filters included in this work are very similar. Both are based on a scaling function which is limited to values between one and zero and is presented as equation 4.2 below. As can be seen, the filter utilizes the output of the FF-sensor (u_{filter}) to produce the output (y_{filter}) signal. By multiplying with -1, the measured thermal disturbance is transformed into a counter-acting cooling power signal. However, the output of the FF-sensor is also scaled by the filter; by how much is dependent on the current distance between the room temperature and the temperature setpoint ($t_{r,sp}$). This means that the counter-acting FF-control signal will be large if the room temperature is close to the upper boundary of the dead-band. That is, when the disturbance rejection features are important and the control system have to act massively to avoid breaching the setpoint. Further away, the response of the control system is more dampened.

$$y_{filter} = -\left(\left. \left(\frac{t_r - t_{r,act}}{t_{r,sp} - t_{r,act}} \right) \right|_{0,1} \cdot u_{filter} \right) \qquad \text{(eq. 4.2)}$$

As mentioned previously, there is no point of performing a total rejection of a thermal disturbance if the room temperature is inside the dead-band. In the model-based filter cases, this was handled by calculating the supply required to end up at the setpoint. In the parameter-based case, this is managed by the scaling function. The scaling properties of the filter is set by the design parameter denoted as the activation temperature ($t_{r,act}$). The activation temperature is defined as the maximum allowed distance between room and setpoint for the filter to contribute to the control. When the room is below the activation temperature, the input to the filter is multiplied by zero. Between the activation temperature and the setpoint, the input is scaled linearly by multiplying with values larger than zero and smaller than one. Above the setpoint, the entire output of the FF-sensor is utilized as cooling power.

Figure 4.5 illustrates the responses of the two parameter-based filters in the scenario discussed in the beginning of this section. That is, a room is subjected to a thermal disturbance but the output of the filters was blocked before it reaches the FCU. In this case, the output of the filters consists of the scaled output of the FF-sensor which in turn consists of the measure disturbance of magnitude two. Since the value of the scaling function depends on the distance between the room temperature and the setpoint, it is in this case assumed that the room temperature initially is located one degree below the upper boundary of the dead-band. Since the disturbance affects the room with one degree, the new steady-state room temperature after 3 h corresponds exactly to the setpoint.



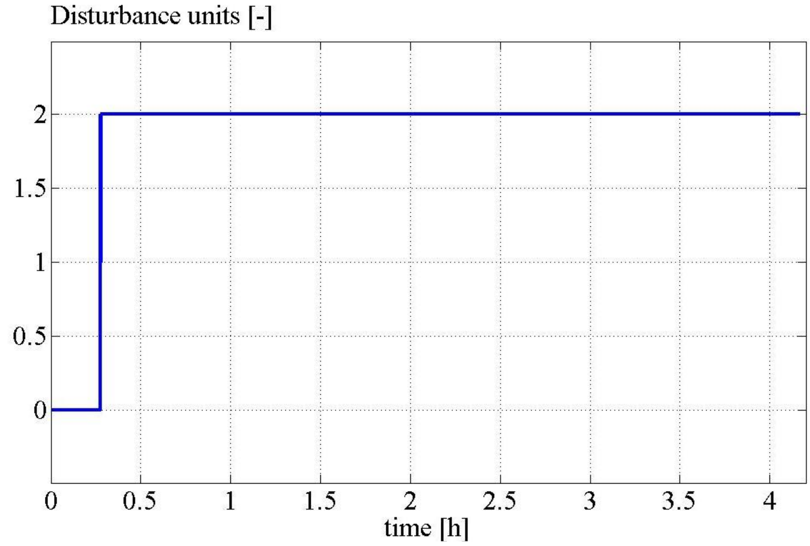


Figure 4.5 Response of uncontrolled room as well as direct and direct dynamic static parameter-based FF-filters (upper figure) to a step-shaped thermal disturbance (lower figure). The units are presented in general terms

The direct filter
The direct filter consists of equation 4.2. Hence, the required inputs are the output of the FF-sensor as well as current state measurements of the room temperature. In figure 4.5, the activation temperature corresponds to value of 0.5. That is, the FF-part is not activated until the room temperature exceeds the temperature 0.5 degrees below the upper part of the dead-band.

The direct dynamic filter
The second parameter-based filter, the direct dynamic, is also based on equation 4.2 but is also equipped by the dominant parts of the dynamic properties of the room. This was done by approximating the thermal inertia of the building structure by a simple first order linear transfer function. In figure 4.5, the scaling was deactivated. Hence, the output of the direct dynamic filter only consists of the dynamic part included in the filter.

4.3.2.2 FF-filters for IAQ control

In this work, two filters for IAQ control are evaluated. These are referred to as dynamic and non-dynamic and both are based on a mass-balance of the room; a dynamical version in the dynamic filter and a static version in the non-dynamic filter. When an IAQ related disturbance is measured, both filters calculates and returns the supply flow rate of fresh air required to maintain the CO_2 setpoint. Both of the filters do this by assuming an infinite duration of the disturbance.

Since the thermal correspondence to a mass-balance is an energy-balance, the essence of the two filter for IAQ control are also captured in figure 4.4 by comparing the dynamic and static filters respectively.

As mentioned, the classification of filters for thermal control into model- and parameter-based was done based on the complexity of the filter designs. Filters containing energy balances were classified as model-based since they require many inputs- and model parameters. It was also stated that the model-based filters were included as a comparison to the parameter-based and were not considered as implementable. Hence, even though the two filters for IAQ control corresponds to the model-based thermal filters from a design point of view they are regarded as parameter-based since they are considered as implementable. This is motivated by that the required inputs and model parameters to the mass-balance are (c.f. equation G.2 in appendix G):

- The volume of the room
- The supply air flow rate to the room
- The CO_2 concentration in the supply air
- The magnitude of CO_2 emission source

Out of these, the latter is the only variable which can't be determined by relatively simple measures. However, this signal is a property of the FF-sensor and the associated estimation complexity is evaluated separately in the sensitivity analysis in chapter 10. Hence, by assuming that the output of the FF-sensor is given, the complexity of the emission source is not inherited by the FF-filter.

4.3.2.3 FF-filters for motion sensors

In this work, the output of motion sensors is evaluated both for thermal and IAQ control. As mentioned, the output of a motion sensor is limited to 1 or 0. The value 1 denotes an active period, i.e. a period with occupancy, and the value zero an inactive period, i.e. when the room is empty. That is, the output is not proportional to the load and must therefore be processed using alternative methods.

In this work, three different types of filters are evaluated. The main difference between these filter and the previous discussed cases is that the output of the FF-sensor is not used to generate the output of the FF-filters. Instead, the output of the filters is determined by gathering information about the control signals produced by the FB part. The output of the motion sensor is in turn used to determine when this information should be gathered and when the filter should use it to generate an output.

The three filters are named “block and zero”, “block” and “init flow rate” in this work. All three methods act the same when the motion sensor output turns from 1 to 0; then the supplies from the HVAC-components controlled by the FF-part is shut off. Their differences are related to how a motion sensor shift from 0 to 1 is managed.

The filters need a learning period, consisting of one active period, before they can contribute to the control task. The reason is that the filters are using information from the FB-output to generate a potential output for the next active period. The term potential refers in this case to signals stored in the filter during inactive periods to be utilized for the next active period.

How the different filters utilize the information of the FB-controller is presented in figure 4.6 below. The figure illustrates the learning period, i.e. the contribution of the filters is initially zero, and how the potential output of the filters next active period is generated.

The graph furthers to the left illustrates the output of the FB-controller. This output can either be associated by a thermal power or a fresh air supply. The active period (1) constitutes of the learning period of the three filters. The three graphs to the right illustrate how the potential output of the filters is generated during the learning period. The second inactive period (0), marked in grey in the figure, indicates the magnitude of the potential output for the next active period and how this is dependent on the duration of this period.

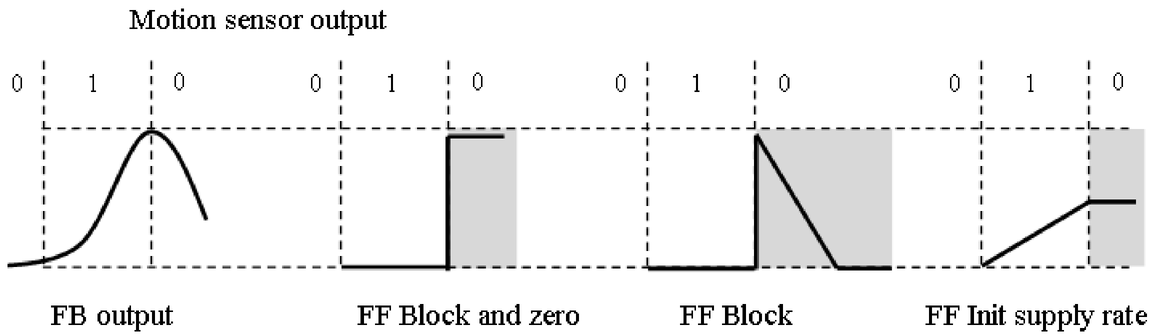


Figure 4.6 The learning period (1) of the filters for motion sensors. The three graphs to the right illustrate how the potential output for the next active period is generated during the learning period for the filters. The second inactive period (0), marked in grey, indicates the potential output stored in the filters for the next active period and how this is dependent on the duration of this period.

The **block and zero** filter as well as the block filter are gathering information about the FB-controller output when the motion sensor shift from 1 to 0. Hence, they are saving the last output signal of the active period to generate their output for the next. As can be seen in the figure, the block and zero filter does that without any constraints; when a room goes from occupied to empty the last quantity of supply air flow rate from the diffuser and/or the last quantity of power supply from the FCU are saved. When the room becomes occupied again, the control signals generated by the FF part correspond to these quantities.

The **block** method does that with a time constrain; the longer a room is empty the more are the saved outputs declining linearly. The decay-time is set to 12 h which correspond to the approximate time duration of one night. Hence, the first motion sensors response of the day does not result in a contribution by the FF-part as can be seen in the figure.

The **init supply rate** filter is utilizing the mean FB output of the entire active period to generate its output for the next.

4.3.2.4 Limitations to ideal control

The FF-controllers in this work are designed to respond to a disturbance by a fast and accurate supply. An accurate supply should be close to the amount of fresh air and power needed to end up or stay at the setpoint. Ideally, the supply is exactly what's needed. Then, the rejection of a disturbance will be total. However, as discussed above, most FF-control designs in this study are simplified and lacking the necessary information to supply the exact amounts. Instead, the supply generated by the FF-controller becomes an approximation of the actual demand.

However, even for an ideal FF-controller, practical limitations prevent a perfect match between supply and demand. These limitations can either be allocated to the FF-filter or HVAC-systems. The limitations allocated to the FF-filters refer to phenomena's which are indescribable in the types of filters used in this work. Hence, by not taking these into account, there will be a gap between the supply determined by the filter and the actual demand.

The limitations allocated to the HVAC-system are referring to design aspects which prevents that the supply determined by the filters can be realised in the same way by the actuators. Such limitations have been avoided as far as possible when the reference system was designed. For example, the HVAC-system was designed with separate system for fresh air and thermal power supply as discussed in section 3.1.3. Further, the distribution systems for air and water was modelled as flow controlled, which also simplified the models, as discussed in the beginning of chapter 3 and section 3.2.2.

Below, the most dominating limitations for an exact match of supply and demand that could not be avoided without aggravating the closeness to reality are listed and further explained.

- Effects not describable in the FF-filter
  - Variable dynamics of the room air
  - Transport-delays
- HVAC-system limitations
  - secondary FB-control loops

Variable dynamics of the room air

As mentioned in section 4.3.1, an HVAC-system with variable supply air flow results in a non-linear behaviour of both room temperature and CO_2 concentration. Another aspect is that a variable air flow also results in variable time-constant of the room air regarding both temperature and CO_2 concentration. This effect is regarded as a FF-filter related issue since variable time-constants are difficult to describe when the model is designed to return the desired fresh air or thermal power supply. Instead, the dynamics of the room air was described using a constant supply air flow rate.

On the other hand, this simplification is almost negligible in the room temperature case. The reason is that the temperature time-constant of the room is totally dominated by the building structure whose dynamics is more or less independent of the supply air flow rate. On the other hand, the IAQ related time-constant of the room is only determined by the supply air flow rate and the room volume which means that the simplification might have a larger influence.

Transport-delays

The dynamical properties of the actuators should preferable be included in the FF-filter. In section 2.2, it was mentioned that time-constants are met by adding a corresponding D-action to the FF-filter to boost the control. However, transport-delays in the distribution system can’t be compensated for in the same manner. Instead, their effect is left out of the FF-model.

Once again, this simplification mainly affects the IAQ control since the largest transport-delays are allocated to the transportation of fresh air. On the other hand, the dynamics of the duct system is small compared to the dynamics of the room.

Secondary FB-control loops

The most important system related issue is the influence of secondary FB-control loops. As presented in section 3.3.1, the temperature control of the air supplied by the FCU is managed by internal FB-control loops. If the demand for cooling or heating is increased, the flow rate through the FCU is increased, and if this is done rapidly the FB-loops has a hard time following. This means that the FCU supply air temperature for a short while will deviate from the setpoint which of course has an effect on the cooling or heating supply rate.

Impacts

The combined effect of the previously discussed limitations on the disturbance rejection properties of FF-controllers is visualized in figure 4.7 and 4.8 respectively. The first figure is referring to room temperature and the distance between two horizontal lines corresponds to 0.2 °C. The second figure is referring to CO_2 concentration and the distance corresponds in this case to 100 PPM.

The figures are produced by simulating the temperature and CO_2 response when a heavy structure meeting room is subjected to a step-shaped occupancy disturbance. The HVAC-system corresponds to the reference system presented in section 3.1. The different graphs are produced by utilizing different control systems as well as adding and removing the limitations discussed above. The FB-control systems are designed according to the procedure discussed in section 3.3.3 and the FF-filter corresponds to the model-based dynamic filter discussed in section 4.3.2.1.

As can be seen, the ideal FF-controller results in a perfect disturbance rejection, i.e. the effect of the disturbance are totally eliminated, which also means that the match between supply and demand is perfect.

The plot denoted as FF-control, refers to the case when all limitations discussed above are taken into account. The control is aggravated but the effect is still extremely small compared to control systems of PI-, and especially, P-controllers.

To clarify, two aspect of this work should be pointed out. First, all of the limitations presented in this section are taken into account in the studied presented in the following chapters. Second, all FB based control systems in this work consists of PI-controllers. The P-controller is only included in this example as a reflection, since this is the most common method for temperature control. However, regarding ventilations system with modulating variable air flow for hygienic purposes, PI-controllers are most often used.

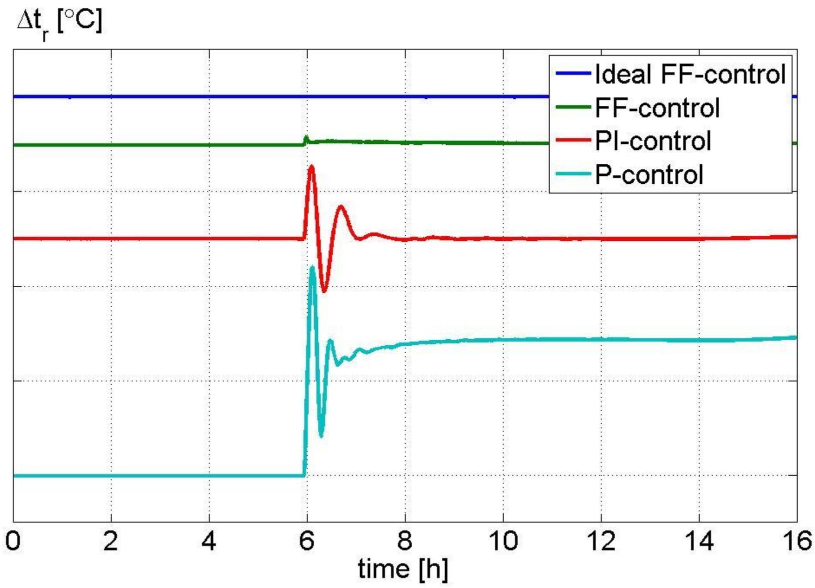


Figure 4.7 The influence of system effects and type of control method on rejection of thermal disturbances. The distance between two horizontal lines corresponds to 0.2 °C.

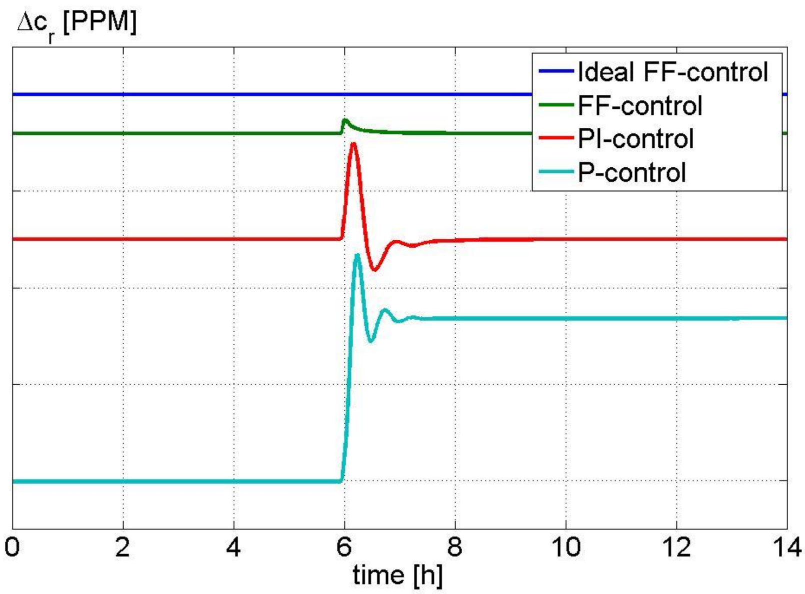


Figure 4.8 The influence of system effects and type of control method on rejection of IAQ related disturbances. The distance between two horizontal lines corresponds to 100 PPM.

5 Local control systems

The challenge of accomplish a high performance feed-forward control is divided into two parts. The first part is related to the properties of the FF-sensor and involves how to produce useful disturbance measurements. Hence, this challenge involves designing the component denoted as F_v in figure 2.6. The second challenge is then to process the output of the sensor in the most favourable way. Hence, to design the filter denoted as F_{ff} in the same figure.

When it comes to thermal control, four different signal processing methods were previously discussed in chapter 4; two designs with a large number of inputs and model parameters (dynamic and static FF) and two less complex (direct dynamic and direct FF). Regarding control of IAQ, two filters, both with a relative simple design, were presented, i.e. the dynamic and the non-dynamic. Furthermore, three methods related to motion sensor based measurements of occupancy were also presented in the same chapter.

In this part, these FF-filters are evaluated for different types of FF-sensors in single-zone platforms. The evaluation is based on the procedure discussed in section 1.3, i.e. based on the energy and peak power indicators connected to the most vital HVAC-components. The different filter and sensor combinations are evaluated for different test setups defined by a set of conditions parameters. These conditions parameters are related to variations which real buildings and HVAC-systems must cope with.

The purpose of this chapter is to determine the best combination of filter and sensor and to identify the type of disturbances with the highest gain in FF applications. The outcome is the FF-controller designs with the best performance for local control regarding energy usage and required peak power of the most vital HVAC-components.

5.1 Work in this field

Just a handful of publications involving FF-control methods for indoor climate control have been found. On the other hand, control methods involving MPC controllers are common and the publications that have been found stretch over a broad area of HVAC related applications.

As indicated in chapter 2.3, MPC can in some cases be viewed as a combination of a FB and a FF part that works together. The main difference between a FF and an MPC controller is that the control signals of an MPC is generated by solving an optimization problem. Otherwise, MPC controllers are based on dynamic models of the control object and measurable disturbances are used as input. This means that MPC controllers can be used in the same way as FF-controllers are used in this work. For that reason, publications regarding MPC controller application in buildings are relevant to study in this part of the work.

5.1.1 Citations regarding FF

The most extensive reference about FF control in building that has been found is the PhD thesis by M.Soleimani-Mohseni[62]. Parts of this work are also presented in a journal article[65].

The thesis by Soleimani-Mohseni partly covers the same grounds as dealt with in this work. Also his work focuses on FF design and is based on simulation studies. Static and dynamic FF-filters are evaluated and the evaluation is performed by comparing the energy usage and the mean temperature range to FB-controllers.

However, there are a number of differences between his work and the work presented in this report. The most important ones are presented below. The work by M.Soleimani-Mohseni...

- ...only covers temperature control while this work also covers IAQ control
- ...deals only with arbitrary single-zone platforms while this work involves various kinds of building designs as well as multi-zone platforms
- ...describes the HVAC-components and sensors by simple linear transfer functions in an arbitrary way, while the HVAC-components in this work are based on the behaviour of real components
- ...are including evaluations of FF-control systems with P-controllers while FF-control systems with PI-controllers are evaluated in this work
- ...focuses only on disturbances from electrical equipment with arbitrary duty cycles. All other modelled disturbances are treated as constants. In this work, all relevant disturbances are varied according to profiles based on measurements or on estimations
- ...focuses on a constant temperature setpoint while a temperature dead-band is used in this work
- ...does not review the differences in required peak power of the HVAC-components when FF and FB controllers are used

The results show that the dynamic filter is preferable from a performance point of view but that the static filter can be used as an approximation. Soleimani-Mohseni concludes that the energy usage is not affected very much when FF-controllers are used but the mean temperature range becomes much smaller. Hence, the control of the room temperature becomes more stable since the disturbance rejection feature of the control system is improved.

Also the work presented by T.Lu in a journal article is relevant in this context[44]. The work is applied to ventilation control in a training facility. T.Lu utilizes that the times the facility is occupied are scheduled and the number of participants are known in advance. This information is used as input to a FF-filter consisting of a mass-balance of the room. The aim is to keep the CO_2 concentration close to 1000 PPM without breaching this limit and by that reducing the supply air flow. Compared to a CAV (constant air volume) ventilation system, the energy usage is reduced by about 50 %.

Another work similar to the work of T.Lu above is presented by X.B.Yang in a journal article[73]. Yang estimates the number of persons inside the building by comparing the inside and outside CO_2 concentration. The number of persons is used as input to the ventilation control and the purpose is to minimize the outside air flow to reduce the energy usage. The maximum estimation error of number of persons was 8.7 % and the strategy result in an energy savings of about 15 %

The work by A.E.Ruano deals with FF-based temperature control[57]. The FF-filter consists of a thermal black-box model of a room and the HVAC-system is of an all-air type. The FF-filter is used to predict the room temperature 30 min ahead. When a disturbance is measured, the prediction is performed and if the temperature is assessed to breach the dead-band within 30 minutes a control action is performed. If the temperature is predicted within the dead-band, nothing is done. The results show that the temperature is kept within the dead-band at all times and that the duration of the HVAC-system is decreased by 27 %.

A little more peripheral but still relevant work is published in a journal article by M.H.Sherman[59]. This work deals with ventilation control which utilises the time of the day and the quality of the outdoor air as inputs. The time of the day is used to reduce the air flow during periods when the occupancy is known to be low. The outdoor air quality is used to shut off the ventilation during periods when the pollutants outside is large. These strategies reduce the energy usage by about 60 % and the duration of the ventilation system with about 30 %. At the same time is the IAQ improved.

Two quite similar articles presented by D.Shiming[60] and Zaheer-Uddin[75] are describing a quite different application for FF-control. Both articles uses local FF-controllers for temperature control in an AHU. The work by D.Shiming involves cooling of humid air and the work by Zaheer-Uddin involves heating. D.Shiming identifies that problems regarding instability often occur when a FB-controller based on a sensor at the outlet of the AHU is used due to inaccurate measurements. The work aims to solve this problem using a FF-controller including a FF-filter based on an energy balance. The results show that the temperature control is increased; from a several minutes to 90 s. Zaheer-Uddin uses the FF-controller to minimize the effect of the ambient climate to increase the stability of the control.

5.1.2 Citations regarding MPC and optimized control

In two articles, presented by M.Castilla[15] and P.D.Morosan[47] the number of persons inside an office room as well as the opening of the door is used as input to local MPC controllers. The MPC is used to control the heating supplied to the room. The work aims to optimize the PMV (predicted mean vote) and to minimize the variations in room temperature. The results show that that the thermal comfort is increased, and that the total energy usage is decreased by about 14-12 % compared to when conventional controllers are used.

In the work by D.Kolokotsa, an extensive MPC control strategy is implemented in two office buildings with all-air systems in Greece[39]. The controller is used to control the HVAC-system and lighting and can also perform passive measures such as shading of the sun and to induce natural ventilation by open and close the windows. The inputs to the controller are;

- Mean radiant temperature
- Indoor temperature
- Indoor relative humidity

- Air velocity
- Indoor CO_2 concentration
- Indoor illuminance
- OAT
- Outdoor relative humidity

The purpose is to optimize the lighting control and thermal climate according to PMV and the passive methods are prioritized to reduce the energy usage. The results show that the annual total energy usage is reduced with 38 % compared to conventional on/off control of the HVAC-system and manual lighting control.

Three other articles related to D.Kolokotsas work are presented by F.Oldewurtel[51], B.Paris[52] and M.Kintner-Meyer[37]. In these, weather forecasts are used as input to an optimal controller and the energy for temperature control can be generated by different units such as boilers, cooling machines, solar panels and evaporative cooling. The controller is used to maximize the renewable energy part by using the building structure as heat storage. The results shows that the non-renewable energy usage was reduce between about 35 and 0.5 % compared to simple on/off control. Kintner-Meyer has also showed that the peak electrical power is reduced by about 40 %.

Two other articles similar to the ones above are presented by S.Privara[56] and J.Siroky[61]. In both articles, MPC controllers are used to control the heat supplied in office buildings. Both Privara and Siroky utilize weather forecasts as input to the MPC, but Siroky also utilizes the occupancy. Privaras results shows that energy savings of about 20 % are possible compared to the when conventional control is used. Sirosky shows that the energy savings are dependent on the building structure. In well-isolated buildings the energy savings are about 30 % and in poor-isolated buildings, the savings are about 20 % compared to when conventional controllers.

In the work by V.Congradac[18] an optimal controller is instead used for ventilation control. The application is close to the work presented in this report and the purpose is to mitigate the effect of disturbances to be able move the CO_2 setpoint closer to a threshold limit in an office building in Belgrad. The results show that the duration of a chiller can be reduced by about 20 %.

5.2 Methodology

The study presented in this chapter was carried out by simulating a single-zone platform for a specific set of setup-parameters. That is, one room was simulated at the time and the entire reference HVAC-system was dedicated for that room. One simulation is extending over a 12 hours long working day. In addition, 3 hours are added before and after the 12 hours period for stabilization.

This part of the work only handles the control of supply air flow rate and the FCU operation for heating and cooling. The control of the SAT is not yet included as a control objective and the corresponding setpoint is set as a constant during one simulation. However, in some sense the SAT reflects the outside temperature since the setpoint is set to 15 °C during summer and to 18 °C during winter conditions.

5.2.1 Test setup design

A simulation test setup is defined by the set of parameters presented below.

1. Type of FF-control design
   - Type of FF-filter
   - Type of FF-sensor
2. Condition parameters
   - Type of ambient climate
   - Type of building density
   - Type of relation between internal and external disturbances, i.e. meeting or office room

In this study, the cause-and-effect of these setup-parameters on the performance of FF-controllers is evaluated. These setup-parameters are given as input parameters to the simulations in the study. Hence, they are changed from one simulation to another and different combinations are evaluated. One combination of parameters is referred to as a test setup in the following text.

Relation between internal and external disturbances
In the test setups, this test-parameter is realized by distinguishing between the two types of rooms presented in section 3.4.1.1. From a size point of view, the two rooms are intended to represent an office room and a meeting room. From a disturbance point of view, the two rooms along with its disturbance profiles results in two distinguished cases regarding the relation between external and internal disturbances. Since the office room is designed for one person and one of the long-sides is facing the ambience, the external disturbances become relative large. On the contrary, in the meeting room case, all walls are interior and the internal heat loads are of a larger magnitude.

As will be discussed in the following section, the external disturbances vary slowly while the internal are introduced as steps with relative large heights. That is, this test-parameter primary investigates the influence of the amplitude and the frequency of the disturbances on the potential of FF-control.

Building density
The test setups also aim to determine the influence of building structure density on FF-control. This is carried out by including a heavy and a light version of the rooms as presented in section 3.4.1.1. As mentioned, the heavy structure is based on concrete and the light is based on gypsum and wood. The time constant relation between these structures is of a magnitude of about two.

Ambient climate
Test setups also distinguish between summer and winter outside conditions to evaluate the influence of ambient climate on the potential of FF-control. These conditions are based on climate data of a normal year between 1961-1990 from the city of Helsingborg in Sweden.

5.2.2 Disturbance profiles

As indicated, one simulation consists of a 12 h working day, from 7:00 to 19:00, with a 3 hours stabilization period before and after. During the simulation, the room is subjected to both time varying internal and external disturbances.

Below in figure 5.1 and 5.2, the internal disturbance profiles for the meeting and office room are presented respectively. The lighting corresponds in both cases to 10 W/m^2 of floor area and the profile follows the occupancy; when the rooms are empty the lighting is off and vice versa. The occupancy is intended to follow regular working-hours; the room is mostly occupied from 7:00 to 18:00 except during lunch-time.

Meeting room

The figure below illustrates the disturbance profiles for the meeting room. These consist entirely of internal types since the meeting room lacking exterior walls. However, the inlet temperature to the AHU corresponds to the OAT presented as summer conditions in the office room case in figure 5.2. It is further assumed that the time that the door to the meeting room is opened is negligible.

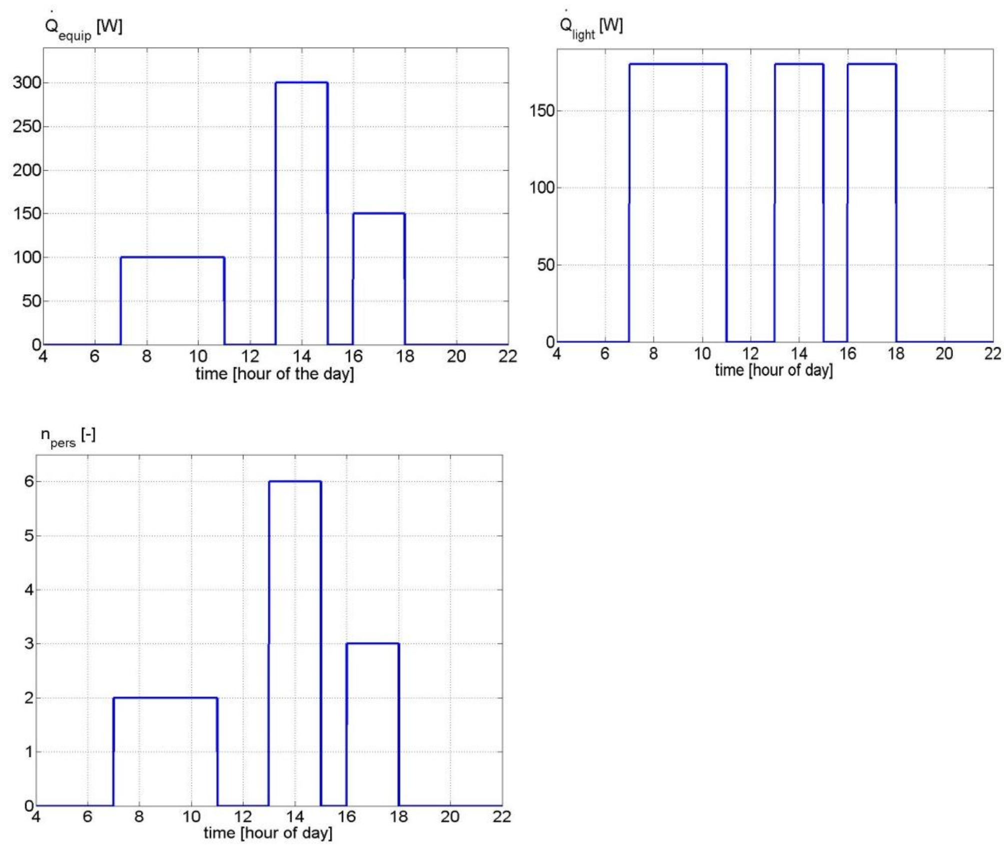


Figure 5.1 Disturbance profiles for meeting room case. Heat supplied from equipment and lighting, number of persons inside the room

The maximum occupancy of six persons in figure 5.1 corresponds to an occupancy factor (OF), defined in equation 5.1, of 0.7. This is in agreement with the load profiles derived from measurements in office buildings presented by M. Maripuu [45]. Each person emits 70 W of sensible body heat as well as additionally

50 W originated by equipment. Hence, the equipment disturbance is assumed to be correlated and proportional with occupancy.

$$OF = \frac{n_{people}}{n_{people,design}} \qquad \text{(eq. 5.1)}$$

Office room

Below, the corresponding profiles for the office room are illustrated. The internal disturbances in figure 5.2 also include the opening and closing of the door which in this case also is treated as a disturbance. The equipment is represented as one computer that is running during the day.

The room is designed for one person and the occupancy factor is 1 for most of the simulated time. In figure 5.3 and 5.4 the external disturbances are shown for the summer and winter ambient climate respectively.

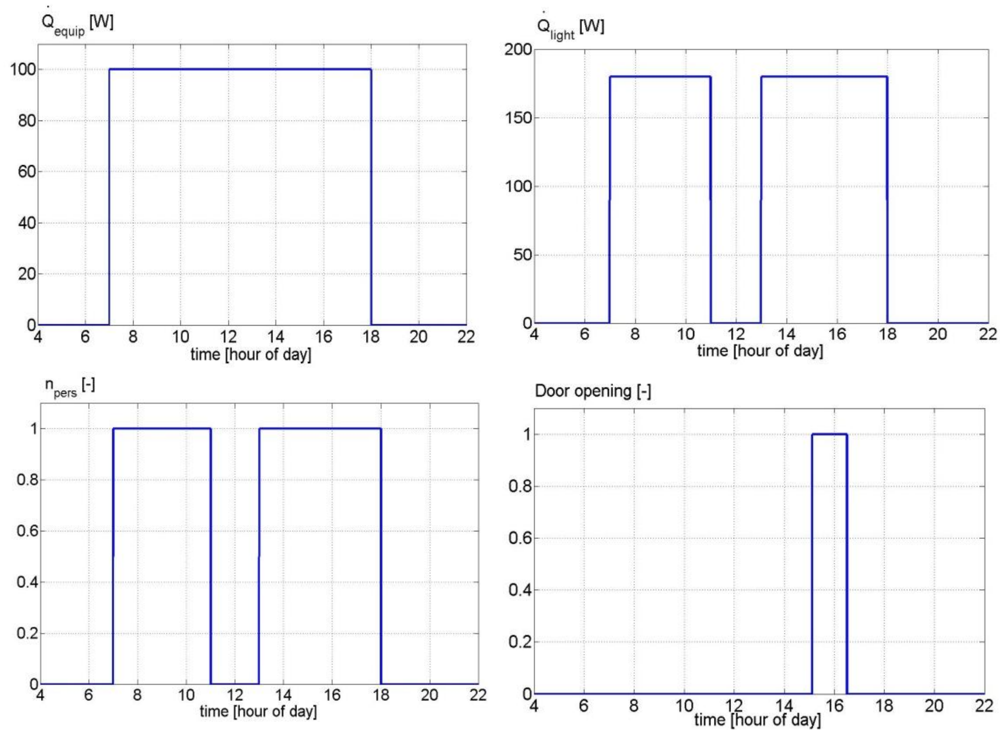


Figure 5.2 Internal disturbance profiles for office room case. Heat supplied from equipment and lighting, number of persons inside the room and door opening

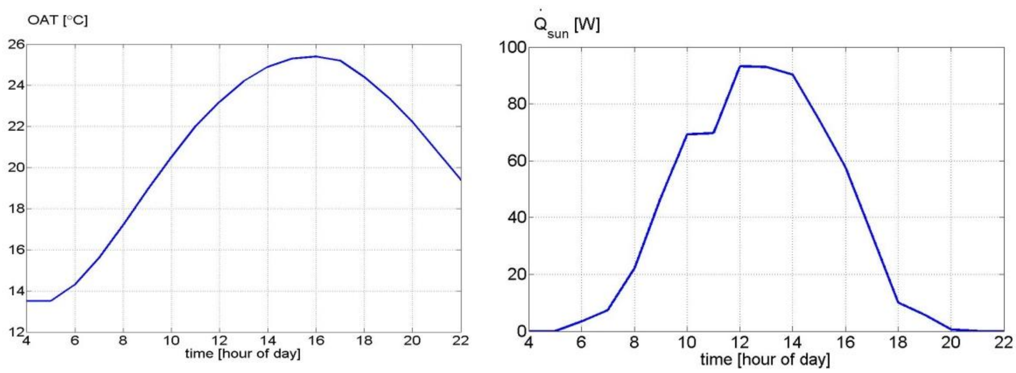


Figure 5.3 External disturbance profiles for office room case during summer conditions. Outdoor air temperature and solar heat supplied to the room respectively. Normal year climate data from Helsingborg, Sweden

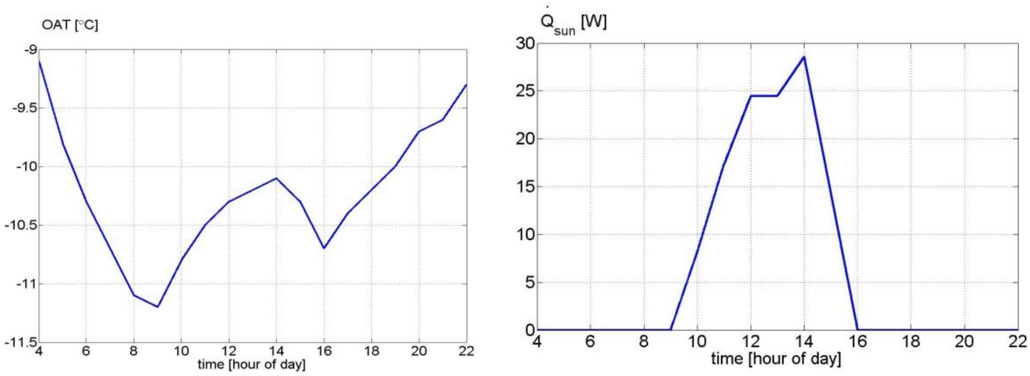


Figure 5.4 External disturbance profiles for office room case during winter conditions. Outdoor air temperature and solar heat supplied to the room respectively. Normal year climate data from Helsingborg, Sweden

Thermal power demand

Due to the isothermal boundary conditions and the relatively high internal heat loads, there is no demand for heating in the meeting room cases. Hence, the thermal energy used by the FCU only consists of supplied cooling. This is also the case for the office room during the summer condition, due to a relatively high outside temperature, even though the internal heat loads are relatively small. These two cases are hence treated as pure cooling demand cases. On the contrary, the office room during winter case only has a demand for heating due to the low outside temperature.

5.2.3 FF design selection process

The selection process consists of two steps. The overall purpose is to find the FF-controllers with the best performance regarding energy usage and peak-power required of the most vital HVAC-components.

- The first step aims to find the best FF-filter and FF-sensor combinations.

- The second step aims to further combine these selected FF-sensor and FF- from the first step to find the overall best FF-control system. This FF-control system is further evaluated in the following chapters.

During the entire selection process, the model-based filters and parameter-based filters have been evaluated in parallel. This means that the outcome of this study is the two most suitable FF-control systems; one with a model-based filter and one with a parameter-based filter. The purpose is to show the potential of FF-control using complex but accurate filters as well as show present an alternative which is more close to implementation.

In the first step it is easy to distinguish between FF-controller with model- and parameter-based filters. However, in the second step, filter for IAQ and thermal control are combined which means that one FF-control system can contain both model-based and parameter-based filters. In these cases, a FF-control system containing at least one filter that is based on an energy-balance is referred to as model-based.

5.2.3.1 Step 1

The process of selecting the most suitable filter and sensor combinations is carried out by treating the test setups individually. This means that for each setup, the most suitable combination is selected.

The process aim to evaluate each FF-sensor for the proposed FF-filters presented in chapter 4. In most cases, this is done in a direct manner by simulating the setups and evaluating the performance of the control systems.

Indirect selection
Another method that has been used is to select the most favourable filter-sensors combination indirectly based on the combinations that were directly selected in other setups. An indirect selection from one setup to another was only allowed if the prerequisites for the control task and the properties of the room essentially were the same in both cases. This scenario is true for setups with the same governing conditions parameters.

When it comes to FF-systems for thermal control, the governing condition parameters were considered as the ambient climate and the building structure density. The main reason is that the dynamical thermal property of the room is affected by the density of the building and that the type of thermal power demand might vary between winter and summer. This means that a selected FF-design for thermal control can be transferred between setups of equal ambient climate and building structure density.

When it comes to FF-systems for IAQ control, the governing condition parameter was considered as the size of the room. The main reason is that the dynamic property of the mass-balance is dependent on the air-volume. This means that a selected FF-design for IAQ control can be transferred between setups with the same type of rooms.

Occupancy and door opening are affecting both the thermal climate and the IAQ. The filter for thermal control was in these cases derived indirectly based on the type of filter selected for the electrical power sensor. This was considered as valid since these three thermal disturbances are all associated with a heat load that directly affects the room air. Further, the characteristics of the equipment are very similar to the heat supply characteristics of the persons and door. Hence, the personal counter and the door opening sensor are only evaluated for the IAQ controlling filters.

It should be mentioned that the indirect selection approach has been verified a number of times. This was done by simulating randomly selected setups in which indirect selection had been applied. In all cases, the simulated results corresponded with the indirect conclusion.

Selection criterions
The first step is a course selective process to find the most suitable FF-filter for a certain FF-sensor. It is treated a bit differently than the other selection processes in this work which are based on the methodology described in section 1.3. Step 1 also includes an elimination part; if any FF-sensors show no potential, it is eliminated and rejected from further testing.

This step is carried out by designing the control systems to maintain a fixed set-point of both CO_2 and temperature. The FF-controllers were evaluated by comparing their performances to the performance of a FB controller. The performance was measured according to the nine criterions below.

- Maximum overshoot of CO_2-setpoint
- Maximum overshoot of temperature-setpoint
- Use of electrical energy
- Use of cooling energy
- Use of heating energy
- Total energy use
- Peaking power of AHU
- Peaking power of central fan
- Peaking power of FCU

The results are given as grades that were calculated for each criterion using equation 5.2. In this equation, *x* denotes the value of the criterion in question. A grade is relative within the group of filters evaluated for the same FF-sensor. The difference between the FF and FB-controller is divided by the results of the filter that showed the best performance for the specific filter-sensor combination. To determine the overall performance of a certain FF-controller, the values of the 9 grades are summed up.

$$\sum_{i=1}^{N=9} \left. \frac{x_{FB} - x_{i,FF}}{x_{max,FF}} \right|_{0,1} \quad [-] \qquad \text{(eq. 5.2)}$$

Note that a grade is limited to values between one and zero. When a filter is awarded a grade of one, it means that this filter is the most suitable for that FF-sensor during the prevailing condition parameters. A value of zero represents equal or worse performance compared to the corresponding FB-controller. That is, each filter can score a maximum of 9 and a minimum of 0. In the cases when two filters ends up with the same grade, the simples one regarding complexity of the FF-filter is automatically selected.

Relative grades are used since the aim is to find the most appropriate FF-filter for a certain FF-sensor. Hence, it makes sense to compare the filters FF-sensor wise. However, even though the grading is relative, the potential of the different FF filters can be compared across the sensors. That is, because all the filters are compared to the same FB-controller, the values of the grades also reflect their performance in absolute terms.

Meeting room

To begin the selective process, the first step was initially carried out for the test setups consisting of a meeting room with heavy and light building structure respectively. From this study it could be concluded;

- Which filters that should be used together with
  - Personal counters (IAQ part) in meeting rooms
  - Motion sensors in meeting rooms
  - Electrical power sensors in light and heavy structures with cooling demand

According to the concept of indirect selection discussed above, the third conclusion was also valid for;

- How to treat the same disturbances in the cooling demand office room setups
- How to treat the thermal part of the filter for a person counter in the meeting and office cooling demand cases

In figure 5.5, a schematic picture of this part of the selection process is presented. The red boxes indicate the filter-sensor combinations that were evaluated directly. The number of combinations that were tested is indicated by the value in the parenthesis, which is the same as the number of filters in the set. The outcome of this part of the evaluation consists of the most favourable filter-sensor associated with each red box.

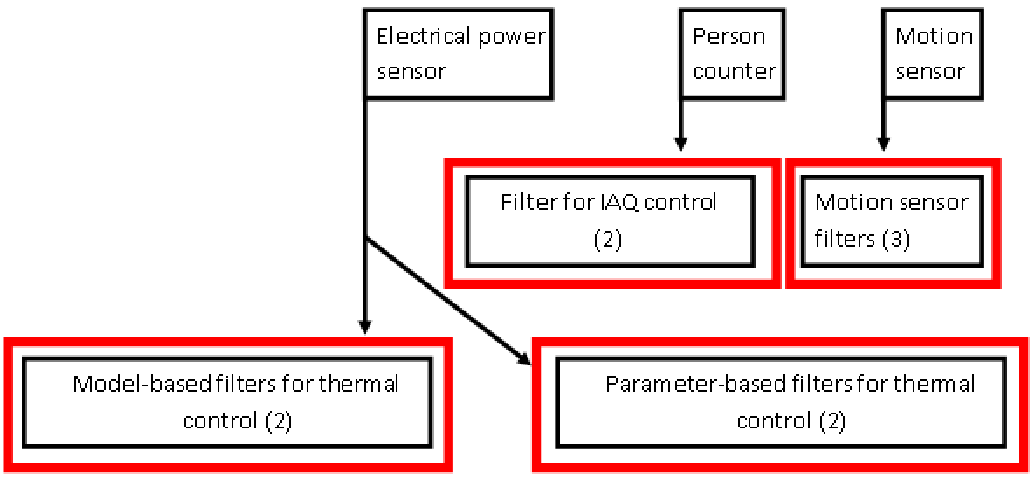


Figure 5.5 Schematic picture of the selection process conducted in the meeting room setup. The most favourable filter-sensor combination was selected from each red box.

Office with cooling demand
In the next step, the remaining sensor-filter combinations that were not indirectly derived from the meeting room setup were evaluated for heavy and light structures office rooms during summer ambient conditions. From these studies it was concluded:

- Which filters that should be used together with
  - Personal counters (IAQ part) in office rooms
  - Door opening sensor (IAQ part) in office rooms
  - Solar heat gain sensor in light and heavy structures with cooling demand
  - OAT sensor in light and heavy structures with cooling demand

According to the concept of indirect selection, the first and second conclusion also applies in the office rooms with heating demand setups.

In figure 5.6, a schematic picture of this part of the selection process is presented. The red boxes indicate the filter-sensor combinations that were evaluated directly. The number of combinations that were tested is indicated by the value in the parenthesis, which is the same as the number of filters in the set. The outcome of this part of the evaluation consists of the most favourable filter-sensor associated with each red box.

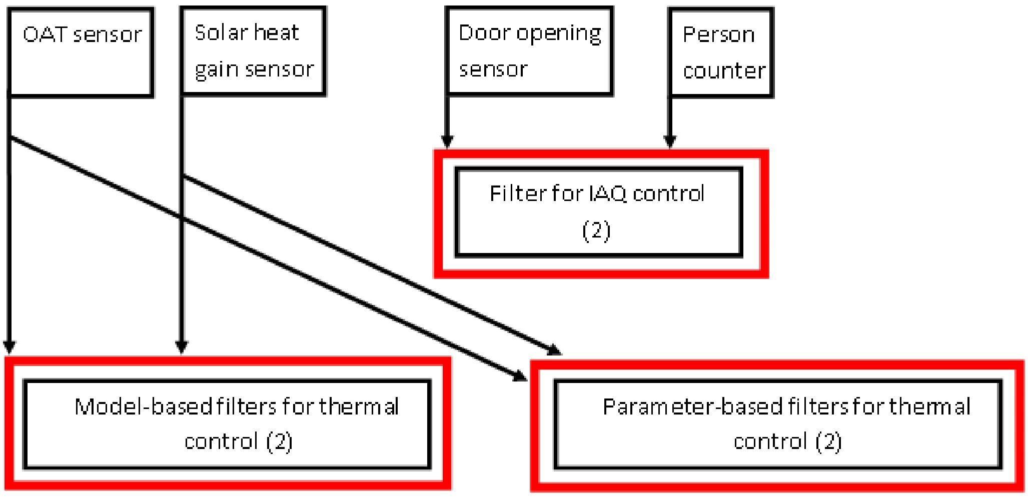


Figure 5.6 Schematic picture of the selection process conducted in the office room with cooling demand setup. The most favourable filter-sensor combination was selected from each red box.

Office with heating demand
Finally, the setups consisting of office rooms with heat demand was evaluated. Since the governing parameter of ambient climate was changed, all of the thermal sensor-filter combinations were evaluated again. From these studies it was concluded:

- Which filters that should be used together with
  - Electrical power sensor in light and heavy structures with heating demand
  - Solar heat gain sensor in light and heavy structures with heating demand

- OAT sensor in light and heavy structures with heating demand

According to the concept of indirect selection, the first conclusion was also valid for how to treat the thermal part of the filter for a person counter in the office heating demand setups.

In figure 5.7, a schematic picture of this part of the selection process is presented. The red boxes indicate the filter-sensor combinations that were evaluated directly. The number of combinations that were tested is indicated by the value in the parenthesis, which is the same as the number of filters in the set. The outcome of this part of the evaluation consists of the most favourable filter-sensor associated with each red box.

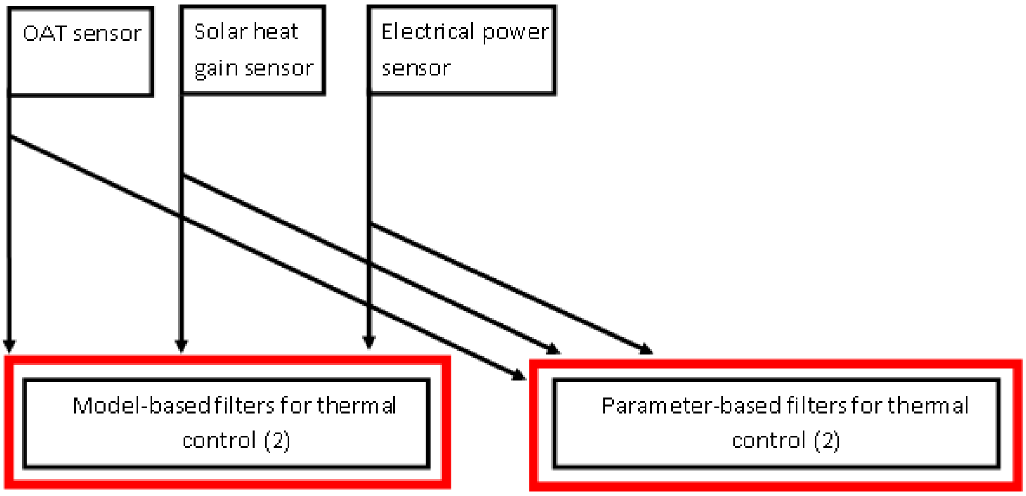


Figure 5.7 Schematic picture of the selection process conducted in the office room with heating demand setup. The most favourable filter-sensor combination was selected from each red box.

5.2.3.2 Step 2

As mentioned, in the second step the selected sensor-filters combinations from step 1 are further combined with each other. The purpose is to determine the overall best FF-control designs.

This step is carried out step-wise; starting off with one selected combination, and to include one more FF-sensor and its FF-filter at the time. That is, to take one more disturbance into account at the time. The final output from this step is the two FF-control system with the overall best performance; one parameter-based and one model-based.

Selection criterions
The evaluation of the FF-control systems is done by comparing the performance to FB-control systems. The evaluation is in this case performed according to the methodology described in section 1.3. That is, the performances of two control systems are comparable and thereby referred to as corresponding if equal fulfilment of the control task is accomplished. Both the temperature and the CO_2 criterion are taken into account in this part of the work. Hence, the CO_2 peaks of all control systems coincide at 1000 PPM. Also, two compared control systems results in equal degree hours outside the temperature dead-band.

The evaluation is performed using the energy indicators presented in section 1.3.3, i.e. thermal, electrical and total energy, as well as the peak power indicators related to central fan, AHU and FCU. Since the energy indicators are sums of detached terms, these results are not very transparent when it comes to understanding the fundamental differences of FF and FB control. As a complement to make this clearer, additional data of mean supply values from the different HVAC-components are included.

Simulation initial values

The setpoints of the FB-controllers were tuned to achieve the same fulfilment of the control task as the FF-controller it is compared to. This is visual in the simulated plots as different initial values regarding room temperature depending on which controller that is used.

The initial values of the CO_2 concentration are set to reflect that the simulated period is preceded by a short inactive period. A value 100 PPM below the setpoint of the FB-controller was chosen in all cases. Hence, the duration of the intended preceding inactive period is not long enough to reach the outdoor concentration of CO_2 due to infiltration. But it is long enough for the CO_2 concentration to drop 100 PPM below the setpoint.

5.3 Results

The main body of results from this study are presented in appendixes A and B. All results from step 1 are presented in appendix A and the main body of results from step 2 are presented in appendix B. In the section below, the primary output of step 2 are presented. That is, two FF-control systems, one model-based and one parameter-based, with the highest performance in each test setup.

5.3.1 Meeting room

Below, the primary output of step 2 is presented for the meeting room setups. The output consists of the model-based and the parameter-based FF-control systems for both thermal and IAQ control with the highest performance. The performance, i.e. the energy and peak power indicators, are presented in tables and simulated evolution of room temperatures and CO_2 concentrations are presented in figures.

5.3.1.1 Selected filter for IAQ control

The filter-sensor combination that was proven most favourable for IAQ control was person counter and non-dynamic filter. In figure 5.8, the evolution of the CO_2 concentration is shown when this FF-controller is used in the meeting room. In the same figure, the corresponding evolution when the IAQ control is managed by a FB-controller is also presented. As can be seen, both of the control systems results in a maximum CO_2 peak of 1000 PPM which is the criterion for comparability of IAQ controllers.

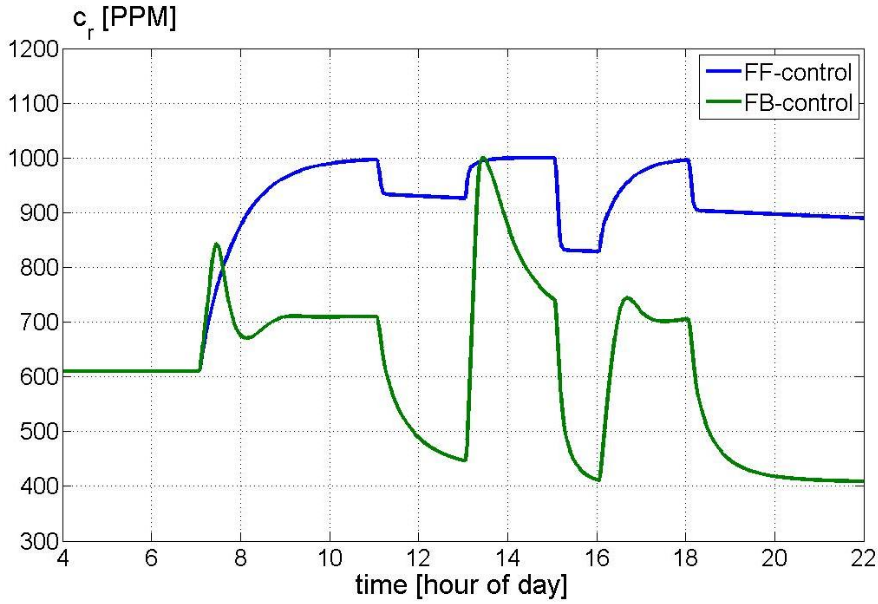


Figure 5.8 Evolution of room CO_2 concentration during the simulated 12 h. Comparison between FB control and FF control for the meeting room setup

5.3.1.2 Primary results

Heavy structure

In table 5.1 and 5.2, as well as in figure 5.7 and 5.8, the primary results for the test setup consisting of a meeting room with heavy building structure are presented. Table 5.1 presents the energy indicators and table 5.2 the peak power indicators. The first row of results in each table corresponds to the selected parameter-based FF-control system and the second row to the selected model-based FF-control system.

Table 5.1 Final result of step 2 in the FF-control design selection process. Potential of FF regarding energy usage compared to a corresponding FB control system. Test setup: meeting room, heavy building structure.

	Cooling energy savings [%]	Electrical energy savings [%]	Total energy savings [%]
Person counter and electrical power sensor (IAQ non-dynamic and thermal direct)	17	70	28
Person counter and electrical power sensor (IAQ non-dynamic and thermal dynamic)	25	70	34

Table 5.2 Final result of step 2 in the FF-control design selection process. Potential of FF regarding peak power requirement compared to a corresponding FB control system. Test setup: meeting room, heavy building structure

	FCU power savings [%]	AHU power savings [%]	Electrical power savings [%]
Person counter and electrical power sensor (IAQ non-dynamic and thermal direct)	43	44	73
Person counter and electrical power sensor (IAQ non-dynamic and thermal dynamic)	22	45	73

Figure 5.9 and figure 5.10 shows the evolutions of the room temperature when the selected parameter-based and the selected model-based FF-controllers are used respectively. In both figure, also the evolution of the corresponding FB-controller is presented. As mentioned, corresponding refers in the thermal case to the comparability criterion of equal degree-hours outside the temperature dead-band located between 22 and 21 °C.

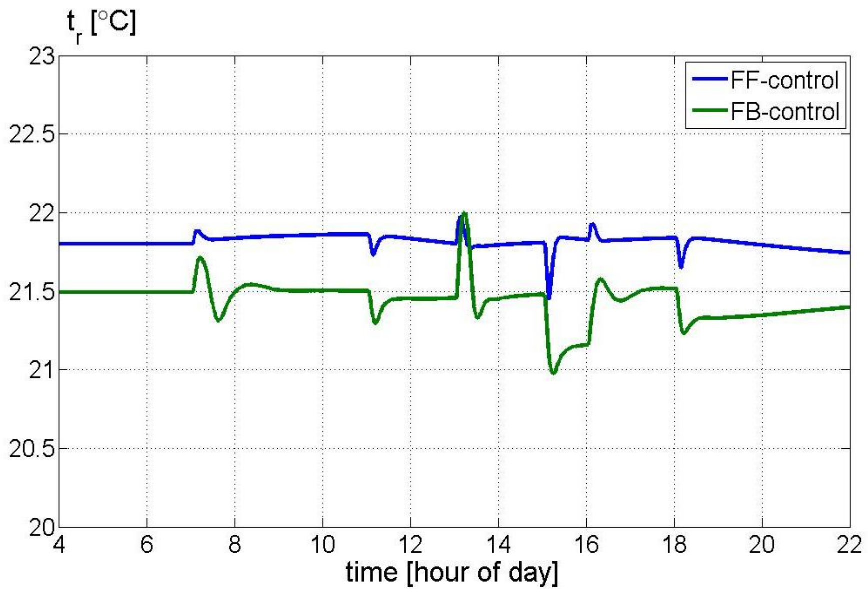


Figure 5.9 Evolution of room air temperature during the simulated 12 h. Comparison between FB control and parameter-based FF control for the meeting room with a heavy building structure

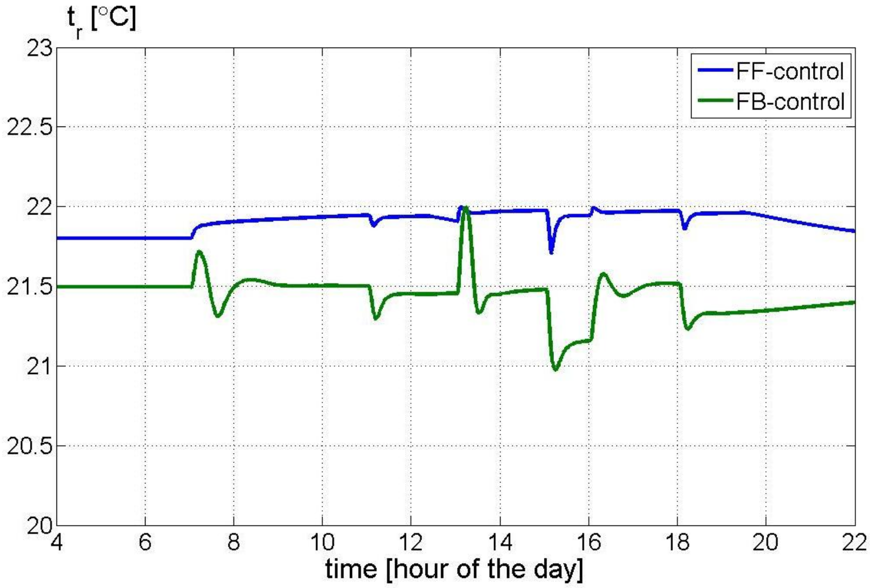


Figure 5.10 Evolution of room air temperature during the simulated 12 h. Comparison between FB control and model-based FF control for the meeting room with a heavy building structure

Light structure

Below, the results for the meeting room with a light building structure are presented. Table 5.3 present the energy indicators and table 5.4 the peak power indicators. Just like before, the first row of results in the tables corresponds to the selected parameter-based FF-controller and the second row to the selected model-based.

Table 5.3 Final result of step 2 in the FF-control design selection process. Potential of FF regarding energy usage compared to a corresponding FB control system. Test setup: meeting room, light building structure.

	Cooling energy savings [%]	Electrical energy savings [%]	Total energy savings [%]
Person counter and electrical power sensor (IAQ non-dynamic and thermal direct)	6	67	18
Person counter and electrical power sensor (IAQ non-dynamic and thermal dynamic)	7	68	19

Table 5.4 Final result of step 2 in the FF-control design selection process. Potential of FF regarding peak power requirement compared to a corresponding FB control system. Test setup: meeting room, light building structure

	FCU power savings [%]	AHU power savings [%]	Electrical power savings [%]
Person counter and electrical power sensor (IAQ non-dynamic and thermal direct)	13	44	73
Person counter and electrical power sensor (IAQ non-dynamic and thermal dynamic)	16	44	73

Figure 5.11 and figure 5.12 shows the evolutions of the room temperature when the selected parameter-based and the selected model-based FF-controllers are used respectively. In both figure, also the evolution of the corresponding FB-controller is presented.

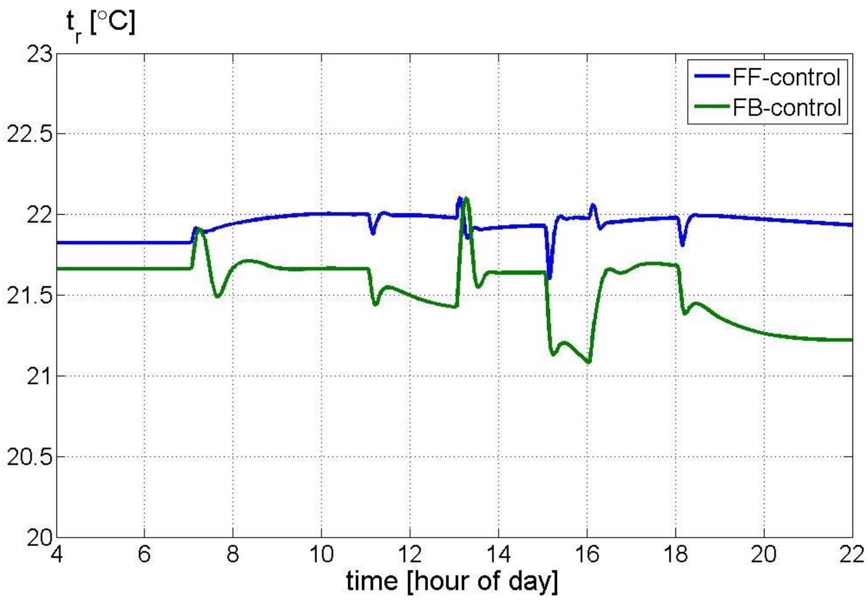


Figure 5.11 Evolution of room air temperature during the simulated 12 h. Comparison between FB control and parameter-based FF control for the meeting room with a light building structure

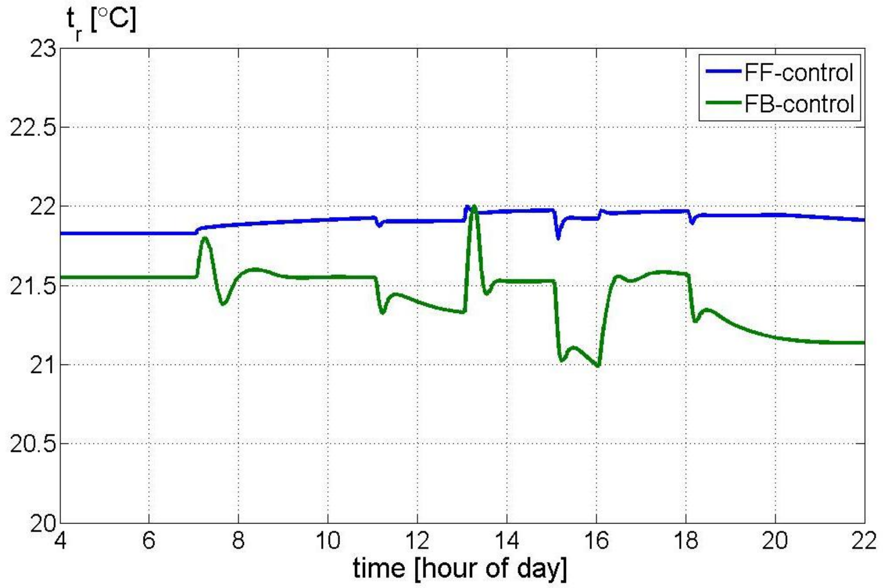


Figure 5.12 Evolution of room air temperature during the simulated 12 h. Comparison between FB control and model-based FF control for the meeting room with a light building structure

5.3.1.3 Additional results

Above, the results that make up the basis for the selection process were presented. But as mentioned previously, there is a need for complementary data to further clarify the differences between FF and FB control. These are presented as the ratio between the supply mean values from the different HVAC-components when FF- and FB-controllers are used. Table 5.5 is associated with heavy building structures and table 5.6 to light.

Table 5.5 Comparison of mean values of supplied fresh air and thermal energy. Ratio between the FF-control and FB-control systems when applied to a meeting room in a heavy building structure

	Mean supply air flow rate, FF/FB [-]	Mean FCU thermal energy supply, FF/FB [-]	Mean AHU thermal energy supply, FF/FB [-]
Person counter and electrical power sensor (IAQ non-dynamic and thermal direct)	0.5	1.8	0.5
Person counter and electrical power sensor (IAQ non-dynamic and thermal dynamic)	0.5	1.5	0.5

Table 5.6 Comparison of mean values of supplied fresh air and thermal energy. Ratio between the FF-control and FB-control systems when applied to a meeting room in a light building structure

	Mean supply air flow rate, FF/FB [-]	Mean FCU thermal energy supply, FF/FB [-]	Mean AHU thermal energy supply, FF/FB [-]
Person counter and electrical power sensor (IAQ non-dynamic and thermal direct)	0.5	1.9	0.5
Person counter and electrical power sensor (IAQ non-dynamic and thermal dynamic)	0.5	1.8	0.5

5.3.2 Office room

Below, the primary results of the office room setups are presented. Just as in the meeting room case, these results consist of the two FF-controller for both thermal and IAQ control, one model-based and one parameter-based, with the highest performance in each test setup.

5.3.2.1 Selected filter for IAQ control

In the office setups, the IAQ is affected by the occupancy and the door opening. Both of the corresponding FF-sensors were preferable combined with the non-dynamic filter. Figure 5.13 presents the evolution of the CO_2 concentration in the office room setup when both of these disturbances are utilized in the control system. The door is opened between 15:00 and 16:30 according which previously was visualized in figure 5.2 in section 5.2.2.

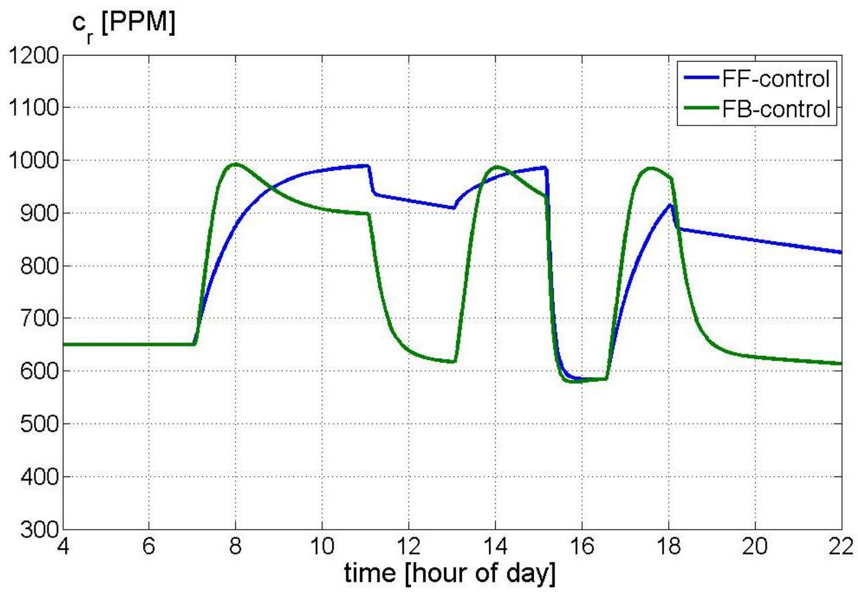


Figure 5.13 Evolution of room CO_2 concentration during the simulated 12 h. Comparison between FB control and FF control for the office room

5.3.2.2 Primary results

Heavy structure ambient summer setup
In the following tables the results for the office room with a heavy building structure during summer conditions are presented. Table 5.7 present the energy indicators and table 5.8 the peak power indicators. The first row of results in the tables corresponds to the parameter-based FF-controller and the second row to the model-based.

Table 5.7 Final result of step two in FF-control design selection process. Potential of FF regarding energy usage compared to a corresponding FB control system. Test setup: office room, heavy building structure, summer outside conditions

	Cooling energy savings [%]	Electrical energy savings [%]	Total energy savings [%]
Person counter, electrical power sensor and door opening sensor (IAQ non-dynamic and thermal direct)	3	17	4
Person counter, electrical power sensor and door opening sensor (IAQ non-dynamic and thermal dynamic)	6	18	7

Table 5.8 Final result of step two in FF-control design selection process. Potential of FF regarding power peak reduction compared to a corresponding FB control system. Test setup: office room, heavy building structure, summer outside conditions

	FCU power savings [%]	AHU power savings [%]	Electrical power savings [%]
Person counter, electrical power sensor and door opening sensor (IAQ non-dynamic and thermal direct)	10	16	35
Person counter, electrical power sensor and door opening sensor (IAQ non-dynamic and thermal dynamic)	9	16	35

Figure 5.14 and figure 5.15 shows the evolutions of the room temperature when the selected parameter-based and the selected model-based FF-controllers are used respectively. In both figure, also the evolution of the corresponding FB-controller is presented.

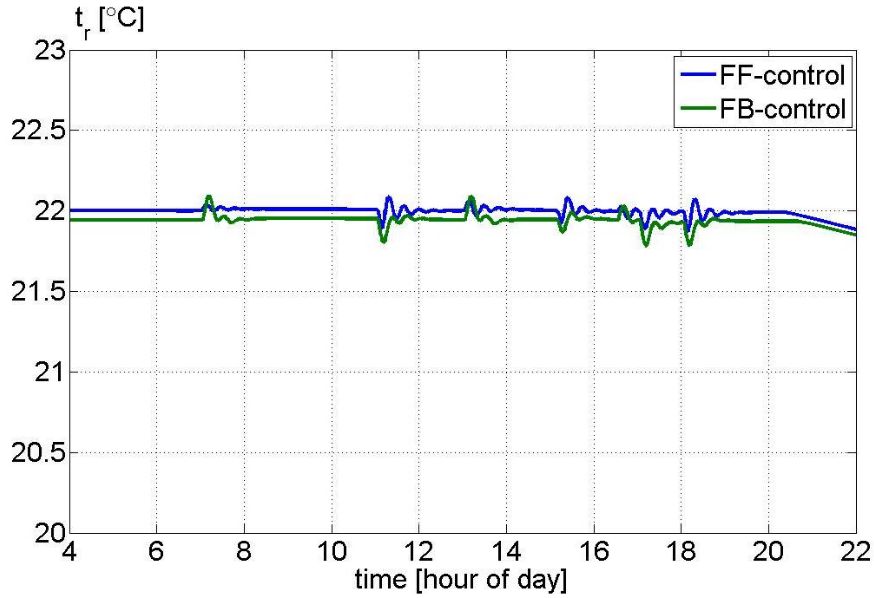


Figure 5.14 Evolution of room air temperature during the simulated 12 h. Comparison between FB control and parameter-based FF control for the office room, summer ambient climate and heavy structure

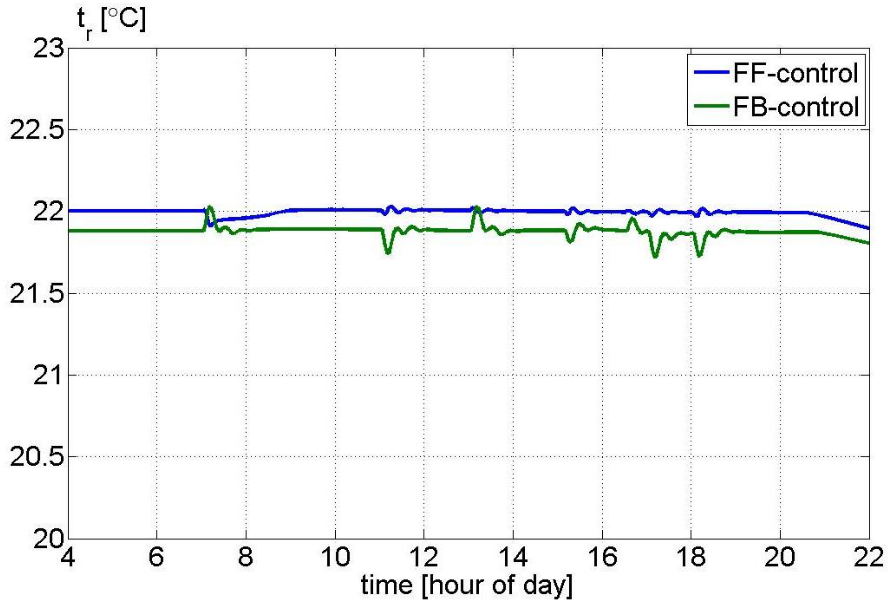


Figure 5.15 Evolution of room air temperature during the simulated 12 h. Comparison between FB control and parameter-based FF control for the office room, summer ambient climate and heavy structure

Light structure ambient summer setup

In the following tables the results for the office room with a light building structure during summer conditions are presented. Table 5.9 present the energy indicators and table 5.10 the peak power indicators. The first row of results in the tables corresponds to the parameter-based FF-controller and the second row to the model-based.

Table 5.9 Final result of step two in FF-control design selection process. Potential of FF regarding energy usage compared to a corresponding FB control system. Test setup: office room, light building structure, summer outside conditions

	Cooling energy savings [%]	Electrical energy savings [%]	Total energy savings [%]
Person counter, electrical power sensor and door opening sensor (IAQ non-dynamic and thermal direct)	2	16	3
Person counter, electrical power sensor and door opening sensor (IAQ non-dynamic and thermal dynamic)	4	17	5

Table 5.10 Final result of step two in FF-control design selection process. Potential of FF regarding power peak reduction compared to a corresponding FB control system. Test setup: office room, light building structure, summer outside conditions

	FCU power savings [%]	AHU power savings [%]	Electrical power savings [%]
Person counter, electrical power sensor and door opening sensor (IAQ non-dynamic and thermal direct)	5	16	35
Person counter, electrical power sensor and door opening sensor (IAQ non-dynamic and thermal dynamic)	7	16	35

Figure 5.16 and figure 5.17 shows the evolutions of the room temperature when the selected parameter-based and the selected model-based FF-controllers are used respectively. In both figure, also the evolution of the corresponding FB-controller is presented.

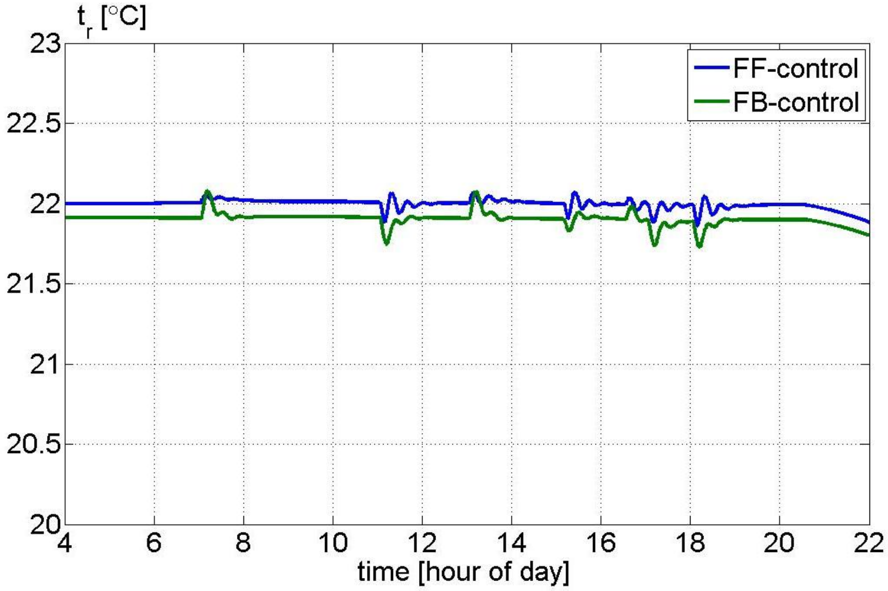


Figure 5.16 Evolution of room air temperature during the simulated 12 h. Comparison between FB control and parameter-based FF control for the office room, summer ambient climate and light structure

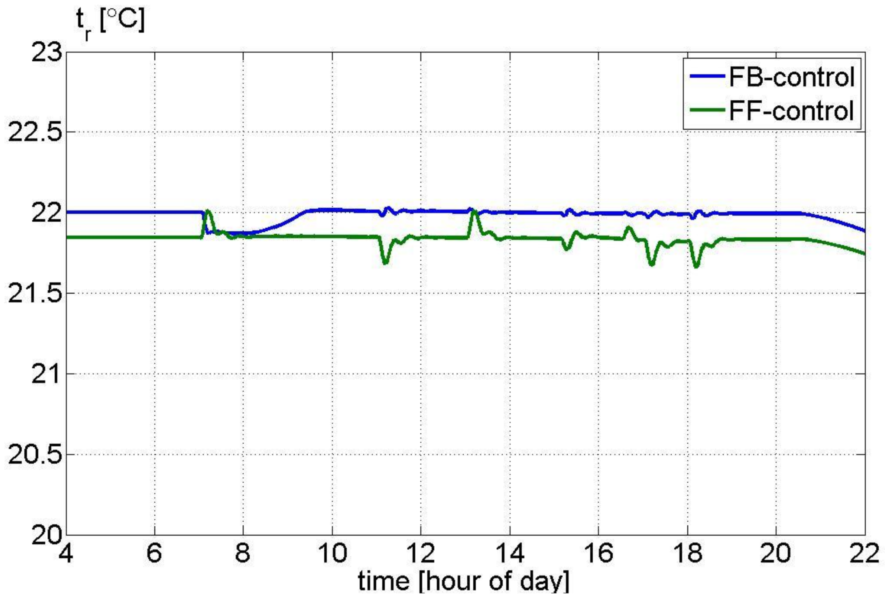


Figure 5.17 Evolution of room air temperature during the simulated 12 h. Comparison between FB control and model-based FF control for the office room, summer ambient climate and heavy structure

Heavy structure ambient winter setup
In the following tables the results for the office room with a heavy building structure during winter conditions are presented. Table 5.11 present the energy indicators and table 5.12 the peak power indicators. The first row of results in the tables corresponds to the parameter-based FF-controller and the second row to the model-based.

Table 5.11 Final result of step two in FF-control design selection process. Potential of FF regarding energy usage compared to a corresponding FB control system. Test setup: office room, heavy building structure, winter outside conditions

	Heating energy savings [%]	Electrical energy savings [%]	Total energy savings [%]
Person counter and door opening sensor (IAQ non-dynamic and thermal direct)	6	18	10
Person counter, electrical power sensor and door opening sensor (IAQ non-dynamic and thermal dynamic)	12	18	13

Table 5.12 Final result of step two in FF-control design selection process. Potential of FF regarding power peak reduction compared to a corresponding FB control system. Test setup: office room, heavy building structure, winter outside conditions

	FCU power savings [%]	AHU power savings [%]	Electrical power savings [%]
Person counter and door opening sensor (IAQ non-dynamic and thermal direct)	0	19	35
Person counter, electrical power sensor and door opening sensor (IAQ non-dynamic and thermal dynamic)	0	18	35

Figure 5.18 and figure 5.19 shows the evolutions of the room temperature when the selected parameter-based and the selected model-based FF-controllers are used respectively. In both figure, also the evolution of the corresponding FB-controller is presented.

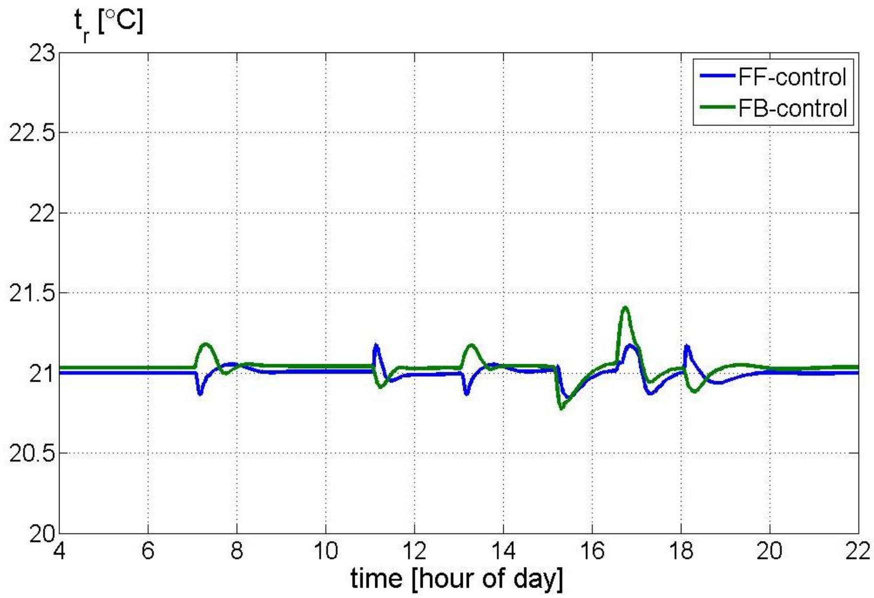


Figure 5.18 Evolution of room air temperature during the simulated 12 h. Comparison between FB control and parameter-based FF control for the office room with a heavy building structure

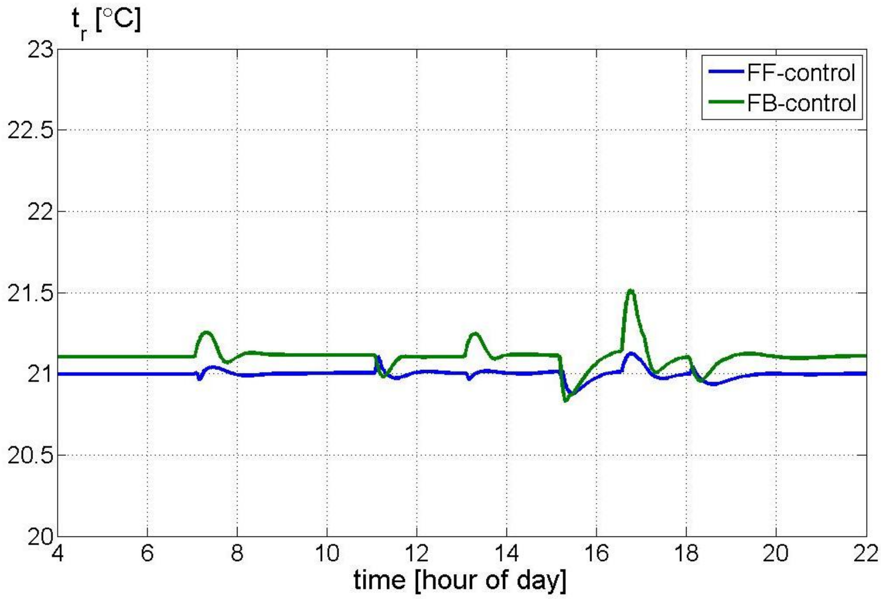


Figure 5.19 Evolution of room air temperature during the simulated 12 h. Comparison between FB control and model-based FF control for the office room with a heavy building structure

Light structure ambient winter setup

In the following tables the results for the office room with a light building structure during winter conditions are presented. Table 5.13 present the energy indicators and table 5.14 the peak power indicators. The first row of results in the tables corresponds to the parameter-based FF-controller and the second row to the model-based.

Table 5.13 Final result of step two in FF-control design selection process. Potential of FF regarding energy usage compared to a corresponding FB control system. Test setup: office room, light building structure, winter outside conditions

	Heating energy savings [%]	Electrical energy savings [%]	Total energy savings [%]
Person counter and door opening sensor (IAQ non-dynamic and thermal direct)	4	17	8
Person counter, electrical power sensor and door opening sensor (IAQ non-dynamic and thermal dynamic)	10	18	11

Table 5.14 Final result of step two in FF-control design selection process. Potential of FF regarding power peak reduction compared to a corresponding FB control system. Test setup: office room, light building structure, winter outside conditions

	FCU power savings [%]	AHU power savings [%]	Electrical power savings [%]
Person counter and door opening sensor (IAQ non-dynamic and thermal direct)	0	19	35
Person counter, electrical power sensor and door opening sensor (IAQ non-dynamic and thermal dynamic)	0	18	35

Figure 5.20 and figure 5.21 shows the evolutions of the room temperature when the selected parameter-based and the selected model-based FF-controllers are used respectively. In both figure, also the evolution of the corresponding FB-controller is presented.

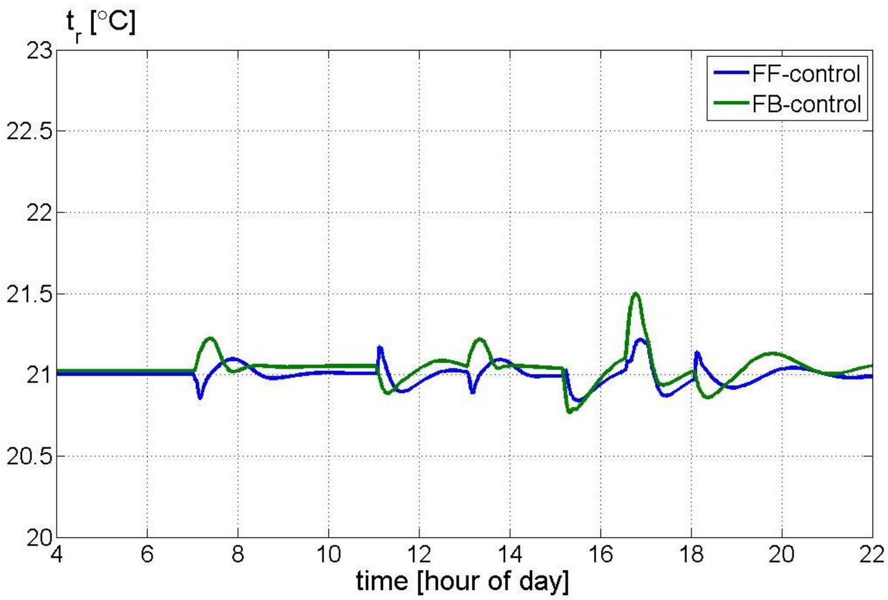


Figure 5.20 Evolution of room air temperature during the simulated 12 h. Comparison between FB control and parameter-based FF control for the office room with a light building structure

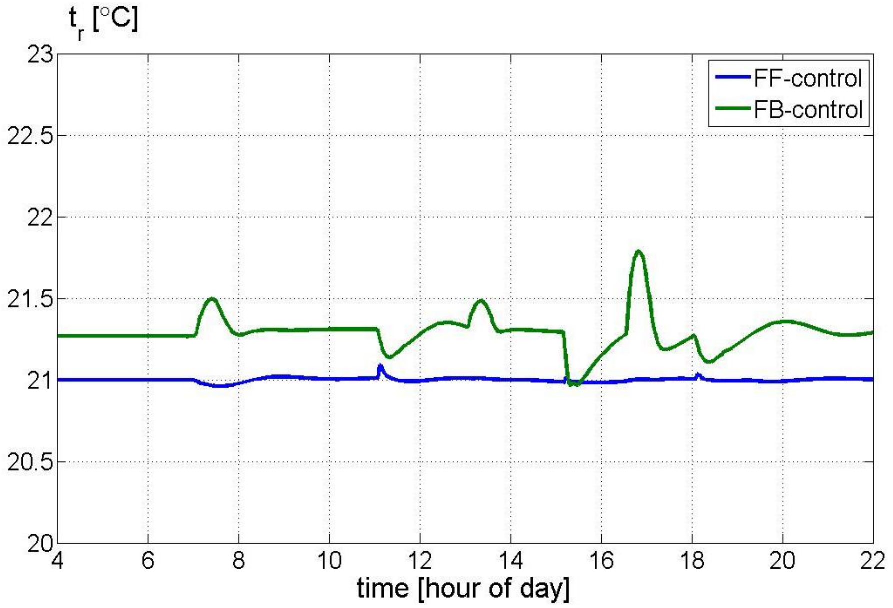


Figure 5.21 Evolution of room air temperature during the simulated 12 h. Comparison between FB control and model-based FF control for the office room with a light building structure

5.3.2.3 Additional results

Below the additional data from the office room setups are presented. As in the meeting room case they consist of the relative mean values of operation. In this case, these data are quite similar from one setup to another. Hence only the setup for heavy structure and heating demand is presented.

Table 5.15 Comparison of mean values of supplied fresh air and thermal energy. Ratio between the FF-control and FB-control systems when applied to an office room in a heavy building structure with heating demand

	Mean supply air flow rate, FF/FB [-]	Mean FCU thermal energy supply, FF/FB [-]	Mean AHU thermal energy supply, FF/FB [-]
Person counter and door opening sensor (IAQ non-dynamic and thermal direct)	0.9	0.9	0.9
Person counter, electrical power sensor and door opening sensor (IAQ non-dynamic and thermal dynamic)	0.9	0.9	0.8

5.4 Summary of results

5.4.1 Meeting room

In the meeting room setups it was shown that the largest energy savings are associated to FF-control of IAQ. The energy savings associated to FF control of thermal climate are on the other hand relatively small but yield a peak power reduction of the FCU.

5.4.1.1 Results

Due to large amplitudes of then occupancy, the setpoint of the FB controller for IAQ needs to be set to about 700 PPM in order to avoid breaching the 1000 PPM boundary. The selected FF-control system, consisting of person counter and non-dynamic filter, could manage the same task with a setpoint of 1000 PPM. The consequence is higher air flow rate in the FB case compared to the FF case.

When the selected FF-control system was used in the meeting room setups, the peak power reduction...

- ...of the AHU was about 45 %
- ...of the central fans was about 70 %
- ...the FCU was between 43 and 16 %

Selected FF-control systems
The selected FF-control system includes a person counter system and an electrical power sensor. By utilizing these two signals, the total energy savings in heavy buildings were about 28 and 34 % for parameter-based and model-based filters respectively. In both cases the IAQ is controlled by a non-dynamic FF-filter. The corresponding result for light structures is 18 and 19 %.

The difference between light and heavy structure is large, but even the light building setups generate large savings. This means that the largest part of the savings is related to the IAQ control since the energy savings contribution of the electrical power sensor is zero in the light building case.

Electrical power sensor
Even though the electrical power sensor was included in the final FF-control system, its contribution is questionable. The only time in this study, when an electrical power sensor alone resulted in savings that are worth mentioning, was in the heavy structure meeting room setup. Then the energy savings were about 5 %. Also the power peak reductions of the FCU were relatively large in this setup; about 27%. However, the results from this setup can be seen as close to the potential maximum and these results should not be regarded as easily attainable.

Motion sensors
In the meeting room setups, also a motion sensor was used as a FF-sensor. By using the most favourable filter “block”, the setpoint of the FB-controller could be increased to about 800 PPM until the limit was breached. The total energy savings when the motion sensor was used for IAQ control were about 17 and 8 % for heavy and light structures respectively. However, when the same filter was used

also for thermal control, the savings were reduced to about 16 and 6 %. Also the peak power was then affected negatively.

5.4.2 Office rooms

Compared to the meeting room setups, yet three disturbances were analysed from a FF point of view;

- Outdoor temperature
- Solar heat gain
- Door opening

In the selection process both the outdoor temperature and the solar heat gain was rejected; the outdoor temperature in step one and solar heat gain in step two. This is further discussed in section 5.5.2.1.

5.4.2.1 Results

In the office room setups, the amplitude of the occupancy is relatively small but the room contain on the other hand less air which means that the controllers must act relatively fast in order to avoid breaching the 1000 PPM limit. For the FB-controller, a setpoint of 750 PPM was required. The selected FF-control system, also in this case consisting of person counter and non-dynamic filter, could manage the same task with a setpoint of 1000 PPM.

When the selected FF-control system was used in the office room setups, the peak power reduction...

- ...of the AHU was between 20 and 16 %
- ...of the central fans was 35 %
- ...the FCU was between 10 and 0 %

Summer ambient climate
The selected FF-control system includes in this case a person counter, an electrical power sensor and a door opening sensor. By utilizing these signals, the total energy savings in heavy buildings were about 4 and 7 % for parameter-based and model-based FF-controllers respectively. The corresponding results for light structures were about 3 and 5 %. Hence, the savings have been drastically reduced compared to the meeting room case.

Winter ambient climate
In the office ambient winter setups, the electrical power sensor was rejected from the selected parameter-based FF-control system. The performance was actually reduced a bit when this sensor was included.

The selected parameter-based FF-control system includes in this case a person counter and a door opening sensor. The final results shows that by utilizing these signals, the total energy savings were about 10 and 8 % for heavy and light structures respectively.

The selected model-based FF-control system also includes the electrical power sensor. In this case, the total energy savings were about 13 and 11 % for heavy and light structures respectively. Hence, the savings are higher compared to the office summer setups due to a more energy intense air-handling. In the winter setups, the FCU power peak reductions are close to zero.

5.5 Conclusions and discussion

Below, the results from the previous section are discussed further. The aim is to analyse the effect of utilizing FF-control in the current application and to determine the influence of the setup condition parameters.

Some general conclusions that are more or less independent on condition parameters of the setups have been identified. These have been structured in three different groups as presented below. In the upcoming text, these conclusions are further explained and analysed together with the most important setups dependent results.

1. Selection of FF-filters
    - For IAQ control, the non-dynamic filter was proven most favourable
    - For thermal control, the most preferable parameter-based filter is the direct one and the most preferable model-based filter is the dynamic one

2. Disturbance aspects
    - Internal disturbances have larger potential for FF-control than external disturbances
    - In general, the more internal disturbances that are included, the higher is the potential regarding both energy usage and power peak reductions
    - The FF-model for how to treat an open door can just as well be replaced by on/off control of the ventilation
    - The part of the door opening sensor for IAQ control should preferably be complemented with a person counter
    - The thermal part of the door opening sensor has no effect

3. Energy and peak power savings potential
    - There is a trade-off between maintaining the setpoint, energy use and peak power.
    - The largest energy savings are related to FF control of IAQ by taking internal disturbances into account
    - The energy savings related to FF control of thermal climate by taking internal disturbances into account are small
    - The overall savings are in general larger in heavy building structures
    - The overall savings are in general less in office room than in meeting rooms
    - The energy savings are larger during winter than during summer conditions
    - The peak power savings are larger during summer than during winter

The following text is focusing on one of these general conclusions at the time. Even though these conclusions are more or less valid throughout the entire study, some exceptions have been found. Further, the effects that have been leading up to the conclusions and that have been used to identify the exceptions are more evident in some setups than in others. In the following text, these effects are therefore discussed in relation to the setup in which they are most prominent in the results.

5.5.1 Selection of FF-filters

5.5.1.1 IAQ control

Evident in all setups
For IAQ control, the non-dynamic filter was proven most favourable independently on the setup. This effect is a result of that the dominant part of the dynamics of the ventilation system is transport-delays which directly can't be compensated for in FF-control.

By removing the dynamics of the room in the FF-filter (which is done in the non-dynamic filter case), the FF-control becomes faster which in one sense is a contra-action to the time-delays. The increased performance might not be obvious since the time-constant of the room is much larger than the time-delays in the ducts. But one must remember that the dynamic filter is continuously changing its control signals and since each of them are delayed, the system is out of sync during the entire control horizon. This means that the most resent control signal is as delayed as the prior ones, which in turn means that the disturbance rejection is aggravated compared to the non-dynamic filter.

When a disturbance is measured, the non-dynamic filter immediately calculates and sends the steady-state supply air flow rate that corresponds to a room CO_2 concentration of 1000 PPM. This means that the initial supply flow of the non-dynamic filter is larger compare to the supply flow of the dynamic filter. But this also means that the disturbance rejection is improved. Hence, the main reason to why the non-dynamic filter is more favourable is that the response of the room is better dampened by the control system. In turn, this means that the actual value of the CO_2 concentration can reside closer to the limit without crossing it and that the mean supply air flow rate can be decreased.

5.5.1.2 Thermal control

Evident in almost all setups
The conclusions regarding which type of filter that is most favourable for thermal control were more or less consistent over the entire study. Among the parameter-based, the direct filter was most favourable, and among the model-based, it was the dynamic filter.

The selection of the dynamic model-based filter is explained by that this filter is the one most similar to the room. This means that the match between supply and demand is high. When it comes to the parameter-based filters, the direct filter is actually more unlike the room than the dynamic direct one. The selection of the direct filter is instead motivated by its fast response (c.f. the selection of the non-dynamic filter for IAQ control previously discussed). The direct filter results in better disturbance rejection features which are particular awarding when it comes to energy usage.

Exception, evident in office, light and heavy, summer setup
One exception occurs in the heavy and light office setup during summer. In this setup, the most favourable model-based filter for the solar heat gain sensor was the static one. However, in step 2, it was shown that the positive effect of this sensor was negligible. For that reason, this exception is not further analysed.

5.5.2 Disturbance aspects

The results show that the character of the disturbance has a large influence of its potential in FF-control. <u>Disturbances with high frequencies and amplitude have large potential while disturbances with low frequencies and amplitude have low potential.</u>

The reason is that the main advantage of using FF-control is a more effective disturbance rejection which in turn results in that the actual values of the room can reside closer to the control limits without breaching them. Hence, energy usage is reduced. This means that the potential of saving energy and reducing peak power by using FF are large in systems that are affected by disturbances with high amplitudes.

When also the disturbances have a high frequency, the total positive effect over a certain period of time is increased. Frequently occurring disturbances with large amplitudes results in that the potential of saving energy and reduce the peak power by using FF is large.

5.5.2.1 Internal disturbances have a larger potential than external disturbances

The difference in potential between internal and external thermal disturbances was indirectly explained in the text above.

Evident in all setups
The heat generated by internal disturbances is partly or entirely transmitted to the room air. The part transmitted to the air will result in a fast response of the room while the part transmitted to the structure is suppressed. This means that disturbances that that to a large extent directly affect the indoor air are more favourable to include in the FF-control.

Since the external disturbances are entirely transmitted to the structure, the corresponding response of the room is slow. A FB-controller is fully capable of dealing with these types of disturbances which means that the potential of using FF becomes small. This is of course only relevant for thermal disturbances since the emission loads are not dampened by the building in the same manner.

The conclusion is that the highest performance is achieved when the FB-part is assigned the external disturbances and the FF-part is assigned the internal ones. In such control system, the FB-part is managing the base load and the FF-part is managing the intermittent variations around this load. The control system can both be fast and accurate when this structure is used.

5.5.2.2 In general, the more internal disturbances that are taken into account, the larger is the potential

Evident in all setups
As indicated above, all of the internal disturbances are favourable to utilize in the control system. Generally...

- ...by using FF for IAQ control, both the energy usage and the peak power is reduced
- ...by using FF for thermal control, the peak power is reduced

The results also show that the more internal disturbances that are a taken into account, the higher is the performance of the FF-controller. It was also shown that the difference between FF-controllers <u>only</u> based on internal disturbances and FF-controllers based <u>on both</u> internal and external disturbances is negligible.

Evident in office room setups
In the office room setups, it was shown that the performance might be aggravated when <u>only</u> external disturbances are taken into account.

The problem is that the external disturbances are usually a small part of the total set of disturbances that are acting on the room. Hence, if only external disturbances are taken into account, the supply of the FF-controller becomes mismatched with the demand. Then, the FB part must both compensate for internal disturbances and the mismatched supply of the FF-controller.

As an example, imagine that only the outside temperature is included in the FF-part. From the FF-point of view, that is the only disturbance acting on the room. Dependent on the outside temperature, the FF will constantly generate heating or cooling to the room by not taking other disturbances into account. This means that during most times the supply will not match the demand which can have a negative effect on the performance. It also means that a larger part of the control is put on the FB part.

Exception, evident in office room winter setup
By comparing the results from the office heating demand setups with the results from the office cooling demand setups, it was concluded that the thermal control performance is in general more sensitive to FF-filter modelling errors in the heating demand cases.

- In the cooling demand cases, both the signals from the FF- and FB-part are used to increase the cooling supply of the FCU which means that the two controllers are working together.
- In the heating demand cases, on the other hand, situations in which the two parts are counter-acting each other might occur since the signals from the FF-part are used to reduce the heating supply from the FCU.
- This means that if the FF part induces errors, it is more likely that the control performance is aggravated since the FB-part must also correct the errors of the FF-part.

This sensitivity explains a number of differences between the heating and cooling demand cases.

- The sensitivity is mainly reflected as an increased relative peak power of the FCU when single disturbances are incorporated. In those cases, the overall picture is not given to the FF-controller which means that the output is mismatched with the demand. To compensate for the error of the FF, the FB-part is increasing the control signal. However, when the disturbances are reduced, the contribution of the FF-part is rapidly decreased. Due to the time-delays in the FB-loop, the large output signal is maintained for a while which results in a large power peak.

The sensitivity is also the reason why the electrical power sensor has been left out in the final parameter-based FF design in the office winter setup. This sensor is on the other hand used in the final model-based filter:

- The ideal situation in a heating demand case is that a thermal disturbance should be exactly replaced by an equal reduction in heat supply. Hence, the control should ideally be very fast. The parameter-based filters are approaching this scenario when single disturbances are considered. This means that the temperature set-point can be moved closer to the boundary of the dead-band. However, these filters does not include how large part of the heat emission from the disturbance that is directly absorbed by the wall. When more disturbances are included, the error is increasing in the sense that the reduction of heat supply becomes too large and the control is aggravated.
- The model-based filters are, on the other hand, based on a heat balance of the room. This means that an increasingly number of disturbance taken into account does not ruin the control but actually improves the performance. For single disturbances, however, the individual disturbance gets too much weight and the supply does not match the demand.

5.5.2.3 Aspects of the door opening sensor

Evident in office room during summer climate setups

The opening of the door are effecting both on the thermal climate and IAQ of the room. But it was only the IAQ part which had a mentionable effect in the FF-control.

The reason is that the opened door always had a cooling effect on the room, i.e. a sink effect on the room temperature. In the cooling demand cases, the purpose of the thermal FF-controller is to limit the overshoots of the upper temperature dead-band boundary, which means that the thermal part of the door does not improve the disturbance rejection features. The small energy savings visible in the results derive instead from that the FF-controller immediately is decreasing the FCU cooling supply when the door is opened. When instead a FB-controller is used, this is done with a time-lag.

Evident in office room setups

An interesting part of the results is that the IAQ part of the door opening sensor alone gives a moderate contribution to the savings in general. This is also true in the office room setups for the person counter, due to relatively low frequencies and amplitude of the occupancy. However, when these two sensors are combined the savings becomes greater than their individual contributions together.

The reason is that the office rooms are occupied during the time when the door is opened. If the IAQ control is managed by an FF-controller that only includes a person counter, the same amount of fresh air will be supplied also when the door is opened. But when the door is opened, the need for fresh air is dramatically decreased which means that the room becomes over-ventilated. This is not the case for a FB-controller since the generated fresh air is based on CO_2 concentrations which drop when the door is opened.

This error in the FF-part cannot be compensated for by the FB-part since its minimum contribution is zero. Hence the performance of the FF-controller is reduced compared to when the IAQ control is managed by a FB-controller. When the door opening sensor also is included in the FF-control system, the supply air flow rate is decreased during the time the door is opened and the relative savings are increased.

Evident in office room setups
An FF-filter that contains a complete model of the actual flow rate through the door can be replaced by a model that shuts off the ventilation when the door is opened. This is of course dependent on the temperature difference over the door as well as the CO_2 concentration in the infiltrated air. However, these results give a clear indication of the potential of decreasing the complexity of the corresponding FF-filter.

5.5.3 Energy and peak power savings potential

5.5.3.1 Trade-off between maintaining a setpoint, energy use and peak power

Evident in all setups
There is a trade-off between maintaining the room setpoints, i.e. the level of disturbance rejection, and the savings potential. This means that it costs more energy and more power to maintain a more stable control than to have a control with large amplitudes as responses to disturbances. However, an important point to make is that the indoor climate must not be jeopardized in order to reduce the energy usage. To include this aspect in the evaluation, the two comparability criterions, i.e. the same degree hours outside the dead-band and the same CO_2 peak, were used. These are punishing controllers that results in large responses of the room and favours the controllers with an efficient disturbance rejection.

Evident in meeting room setups
The effect of the interplay between the trade-off and the comparability criterions can be visualized by comparing the performance of model- and parameter-based FF-controllers for thermal control in the meeting room setups. For example, the performance indicators of the FF-controllers that only include an electrical power sensor, presented in table B.2 and table B.4 in appendix B, can be compared. The disturbance rejection feature is better for the model-based FF-controller. This means that the temperature setpoint of the FB-controller that corresponds to the

model-based FF-controller must be lower than the one that corresponds to the parameter-based FF-controller in order for the thermal criteria of comparability to be fulfilled. In turn, this yields a larger thermal energy saving potential in the model-based case but at the same time the required peak power of the FCU is increased.

On the other hand, the model-based filters also include the cooling power of the ventilation system. When also the IAQ control is managed by the FF, the response to an emission load is close to a step in supply air flow rate which also results in a step in corresponding cooling power. Since this is taken into account in the thermal control when model-based filters are used, the initial cooling power supply signal from the FF to the FCU can be reduced by the same amount. This favours the FCU peak power savings of model-based filters compared to parameter-based filter. This can be seen in table 5.2.

5.5.3.2 Largest energy savings related to FF control of IAQ by taking internal disturbances into account

Evident in all setups
The definitely largest gain of FF in terms of electrical energy savings is associated with control of IAQ. These savings originates from that the control task can be managed with a lower supply air flow rate compared when FB-controllers are used.

As an example, in table 5.5 and table 5.6, the ratio between mean supply air flow rate generated by FF and FB is shown for meeting room setups. The FF-control can manage the control task by only supplying half of what is required when a FB-controller is used. This effect is due to the relatively fast dynamics of the CO_2 level in the room, which means that the system response in the FB case results in large amplitudes. Hence, a low CO_2 setpoint is needed to avoid breaching 1000 PPM. Due to the disturbance rejection features of the FF-controllers, the actual value of CO_2 can reside very close to 1000 PPM and still avoid it to be breached when a disturbance enters the room.

In table 5.15, the corresponding operational mean values for the office room setups are presented. It is clear that the potential of using FF for IAQ control is lower in these cases. This is due to smaller amplitudes of occupancy. Hence, the occupancy profile of office rooms is more favourable from an FB point of view which also is discussed further on.

Also, the largest part of the cooling energy savings originates from FF-control of IAQ. Due to the relative high supply air flow rate of the FB controlled system, the energy usage for air-handling becomes a very large part of the total energy usage. This also means that a large part of the energy for thermal control of the room comes from the supply air in the FB case. Since the signal that determines the supply air flow rate only is dependent on the need of fresh air to the room, this part of the cooling energy has a poor match to the actual demand for cooling. <u>By reducing the supply flow rate as in the FF case, a larger part of the cooling energy is **shifted** from the AHU to the FCU whose supply is directly dependent on the actual demand.</u>

5.5.3.3 Small energy savings related to FF control of thermal climate by taking internal disturbances into account

Evident in meeting room setups
The energy savings that are generated by utilizing FF control for thermal control are relatively small. The reason is that the thermal properties of the room are well suited for FB-control due to large time-constants. Thus, the relative savings of FF-control becomes low.

On the other hand, the results also show that the peak power of the FCU can be reduced when FF is used for thermal control. When the FF-controllers in this work are managing the thermal control, the thermal power supply is closely related to the demand throughout the entire duration of a disturbance. On the contrary, the thermal power supply generated by FB-controller is initially small and grows as the actual room temperature evolves further away from the setpoint. In most setups, the total supplied thermal energy is more or less the same regardless if FF or FB is managing the thermal control. But due to the skewness of the FB-supply the peak power of the FCU is larger in FB case.

5.5.3.4 Larger savings in heavy than in light buildings

Evident in most setups
The electricity savings potential by using FF for IAQ and thermal control were about the same in the heavy and light building structures setups. The reason is that the largest part of the electricity savings derives from the effects of using FF for IAQ control which is unaffected by the building structure. The small differences seen in for example table 5.3 and table 5.7 originate from different operations of the integrated fan in the FCU.

On the other hand, the potential of saving cooling energy by using FF for IAQ and thermal control were in general lower in the light building structure case. This is mainly due to two effects;

- Primary, this is explained by the type of disturbance profiles that were used in the selection process. A period of activity, which is characterised by large disturbances, is always preceded by a period with small disturbances. The resolution of the disturbances is one hour which means that between two active periods there is at least one hour with low activity. Due to the difference in heat storage capacity of the walls, the room temperature will sink more during the inactive period in the light structure case than in the heavy case. This means that even though the response in temperature due to a disturbance is larger in the light case, the resulting overshoot of the dead-band is larger in the heavy case due to the higher temperature prior to the disturbance. Hence, the setpoint of the FB-controllers has to be a bit lower in the heavy building case in order to fulfil the criteria of equal degree hours outside the dead-band.
- Another reason is that the cooling power which is generated by an FB-controller as a response to a change in occupancy is closer to the demand in the light building structure case than in the heavy building structure case. This means that the relative potential of FF-control is reduced. This is explained by that a larger part of the cooling supply is put on the supply

air in the FB-case. When the occupancy is increased, the supply air flow is increased to maintain the IAQ which means that also the cooling supply is increased. Since the thermal inertia of the light building is smaller than in the heavy, the match between the demand for fresh air and the demand for thermal power is better matched in the light case.

Exception, evident in office winter setup

As indicated before, the energy savings potential is connected to the disturbance rejection ability of the control system. If the disturbance rejection level is high, so is the energy savings potential. According to the previous section, the disturbance rejection ability, and thus also the energy savings potential, is larger in heavy building than in light ones.

One exception is visual by comparing the figures 5.19 and 5.21 (these figures show the resulting evolution of room temperature when the selected model-based FF is used in the setups consisting of heavy and the light offices during winter). These figures clearly show that the disturbance rejection in the light structure case is higher than in the heavy structure. This contradicts the results from the cooling demand setups, even though the energy savings potential is slightly higher in the heavy building case.

The more efficient disturbance rejection in the light case is an effect of the delimitation that the FF-part only is allowed to reduce the heating supply from the FCU. That is, the FF-part is not allowed to increase the heating supply which limits the disturbance rejection features in the heavy building case. This is especially visible in figure 5.19 and 5.21, by comparing the impact of the door which is opened between 15:30 and 16:30, since the door has a cooling effect on the room. In order to limit the impact of the door, the heat supply needs to be increased. This is achieved in the light building case while the impact is relatively large in the heavy case.

The reason is that prior to the opening of the door, the output from the FF is larger in the light case than in the heavy case. That is, the FF-part is suppressing the FB-part more in the light case than in the heavy. This depends on that the building has a larger cooling effect in the heavy structure case; even though the door opens quite late in the day, which means that the structure has been more or less continuously heated by internal disturbances, the temperature of the heavy structure is still relatively low due to the large thermal inertia. The FF-controller is therefore estimating a larger heating demand which means that its output becomes smaller.

Due to the initially higher output of the FF in the light structure case, more heating power can instantly be released by the FF in the light structure case by decreasing the output. However, in the heavy structure case, the output of the FF becomes saturated to zero when the door is opened which leads to that part of the increased heating supply have to be managed by the FB-control. This effect appears from time to time during the simulation which is the reason for the difference in disturbance rejection. This is also explains why the savings potential difference between light and heavy structure setups is less in the heating demand setups than in the cooling demand setups.

5.5.3.5 Larger savings in meeting rooms than in office rooms

Evident in all setups
In general, the largest potential of FF was shown in meeting room cases. The reason is that all disturbances regarded in meeting room setups have a large potential for FF-control, i.e. disturbance with a relative high frequency and amplitude.

In the office case, the disturbances are smaller both regarding amplitude and frequency. The heat gain from equipment is more or less constant and the amplitude of occupancy has a value of one. Furthermore, the influence of external disturbances is much larger. These prerequisites suites the FB-controllers better. Hence, the relative savings is decreased both due to better performance of FB-controllers and due to a smaller contribution from the FF part.

5.5.3.6 Larger energy savings during winter than summer

Evident in office room during winter ambient climate setups
What is noticeable in the winter setups is that the energy for air-handling is dominant. The reason is that the required temperature rise of the outside air is much larger than the correspondent temperature reduction in the summer setups. Related to this, three conclusions have been made;

- The energy intense air-handling means that the shift of thermal energy supply, from AHU to FCU, which is induced by using FF for IAQ control, becomes even more favourable in the winter setups. This effect is visible in the result by comparing the energy savings for office winter and office summer setups when the selected FF-control system is used for thermal and IAQ control (for example table 5.7 and 5.11).
- However, in order to increase the energy saving during winter by using FF for IAQ control, both the door opening sensor and the person counter must be included. As discussed in section 5.5.2.3, the performance of the FF control system is dependent on that both of these sensors are included. This is even more important in the winter setups due to the more energy intense air-handling operation. The results show that the energy usage in some cases actually was increased when a FF-controller only including a person counter was used instead of a FB-controller (see table B.13 and B.22 in appendix B). Hence, by not reducing the supply air flow rate during the time the door is opened has a large negative effect on the energy savings.
- The shift of thermal supply from AHU to FCU is favourable as long as the air-handling in the AHU is more energy-intense than the air-handling in the FCU. That is, as long as the temperature drop or rise over the AHU is larger than the correspondent drop or rise over the FCU. When these are approaching each other, a large part of the energy savings related to FF-control of IAQ will approach zero.
- A comment related to the three points above is that the central cooling of supply air only includes the energy for dry cooling in this work. By also including the latent part, the differences in energy savings between summer and winter setups might be smaller.

5.5.3.7 Larger peak power savings during summer than winter

Evident in office room during winter ambient climate setups
One effect of ambient climate is that the potential of power peak reductions of the FCU is much smaller in winter cases than in summer cases.

As mentioned, in the heating demand cases, the FF is limited to only suppress the heat supply induced by the FB-system. This means that also the peak power saving potential of the FCU is limited. The reason is that it is unlikely that the power peak will occur at the same time as a disturbance takes place, i.e. when the FF has the ability to reduce the power supply of the FCU. Instead, the power peak occurs when the internal heat generation are the smallest, that is, when the heating demand is the largest. This happens in the end of the simulation when there are no internal disturbances which mean that the contribution from the FF-part is zero. This results in that the power peak reduction is close to zero in all heating demand cases. On the other hand, the reduction during operation is of about the same magnitude as the reduction in cooling demand cases. But this is of no interest when it comes to the design capacity of the FCU.

6 Local control tasks

This part of the work aims to further test the selected FF-controllers from chapter 5. The purpose is to determine the combinatorial and individual gain of controlling local supply of heat, cooling and air with FF in a simulated environment more similar to real office buildings. The output of this study is which of the services provided by the HVAC-system that preferable are controlled by FF.

The simulation-platform consists of the multi-zone system described in section 3.4.2 which represents part of a building floor in an office building. Each room, except for the corridor, is equipped with its own local control system that determines the fresh air supply through the diffuser and the power supplied by the FCU.

In this study, this local control system can either consist of FB-controllers or FF-controllers. The FF-controllers are evaluated by comparing the differences in performance using the energy and peak power indicators presented in section 1.3.3. In the FB case, the CO_2 and room air temperature is measured to determine the control signals. In the FF case, the disturbances acting on the room is measured to determine the control signals from the FF-filter.

The simulation time is expanded to reassemble a work-week consisting of five days and nights of office related activities in one sequence. The setpoints of the controllers are the same for night and day. Hence, no specific night-mode is considered in this study which means that the ventilation eventually is shut off during the night and that the temperature in the beginning of a working day will have a value between 21 and 22 °C.

6.1 Method

The evaluation is performed according to the methodology presented in section 1.3. That is, to evaluate the FF-controllers by comparing them to FB-controllers regarding the energy and peak power indicators. This methodology was also used in the second step of the evaluation process in chapter 5. In that case, both criterions for comparability were used. That is, in order to compare the performance of two control system they must result in equal CO_2 peaks and equal degree hours outside the temperature dead-band.

New criterions for comparability
As mentioned, the criterion on CO_2 concentration, i.e. maximum 1000 PPM, is based on a Swedish guideline on IAQ presented in BBR[3]. In chapter 5 it was shown that FF-control is preferable since the CO_2 setpoint can be increased compared to the FB-controllers without violate this limit.

The criterion on thermal climate is instead based on a guideline for thermal comfort stated in Belok[10]. In chapter 5, it was shown that the FF-controllers resulted in a more stable thermal control than FB-controllers. However, it was also shown that even if the temperature dead-band levels were used as setpoints for the FB-controllers, the resulting room temperature fulfilled the guideline in Belok. This means that from a thermal climate point of view, there is no need for

FF-control in office buildings since the FB-controllers fulfil the requirement of a desirable thermal climate.

For that reason, the criterion of equal degree hours outside the dead-band is dropped in this study. This means two control systems from now on are considered as comparable, and are therefore referred to as corresponding, if none of them results in that the CO_2 concentration limit of 1000 PPM is exceeded. This is achieved by inheriting the setpoints from the study in chapter 5.

Evaluation method

Also this study is performed by constructing test setups to analyse the influence of different condition parameters on the potential of FF-control. In general, the same condition parameters are used in this study as in part 2 of chapter 5. That is, the influence of building density (heavy or light) and the effect of ambient conditions (summer or winter). However, since nine office rooms and one meeting room are simulated together in this part; the condition parameter regarding type of room is dropped.

The evaluation of the test setups is performed by comparing the selected FF-controllers from chapter 5 with corresponding FB-controllers in pairs. For each setup, the FF-control system with the highest performance regarding the energy and peak power indicators was selected.

One large difference compared to chapter 5 is that the setups can no longer be treated as pure heating or cooling cases. This is partly due to that the night-time is included which is characterized by zero internal heat gains and relatively high thermal losses due to relatively low outdoor air temperatures. Hence, depending on the structure and ambient conditions, there might be a heating demand during the night even though there was a cooling demand during the day. Furthermore, since many rooms are included in the building there might be both heating and cooling demands at the same time, even during the day. The reason is that the disturbances are unevenly distributed over the building since the total occupancy factor never is 100 %. This is discussed in the following section.

The simultaneous demand of both heating and cooling from a building point of view makes it possible to evaluate the potential of using FF-control for heating, cooling and air supply both individually and in combinations. This is realised by treating the type and number of FF-controlled HVAC components as a parameter in the test setups. The evaluated parameter combinations are;

- FF-control for heating, cooling and air supply
- FF-control for heating and air supply
- FF-control of air supply

In the two latter cases, FB-loops are used to control parts of the HVAC-system not handled by the FF-controller.

As discussed in section 1.3.3, in cases like this, when both heating and cooling is required during the same setup, the peak powers of the HVAC-components for thermal supply refers to the dominating thermal demand. That is, to heating during winter setups and to cooling during summer setups. Regarding the energy indicators, one indicator is associated to cooling energy and one is associated to heating energy. The total energy indicator consists of both energy for heating and cooling as well as of electricity usage.

6.1.1 Disturbance profiles

As was showed in chapter 5, the profile of the disturbances has a large effect on the potential of FF. The differences in frequency and amplitude of the disturbances partly explained the differences between meeting and office room setups. In this study, also the effect of disturbance distribution is introduced since many rooms are simulated together.

Internal disturbances
Instead of more or less arbitrary disturbance profiles as in chapter 5, measurement data of occupancy presented by M.Maripuu[45] is used in this part. The data is based on occupancy recorded during all working days over one year in an office building located in Gothenburg, Sweden. The building is used by administrative staff at a University and consists of 58 office rooms and 5 meeting rooms.

The result is shown in figure 6.1 and consists of the occupancy factor as a function of time during a working day. The figure was used to determine the total number of people that were present during the hours of the day and the distribution was then set arbitrary. According to the M.Maripuu, one general function is not enough to describe the dynamics of the occupancy over an entire year. Instead three distinguished cases are identified; days with high, low and medium occupancy density. The study was carried out by identifying the simulated days with these functions; one day is simulated with high density, one day with low and three days with medium density.

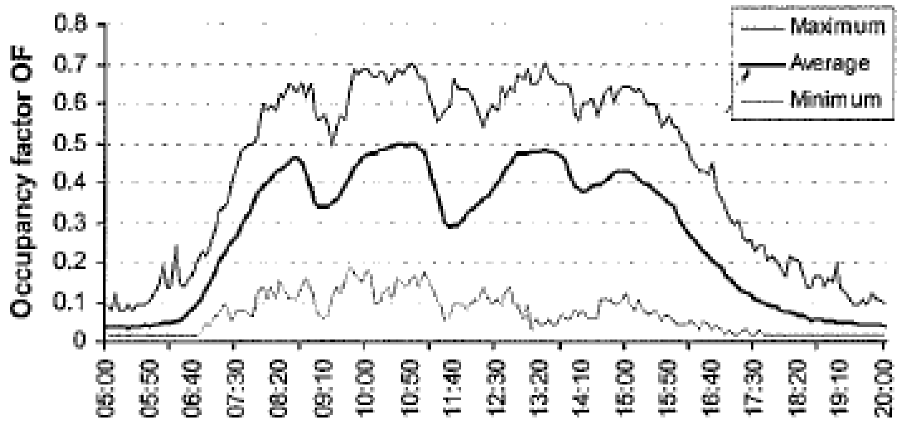


Figure 6.1 Occupancy factor as a function of clock-time during one working-day derived by an annual measurement of occupancy in an office building in Gothenburg , Sweden[45]

The door openings of the office rooms were simulated by assigning each door a value that corresponds to the ratio between the time that a door is opened and the total time that the corresponding room is occupied. Three different values of this ratio was used; 1, 10 or 20 %. For each simulated working day, these ratios are evenly distributed among the office room. The time that the door to the meeting room is opened was regarded as negligible and the door to an empty room is always closed.

The rest of the internal disturbances are managed in a similar way as in the single-zone cases of chapter 5;

- Lighting in the office and meeting rooms corresponds to 10 W/m^2, and is on when the room is occupied.
- Lighting in the corridor corresponds to 280 W and is on during the working hours and off during the nights.
- The heat emitted by equipment corresponds in the office room cases to 100 W during the working day
- The heat emitted by equipment corresponds to 50 W/person in the meeting room case.
- Each person emits 70 W of sensible heat and 18 l_{CO_2}/h.

External disturbances

The external disturbance profiles are also in this case based on climate data from the city of Helsingborg in Sweden. These are presented below, as a function of simulated hours for summer and winter conditions respectively, and consists of data for five coherent days and nights. The time zero in the figures corresponds to a clock time of 2:00 and the occupied period stretches from 7:00 to 19:00.

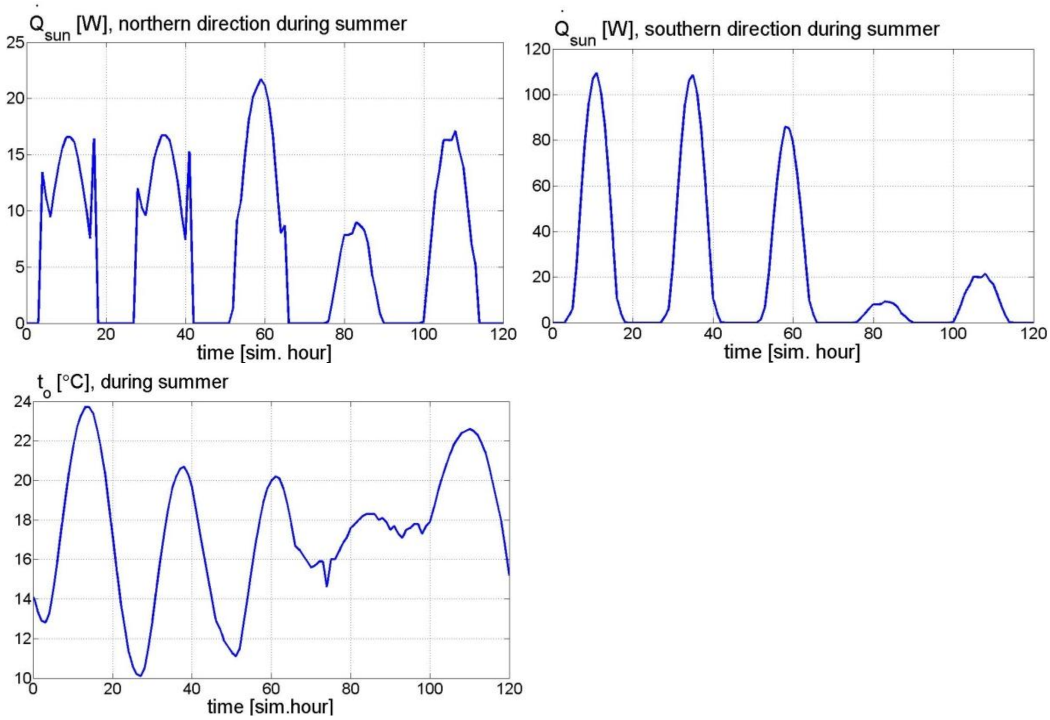


Figure 6.2 External climate data of summer conditions. Solar heat gain in north direction, solar heat gain in south direction and OAT respectively

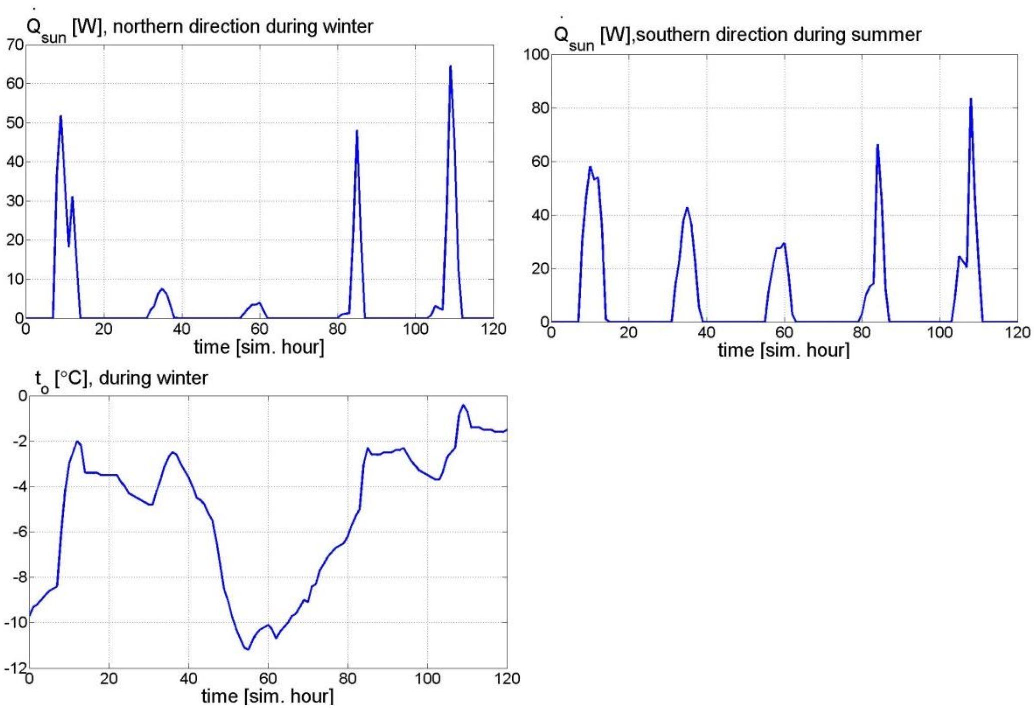


Figure 6.3 External climate data of winter conditions. Solar heat gain in north direction, solar heat gain in south direction and OAT respectively

6.2 Results

In this chapter the selected FF-controllers from chapter 5 was further tested. The purpose was to determine the combinatorial and individual gain of controlling local supply of heat, cooling and air with FF. The results of this study shows which of these services provided by the HVAC-system that preferable are controlled by FF.

6.2.1 Overall results

Throughout this study, it was shown, that when the criterion of equal degree-hours outside the temperature dead-band was dropped, the gain of controlling heating and cooling with FF is small. In most setups, the performance of the FF control system dedicated only for IAQ control had the same or even higher performance compared to the cases when also the thermal part is handled by FF. For that reason, it was concluded that the thermal part should preferably be managed by a FB-controller and the IAQ part by a FF-controller. This means that the FF control systems that are selected from this study consist of a person counter and a non-dynamic FF-filter.

6.2.2 Setup specific results

The main body of results from this chapter are presented in appendix C and in the following text, the primary results are presented. These primary results consist of the energy and peak power indicators of the selected FF control system from each setup. The text below is divided between the setups consisting of heavy or light building structures. The indictors, which originally were presented in chapter

1.3.3, are presented in tables and each table refers to both the summer and winter ambient setups.

As a complement, the total energy usage when FF is used to control

- Heating, cooling and ventilation,
- Heating and ventilation,
- Ventilation…

…are also presented for all setups. It should be pointed out that the absolute values of the total energy usage should not be considered as relevant in this case. Instead, it is the difference between the control systems that are the essence of these results.

6.2.2.1 Heavy structure

Below the results from the setup consisting of heavy building structures are presented. Figures 6.4 and figure 6.5 show the total energy usage for the evaluated FF control systems as well as for the corresponding FB-control system. Figure 6.4 presents the results from the summer ambient conditions and figure 6.5 the results from the winter ambient conditions.

Comment: As explained in the beginning of this chapter, all of these control systems results in the same maximum CO_2 peak of 1000 PPM. Remember that parameter- and model-based refers to the entire FF control system. Hence, a system denoted as model-based are using the dynamic filter for thermal control and a parameter-based are using the direct filter for thermal control. All of the FF systems in this study are using the non-dynamic filter for IAQ control which is considered as a parameter-based filter as explained in section 4.3.2.2.

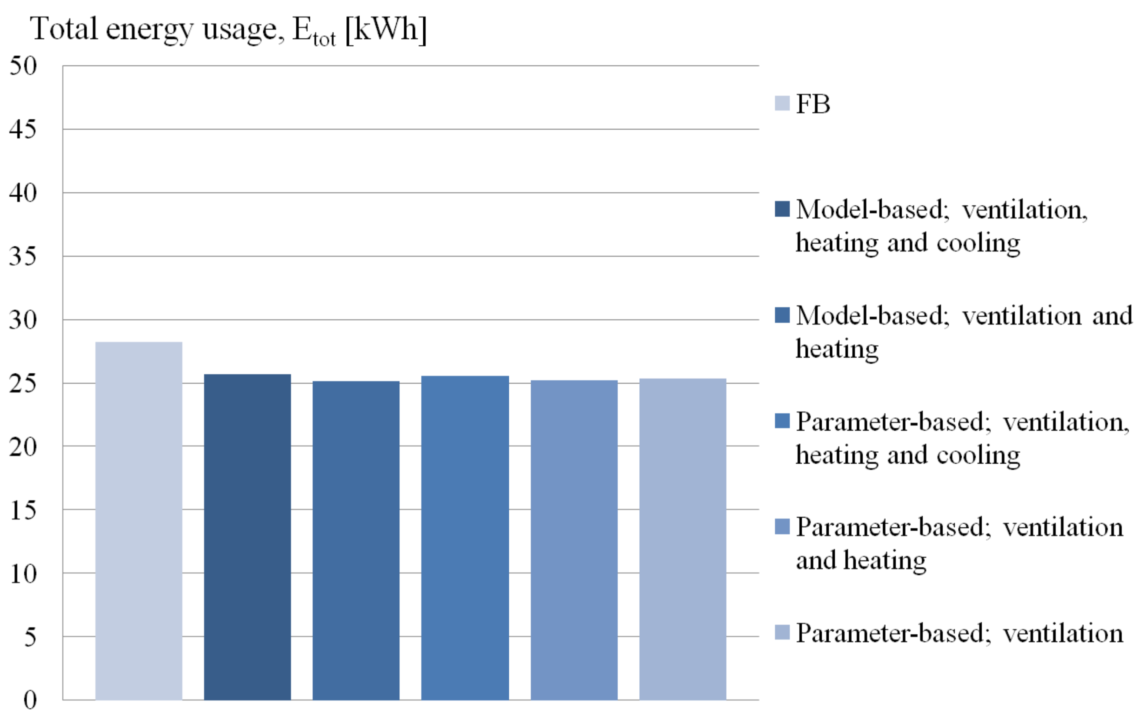


Figure 6.4 Difference in total energy usage for the evaluated FF control systems as well as the corresponding FB control system. Test setup: Multi-zone system, heavy building structure and summer ambient conditions.

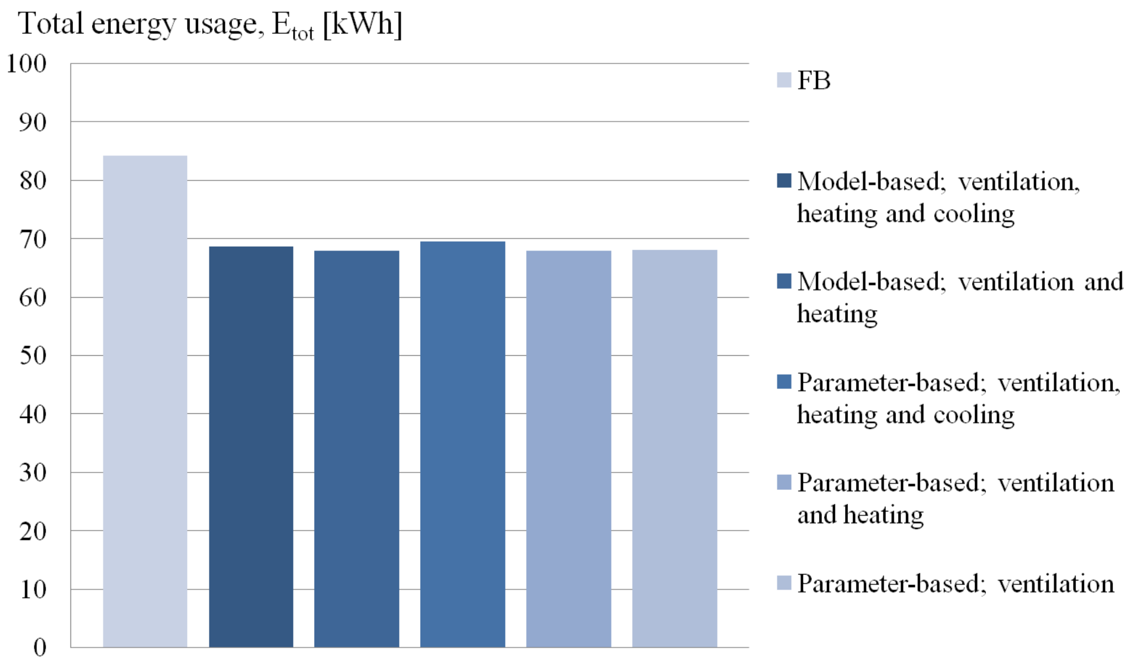


Figure 6.5 Difference in total energy usage for the evaluated FF control systems as well as the corresponding FB control system. Test setup: Multi-zone system, heavy building structure and winter ambient conditions.

Below the energy and peak power indicators for the heavy structure setups are presented. The first table presents the indicators referred to energy and the second the referred to peak power. Remember that the thermal peak power of the AHU and FCU refers to the dominating demand. Hence, during summer setups the peak refers to cooling power and during winter to the heating power.

Table 6.1 Final result regarding energy usage compared to a corresponding FB control system. Test setup: Multi-zone system, heavy building structure, summer and winter outside conditions

	Heating energy savings [%]	Cooling energy savings [%]	Electrical energy savings [%]	Total energy savings [%]
Parameter-based. FF-control of air supply. Ambient summer conditions	13	-1	58	10
Parameter-based. FF-control of air supply. Ambient winter conditions	19	-239	53	19

Table 6.2 Final result regarding required peak power compared to a corresponding FB control system. Test setup: Multi-zone system, heavy building structure, summer and winter outside conditions

	FCU power savings [%]	AHU power savings [%]	Electrical power savings [%]
Parameter-based. FF-control of air supply. Ambient summer conditions	16	28	54
Parameter-based. FF-control of air supply. Ambient winter conditions	0	35	54

6.2.2.2 Light structure

Below the results of the setup consisting of light building structures are presented. Figures 6.6 and figure 6.7 show the total energy usage for the FF control systems as well as for the corresponding FB-control system. Figure 6.6 presents the results from the summer ambient conditions and figure 6.7 from the winter ambient conditions.

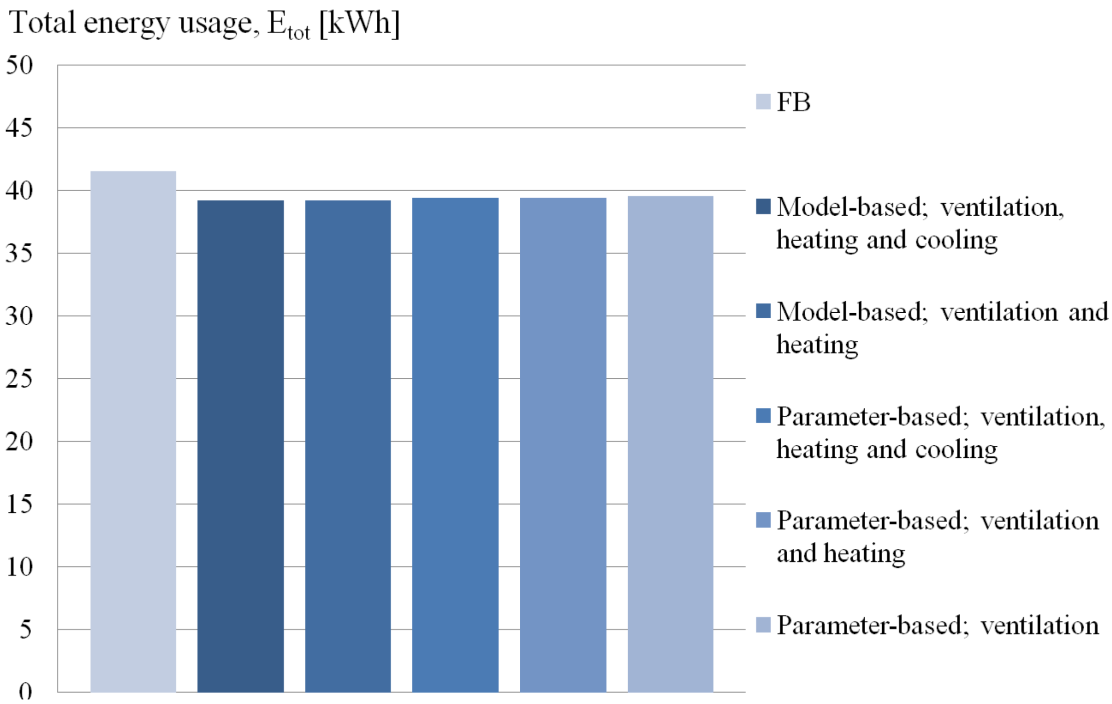


Figure 6.6 Difference in total energy usage for the evaluated FF control systems as well as the corresponding FB control system. Test setup: Multi-zone system, light building structure and summer ambient conditions.

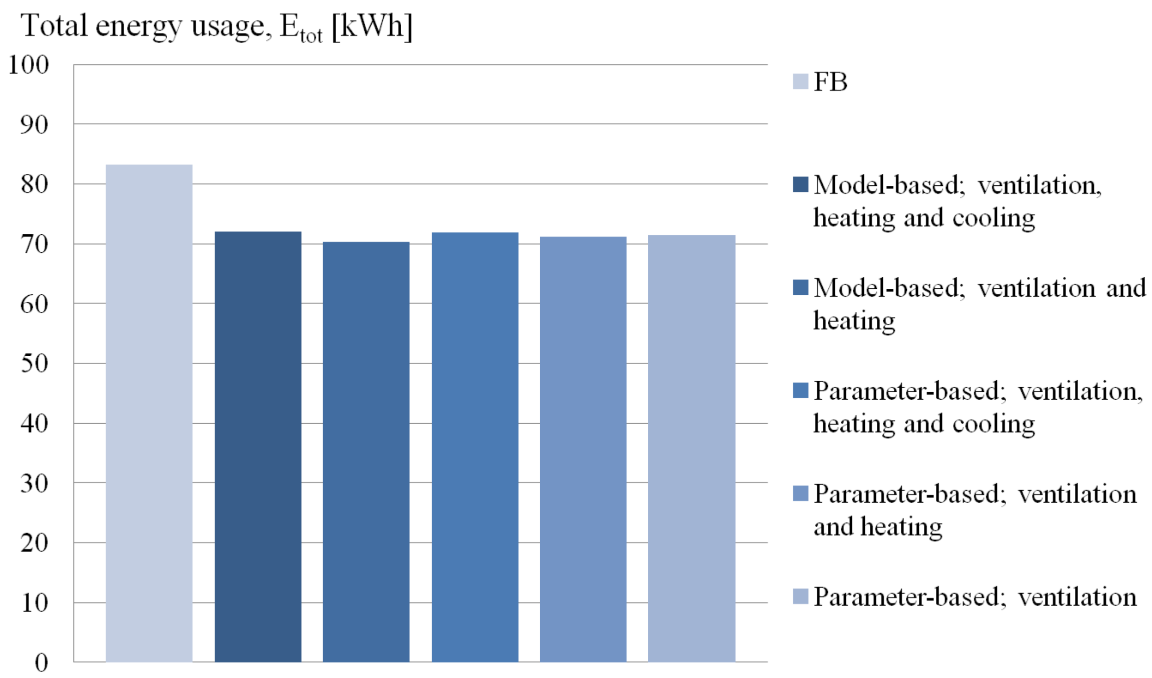


Figure 6.7 Difference in total energy usage for the evaluated FF control systems as well as the corresponding FB control system. Test setup: Multi-zone system, light building structure and winter ambient conditions.

Below the energy and peak power indicators for the heavy structure setups are presented. The first table presents the indicators referred to energy and the second the indicators referred to peak power.

Table 6.3 Final result regarding energy usage compared to a corresponding FB control system. Test setup: Multi-zone system, light building structure, summer and winter outside conditions

	Heating energy savings [%]	Cooling energy savings [%]	Electrical energy savings [%]	Total energy savings [%]
Parameter-based. FF-control of air supply. Ambient summer conditions	9	-3	55	5
Parameter-based. FF-control of air supply. Ambient winter conditions	17	-74	52	14

Table 6.4 Final result regarding required peak power compared to a corresponding FB control system. Test setup: Multi-zone system, light building structure, summer and winter outside conditions

	FCU power savings [%]	AHU power savings [%]	Electrical power savings [%]
Parameter-based. FF-control of air supply. Ambient summer conditions.	10	28	54
Parameter-based. FF-control of air supply. Ambient winter conditions.	0	35	54

6.3 Conclusions and discussion

The results of the multi-zone study correspond very well to the results of the study presented in chapter 5. All of the general conclusions presented in section 5.5 also apply here. In summary;

- The main part of the energy savings are related to FF-control of IAQ
- The peak power reduction is zero during winter ambient conditions
- The energy savings are larger for heavy building setups
- The energy savings are larger for ambient winter conditions

These conclusions have already been thoroughly discussed in section 5.5. The text below is instead focused on the conclusions that are specific to the study which have been presented in this chapter.

Conclusions regarding which distributions systems that preferable are controlled by FF

The most important result is that the performance of the control system is not improved to any larger extent if FF also is used for thermal control compared to when FF only is used for IAQ control. For that reason, the FF-controllers for thermal control is dropped in the rest of the work.

This conclusion corresponds well to the results in chapter 5. Then, it was shown that it was only in the heavy structure meeting room setup that any notable savings were derived from using FF for thermal climate control. In all other setups, the effect was small. Since the multi-zone system is dominated by office rooms, the low potential of using FF for thermal control was expected. Especially since the criterion of equal degree hours outside the temperature dead-band was dropped which means that an efficient rejection of thermal disturbances is not favoured.

In the previous study, the positive effects of using FF for thermal control were mainly related to peak power reductions of the FCU. However, this effect is not seen in the multi-zone case. In fact, the peak power is not reduced at all by including FF for thermal control compared to only utilizing FF for IAQ control. This is entirely dependent on the dropped criterion of equal degree hours outside the temperature dead-band. Without this criterion, larger overshoots of the dead-band is allowed. This means that the temperature setpoints of the FB-controllers doesn't have to be adjusted, which in turn means that these controllers can manage the task with smaller outputs.

Allocation of thermal energy supply

The results, from both the winter and summer setups, indicate that all of the thermal energy that is saved by using FF for IAQ control seems to derive from a reduced heating energy usage. More specific, the results seem to show that by using FF, the usage of heating energy is reduced but the usage of cooling energy is increased, compared to when the corresponding FB-control system is used. This can be seen in table 6.2 and 6.4 by comparing the heating and cooling energy indicators of the selected FF-control system;

- In the summer setups, the cooling energy is only increased a bit when the selected FF-control system is used. This is an effect of that the mean room air temperature is slightly reduced compared to when the corresponding FB-control system is used.
- In the winter setups, the cooling energy usage is increased massively, even though the total energy usage is decreased when the selected FF-control system is used.

The cooling and heating energy indicators of the winter setups are in fact a bit misleading. Their, a bit odd, values are due to a problem regarding the allocation of thermal energy supply that was aroused in this study since the setups couldn't be treated as pure heating or cooling demand cases. Below, the chain of events that is the cause of this problem is presented;

1. As previously discussed, the main effect of using FF for IAQ control is that a main part of the thermal energy supply is shifted from the AHU to the FCU.
2. From an FCU point of view, the supplied thermal energy is easy to characterize as either heating or cooling energy; heating if the temperature of the circulated air is higher than the room temperature and cooling if it's less.
3. However, from an AHU point of view this is harder, especially in the winter setup. The reason is that the central supply air always has a cooling effect on the room, but the outdoor air needs to be heated before it is supplied. This means that the system have to supply heating in order to provide cooling. This energy is characterized as heating energy in this work.
4. Since a larger part of the total thermal energy supply is put on the AHU in the FB-case, a larger part of the total energy derives from heating energy usage.
5. When FF is used, a larger part of the thermal energy supply is put on the FCU which means that the cooling energy indicator is increased at the same time as the heating energy indicator is decreased.

The problem of thermal energy allocation is avoided by instead looking at the total energy usage presented in figure 6.4, 6.5, 6.6 and 6.7 for the different setups.

Total energy savings

In this chapter, it was shown that by mixing office and meeting rooms, the total amount of energy that is saved by using FF ends up somewhere in between of what is expected when the rooms are treated separately. The total energy savings in the heavy building setups was 10 % in the summer case and 19 % in the winter case. The corresponding values for light building structures are 5 and 14 %. As mentioned, the FCU peak power of heating was not affected by the FF-controller while the corresponding cooling peak power was reduced by 16 and 10 % in the heavy and light building setups respectively.

As can be seen in table 6.1-6.4, both the peak power and the electrical energy that is saved by using FF are very similar in all setups. This depends on that the same FF-filter design for IAQ control was used throughout the study. The electrical energy usage is totally dominated by the driving energy to the central fans which means that the small differences in FCU fan operations between the FF and FB cases are in practise not seen in the results. The small differences in the AHU peak power indicator between winter and summer setups depend on the difference in energy intensity of the air-handling operation.

7 Central night-mode control

In chapter 6, the results of a study which extended over an entire work-week were presented. In that case, the focus was on the day-mode and more specific on local control of thermal climate and IAQ. The same temperature and CO_2 setpoints were used during the nights as during the days. This resulted in that;

- The ventilation was shut down during the night since the supply air flow rate only was dependent on CO_2 concentration.
- The room temperatures ended up somewhere in between 21 and 22 °C, depending on the building structure and the ambient climate.

The purpose of the study presented in this section is to evaluate both novel and conventional night-mode control strategies of the HVAC-system. The aim is to decrease energy usage and the required peak power compared to when the control systems makes no difference between day- and night-mode. The relation between this section and the rest of the work is that the novel night-mode strategies are based on the FF-filters that previously were used for local control.

7.1 Work in this field

Night-mode strategies are relatively well covered in the literature. The most common subject is to evaluate the utilization of free-cooling during night to reduce the supplied cooling during day. However, in most cases, the work focuses on a combination of day- and night-mode strategies. For example, many of the references presented in section 5.1 in connection to optimal and FF-control applications also include night-mode strategies. Below some references that are specifically focused on the night-mode are presented.

In the work by J.E.Braun, the setpoints of the room air temperatures that are used during the night in an office building are optimized [14]. A dynamical model of the rooms is used in combination with weather forecasts and the purpose is to minimize the energy usage and the temperature during the night. The room is cooled by free-cooling during the night and the room is heated to 20 °C when the working-day begins. The results showed that the peak-power of the cooling system was reduced by 30 % when the night-mode was used but also that the majority of the people felt that the temperature was too low in the morning.

A quite similar approach are presented in the two articles by M.Kolokotroni[38] and Z.Wang[69] but in these cases a constant air-flow is used to cool office buildings during the night. Different air flow rates are tested, and the results show that the peak temperature during the day is reduced by between 2 and 4 °C dependent on the size of the air-flow. Kolokotroni concludes that night-mode is most favourable in heavy structure buildings and Wang shows that the cooling energy usage can be reduced by 25 %.

In the work by M.Zaheer-Uddin[76] a different approach is used. During the night, an optimization problem is solved in which deviations from a lowered temperature setpoint is penalized and the energy usage is minimized. Hence, the room temperature is reduced when free-cooling is available and is increased again when the day-time is approaching. The results show that the strategy both has a positive effect on the energy use and the thermal comfort.

7.2 Method

The purpose of the night-mode strategies evaluated in this work is to improve the prerequisites for the day-mode control system by applying measures during the night. The intention is to eliminate or decrease the remains of the disturbances which have occurred over the day so that they are not inherited to the next. It was presumed that such strategies might relief the day-mode control, and by that, reduce both energy usage and required peak power.

In this work, the night-mode is entirely managed by the ventilation system. Hence, the FCU is never active during the night. The purpose is to utilize free cooling during the night which is achieved by decreasing the SAT setpoint to 12 °C. A lower temperature might be preferable but was dismissed due to practical reasons such as short-circuiting of the ventilation system and damaging of living plants inside the building.[53]

7.2.1 Simulation platform

This study is performed using the single-zone office room platform presented in section 3.4.1 and used in chapter 5. This platform was chosen to include the external disturbances, and especially OAT, which supposedly will have a large effect on the night-mode. The condition parameters in the setups consist of winter and summer ambient conditions as well as heavy and light building structures.

One difference compared to the platform in chapter 5 is that the symmetrical thermal boundary conditions used for floor and roof are also used for adjacent walls in this case. This boundary condition corresponds to an adiabatic process and represents that the entire building is engaged in the night-mode. Hence, it was assumed that what happens in the simulated room also happens in all the other rooms that are thermally connected to it.

The office room is simulated for five days and nights using the same climate data as presented for a northern cardinal direction in chapter 6. Hence, the ambient climate is reused from the multi-zone case but the rest, such as internal disturbances and the simulation platform are based on the single-zone study presented in chapter 5. During the day, the control is handled by the FB-system used in chapter 6, that is, the only criteria for the day-time controllers is to avoid breaching 1000 PPM during the working period.

7.2.2 Night-mode constraints

The working-hours stretch from 07:00 to19:00 and the night-mode is active between 20:00 and 05:00. Hence, a 24 hours simulated period can be divided in four different periods:

1. A working-hours period of 12 hours in which the day-mode is active
2. A one hour period between the working-hours and the activation of the night-mode. This period is used to delay the start of the night strategy.
3. A night-time period of 9 hours in which the night-mode is active

4. One intermediate period of two hours between the deactivation of the night-mode and the start of day-mode

The delayed start, presented as bullet number 2, above is included to make the study more realistic since the night-mode must not be activated before the building is empty.

During the night-mode, the temperature of the room is allowed to drop below the lower boundary of the dead-band. However, a criterion is that the FB-controllers should be able to bring back the room temperature to the setpoint during intermediate period, presented as bullet number 4 above. <u>If this isn't achieved the night-mode strategy in question is rejected for the setup in question</u>. Dependent on ambient climate and building structure this intermediate period is of more or less importance.

7.2.3 Night-mode strategies

Four different night-mode strategies were evaluated in this work. Two of them are conventional while the other two are based on the FF-filters that previously were used for local control. These are denoted as;

1. Conventional
   - Shut-off
   - Constant flow rate
2. FF-based
   - IAQ focusing
   - Thermal focusing

In the study, these strategies were evaluated by comparing to a control system which makes no difference between night and day, i.e. the FB control system used in chapter 6. The evaluation is based on the same indicators for energy and peak power as described in section 1.3.3.

7.2.3.1 Conventional strategies

When the strategy denoted as shut-off are used, the entire HVAC-system is inactivated during the night-time. The second conventional method, denoted as constant flow rate, will supply air of a constant supply rate throughout the entire night. This constant supply was set arbitrary and corresponds to half of the design air flow rate.

7.2.3.2 FF-based strategies

The two FF-based strategies utilizes the same tools as previous was developed for local FF-control. However, the filters are a bit adapted to this study. Both strategies are designed to calculate the required supply to end up at a predetermined lowered night-setpoint by measuring all disturbances which are of relevance. The thermal focusing strategy is equipped with a temperature setpoint and the IAQ focusing with a CO_2 setpoint.

These strategies are designed to minimize the supply with respect to this predetermined night-setpoint. That is, by only supplying the amount of ventilation air that is necessary for ending up at the night-setpoint at the exact time as the night-mode period ends. This means that the time is introduced as an additional aspect in the control. By doing so, the supply is minimized at each time instant which on the other hand doesn't automatically imply a minimization of the accumulated supply. This is further discussed in the discussion part of this chapter.

IAQ focusing

The IAQ focusing strategy requires measurements of the CO_2 concentration in the supply air as well as the current state of CO_2 concentration in the room. It is based on equation 7.1[22] which is an analytical solution to the differential equation used in the dynamic filter for IAQ control presented in section 4.3.2.2.

By solving for the supply air flow rate, equation 7.2 is instead formulated which is the current equation used in the filter. In this equation:

- The time-index τ denotes the current time
- The time τ_n refers to the duration of the night-time
- The room air concentration at time τ_n refers to the night-setpoint which was set to 600 PPM

By moving the time horizon forward, the required supply air flow rate is calculated and sent to the system at each time instant. In this work, the time resolution was set to 60 seconds.

$$c_r(\tau_n) = c_s + \frac{\dot{M}}{\dot{V}} - \left(c_s + \frac{\dot{M}}{\dot{V}} - c_r(\tau)\right) e^{-\frac{\dot{V}(\tau)}{V}\cdot\tau_n} \text{ [PPM]} \qquad \text{(eq. 7.1)}$$

$$\dot{V}(\tau) = \frac{V}{\tau_n}\left(ln\left(c_r(\tau) - c_s\right) - ln\left(c_r(\tau_n) - c_s\right)\right) \text{ [m}^3\text{/s]} \qquad \text{(eq. 7.2)}$$

Thermal focusing

The strategy denoted as thermal focusing is based on the filter denoted as dynamic in section 4.3.2.1. That is, this strategy is based on a dynamical energy balance of the room. A night-setpoint of 21 °C was used in order to avoid unnecessary heating in the intermediate period.

A thermal room model similar to equation 7.2, which includes the time as a parameter, would, due to the large number of governing differential equations, be much more complicated and was therefore dismissed. Instead, the original dynamical energy-balance with an infinite time-horizon described in section 4.3.2.1 was used. This approach is motivated by that the time-aspect is of less importance in the thermal case compared to the IAQ case due to the large thermal inertia of the building. In the IAQ focusing strategy, the time is included to slow down the controller which otherwise would have reached the night-setpoint in a relative short time. But, in the thermal case, by not including the time as a parameter, the resulting evolution of room temperature in time will correspond to the thermal inertia of the building. Even though this approach is an approximation, the results showed that the accuracy was acceptable both in the heavy and light structure case, and it is considered to reflect the most essential features of the more complex strategy.

7.3 Results

Generally, the positive effects of using night-mode were scarce compared to the original strategy that made no difference between night and day. It was only in the heavy building winter setup that any noticeable positive effects of using night-mode were indicated. For that reason, the results from this setup are considered as the primary result of this study and below, the corresponding energy and peak-power indicators as well as the total energy usage are presented.

In figure 7.1, the resulting total energy usages when the different night-mode strategies are used in the heavy building winter setup are presented. Note that the absolute values should not be considered as relevant in this context. The purpose of the figure is instead to illustrate the difference between the strategies.

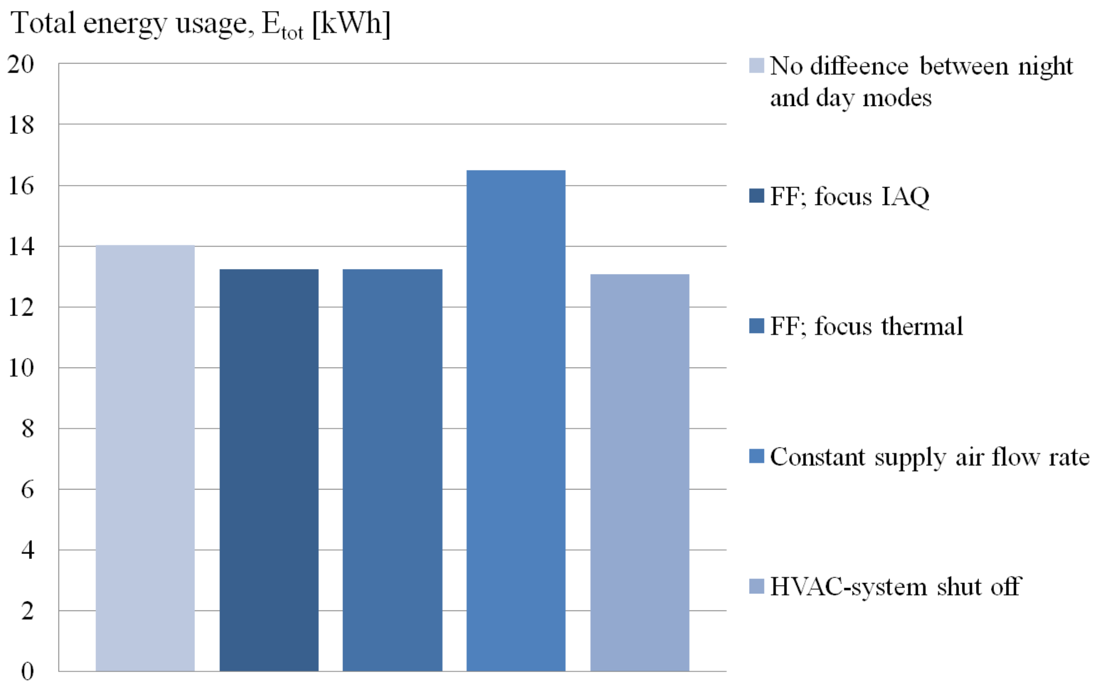


Figure 7.1 Difference in total energy usage by using night-mode compared to using the same mode during night and day. Test setup: Office-room, heavy building structure, winter outside conditions

The first table below presents the energy indicators and the second the peak power indicators for the heavy structure winter ambient setup. As can be seen, the performance of the two FF-based strategies and the shut-off strategy are similar. Among these, the latter is a bit more favourable and was therefore selected for further testing.

Table 7.1 Difference in energy usage by using night-mode compared to using the same mode during night and day. Test setup: Office-room, heavy building structure, winter outside conditions

	Heating energy savings [%]	Cooling energy savings [%]	Electrical energy savings [%]	Total energy savings [%]
IAQ focusing	3	45	2	6
Thermal focusing	4	44	1	6
Constant flow rate	-22	59	-24	-17
Shut-off	5	42	2	7

Table 7.2 Difference in required peak power by using night-mode compared to using the same mode during night and day. Test setup: Office-room, heavy building structure, winter outside conditions

	FCU power savings [%]	AHU power savings [%]	Electrical power savings [%]
IAQ focusing	-335	0	0
Thermal focusing	-326	0	0
Constant flow rate	-389	0	0
Shut-off	-325	0	0

In the light building winter setups, all night-modes failed to fulfil the criteria to get back to the lower boundary of the temperature dead-band during the intermediate period. Hence, the night-mode strategies were rejected in these cases. In all of the summer setups, the night-mode strategies resulted in very small positive effects or even an increased energy usage and/or peak power. These results are presented in appendix D.

7.4 Conclusions

As mentioned, the positive effects of night-mode were scarce. The only setup in which any notable energy savings were distinct was the heavy building structure during winter ambient condition. In this case, the strategy denoted as shut-off was most favourable and resulted in a total energy savings of 7 %. But, on the other hand, the required peak power of the FCU was at the same time increased drastically. The reason is that during winter conditions, the room temperature drops quite quickly during the night; to about 19 °C in the setup in question. During the intermediate period, the temperature is increased to 21 °C which means that the control error, and in turn the control signal, of the FB-controller momentarily becomes large which results in large power outputs of the FCU.

For now, the increased required peak power is overseen due to the indicated energy savings potential and night-mode is still considered as an option to be used in chapter 9 were the selected strategies from the consecutive studies are combined. This is motivated by that the power peak might be reduced when FF day-mode is used in combination with the night-mode strategy.

7.5 Discussion

The positive effect of night-mode is a decreased cooling demand during the day; the building structure is cooled during the night which supports the FCU during the day. However, for night-mode to be feasible some general prerequisites are required;

1. There must be a cooling demand during the day
2. Free-cooling due to the low ambient temperature must be available during the night
3. The thermal inertia of the building should preferable be large
4. The night-mode should preferable be utilizable over a long period of time

The negative aspects of night-mode are;

1. An increased heating energy usage, both related to AHU and FCU
2. An increased electrical energy usage due to the prolonged operation of the central fan

Prerequisites for night-mode

The first and second prerequisites presented above were fulfilled in all setups. Both are quite intuitive since without the first, the effect of night-mode is not desirable and without the second, the night-mode has no effect. The third prerequisite is to avoid large temperature drops of the room air during the night which in turn results in a large heating energy usage in the intermediate period. Instead, it is desirable that the building structure is cooled to a feasible temperature, close to the lower boundary of the temperature dead-band, during the night. Furthermore, a heavy building will have the possibility to accumulate cooling energy and thereby affect the room during a large part of the day. When a light building is cooled during the night, the temperature drop of the walls are large but during the intermediate period a large part of this cooling energy is lost and the remaining effect only last a short while after.

The fourth prerequisite can only be realized if the ambient temperature is low during a large part of the night. This prerequisite is lost in the summer cases and is the main reason to why none of these setups yielded any positive effects. In those cases, the cooling energy supplied during night is not large enough to exceed the increased electrical energy usage. However, a low ambient temperature also results in an increased heating energy usage in the AHU since the temperature of the supply air have to be increased to 12 °C. A large part of this is managed by the heat-recovery system but as the temperature in the building drops, less heat can be extracted from the exhaust system.

Negative aspects
Both of the negative aspects presented above explain the general poor results in this study. In all summer as well as light structure winter setups, the total energy usage was increased since the reduced cooling demand during the day is exceeded by the additional energy usage. The primary reason is an increased heating energy supply during the intermediate period. But in about half of the setups (including the ones presented in appendix D), also the electrical energy usage was increased due to a prolonged fan operation.

In the second half of the setups, the electricity usage was unchanged or even decreased a bit when night-mode was utilized. These results can be explained by three effects;

- **First**, the nightly air-supplies determined by the FF-based strategies, i.e. the thermal and the IAQ focusing strategies, were very low in most setups. This means the corresponding results became very similar to the strategy to shut off the HVAC-system which also is seen in the results.
  When the thermal focusing strategy was used in winter setups, the required air flow to bring down the temperature to 21 °C during the night was almost zero due to large thermal losses to the ambience. When the IAQ focusing strategy was used in general, the delay of the night-mode had a large impact on the air-supply during night since the day-mode had already reduced the CO_2 concentration substantially before the night-mode was engaged. This means that the largest part of the task set out for the night-mode had already been completed which means that the supply air flow rate during the night became very low.
- **Second**, even though the day-mode already had reduced the CO_2 concentration substantially before the IAQ focusing night-mode was engaged, the day-mode had not managed to completely bring the CO_2 concentration below the setpoint. This means that the night-mode was interrupting the day-mode by reducing the supply air flow rate instantaneously. This means that the electricity usage as well as the heating energy used in the AHU was slightly reduced when the IAQ focusing strategy was used.
- **Third**, in some setups, the effect of night-mode was that the cooling energy supplied by the FCU was decreased a bit more than the corresponding increase of supplied heating. Hence, the total electricity usage of the FCU integrated fan was decreased a bit.

7.5.1 Suggestions for improvements

As discussed in section 7.2.3, the purpose of the FF-based night strategies is to minimize the supply at each time instant with respect to the predetermined night-setpoint. However, also as mentioned, this might not be the same as minimizing the accumulated supply during the night. Especially, this question of doubt applies to the thermal control due to the thermal losses; the concept of a more or less evenly distributed free-cooling supply during the night might not be ideal. Alternative methods would be to quickly cool the building right before the intermediate period to reduce thermal losses or to maximize the free-cooling supply during times when the ambient temperature is low. These procedures require other types of control methods than the ones that have been developed in this work and are therefore considered as out of the scope.

IAQ focusing strategy
One limitation that affected the performance of the IAQ focusing strategy negatively was that the night-mode was not activated until one hour after the working-period had ended. As mentioned, during that period, the day-time controller had already ventilated the room to a large extent which limited the possibilities of the night-mode. Suggestively, the IAQ focusing strategy could instead be activated by motion sensors, which means that the need for a delayed start might be avoided.

Thermal focusing strategy
At least three problems regarding the thermal focusing strategy used in this study has been identified. The **first problem** is that the model does not take the time-aspect into account. The current model calculates instead the supply based on an infinite time horizon.

The **second** problem, which presumably has a large effect on the performance, is that the model does not take the future variation of the OAT into account. In the beginning of the night, the supply is too large based on that the OAT most definitely will drop even further. This is compensated for by shutting off the supply when the OAT becomes low enough but due to the exaggerated initial supply this normally doesn’t prevent the night-setpoint to be breached, especially during winter. To avoid this, weather forecast and a predictive controller might be necessary.

Third, the thermal focusing model only calculates the required thermal supply to the room, not taking into account that it is only cooling energy that can be supplied by free-cooling. Hence, the minute the night-setpoint is breached, the model shifts to a heating demand which is not allowed to be supplied during the night. To avoid this, a predictive controller that takes the time aspect into account might be necessary.

8 Central supply air temperature control

The multi-zone system presented in section 3.4.2 was for the first time used as platform in chapter 6. The focus was then on FF for local control and the central SAT control was managed by a constant setpoint.

The aim of the study presented in this chapter is to develop SAT control strategies for decreased energy usage and decreased required peak power. Alternative SAT control strategies are tested and the most favourable strategy is selected for further testing in chapter 9. The relation between this section and the rest of the work is that most of the novel SAT control strategies are based on the FF-filters that previously were used for local control.

The exact same platform, disturbance profiles and condition parameters as were used in chapter 6 are also used in this study. That is, the 10 room multi-zone system of either heavy of light building structure simulated during either summer or winter ambient conditions. The simulated period consists of the third day presented in section 6.1.1 but is limited to the occupied period to avoid the effects of night-time in the results.

To make the study as transparent as possible, the local control of thermal climate and CO_2 level is managed by the FB-control system presented in chapter 6. Hence, the controllers are tuned to avoid breaching 1000 PPM and the temperature dead-band stretches at all time between 21 and 22 °C.

8.1 Work in this field

Control of SAT is quite well covered in the literature but most references are focused on all-air systems. For example, in the works by, S.Schiavon[58], Y.H.Cho[16] and Y.P.Ke[36],the focus is to find a favourable combination of supply air flow and SAT on a local level to control the thermal climate in office buildings.

In this work, on the other hand, the supply air flow rate is locked by the requirements on IAQ which makes most references irrelevant. However, two references have been considered. These are presented below.

In the work by F.Engdahl[24] an optimization problem is solved to minimize the energy usage by optimizing the SAT for different ambient temperatures. Although the application is a VAV-system for thermal control, the methodology is relevant to this work since the focus is on the SAT on a central level. The results shows that, dependent on the ambient climate and internal heat generation, the energy usage can be decreased by about 8 – 27 % compared to the most energy intense strategy.

In the work by [46], the application is an office building designed for 1600 persons with a VAV system for thermal control. The SAT is controlled centrally based on the highest local exhaust air temperature in the system. Hence, the room with the highest heat loads are determining the SAT and the purpose is to minimize the supply air flow rate. Compared to an OAT compensating SAT control, the energy usage is reduced by 40 %.

8.2 Method

Three novel SAT control strategies were evaluated in this study. These are denoted as;

- Optimal mix
- Person counting
- Dynamic model

In the evaluation, these three strategies are compared to a conventional strategy in which the SAT is determined by the OAT.

8.2.1 Conventional SAT control

The conventional strategy consists of a linear relationship between SAT and OAT. This function is presented in figure 8.1 and is based on an example from a real office building with a similar HVAC-system design, i.e. with hygienic ventilation system and local units for heat and cooling supply[53].

The methodology to entirely base the SAT on the OAT implies that the OAT is the dominant thermal disturbance acting on the rooms. However, during normal operation of office buildings, this is not usually the case since the thermal emission from the internal disturbances most often is dominant. On the other hand, OAT is easy to measure and it affects the entire building instead of just parts of it as internal disturbances do.

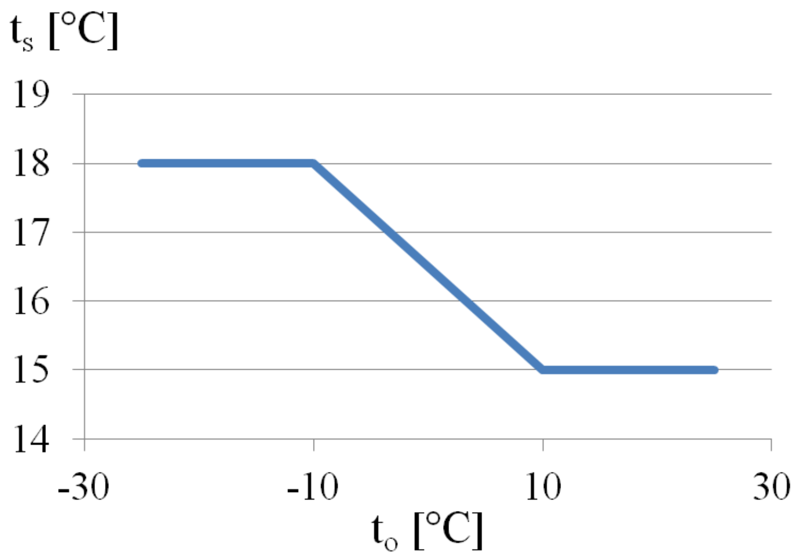


Figure 8.1 FF-model for conventional control of SAT as a function of OAT.

8.2.2 The optimal mix strategy

The term “mix” refers in this case to the operational mix of the cooling supplying HVAC-units, i.e. the AHU and the FCU. The strategy was determined in an iterative approach with the aim to minimize the usage of thermal energy and electricity.

When this control strategy is implemented, the OAT and the cooling supplied by the FCU are used to determine the amount of cooling that should be used for air-conditioning in the AHU. Since the supply air flow rate is locked by the IAQ demand, the SAT can in turn be directly solved by using equation 8.1 below.

$$\dot{Q} = \dot{V} \cdot \rho_{air} \cdot c_{p,air} \cdot (t_s - t_o) \text{ [W]} \qquad \text{(eq. 8.1)}$$

The approach to focus on energy for air-conditioning in the AHU instead of cooling supplied to the room by the ventilation system was chosen to disconnect the control strategy from the actual room temperature. This approach facilitates the strategy to be implemented in multi-zone cases since the local cooling supplied by the ventilation system will vary if the temperatures in the different rooms differ, even though the same supply air flow rate and SAT is supplied.

Design process

The optimal mix strategy was developed in a separate design process in which a single-zone meeting room with cooling demand was used as a platform. The design process was conducted by determining the steady-state optimum operational mix of the AHU and FCU as a function of OAT.

The parameters in the design process were the OAT as well as the part of the total cooling energy allocated to the AHU and FCU respectively, i.e. the cooling energy mix. For each OAT, different mixes were tested by varying the maximum cooling supply by the FCU. The mix that resulted in the lowest overall energy usage was chosen.

The SAT, the temperature setpoint of the room as well as the internal disturbances were set as constants throughout the design process. By setting the SAT to 15 °C, the room temperature setpoint to 22 °C and the internal disturbances originated from people to 500 W, the entire cooling demand could be supplied by hygienic ventilation.

The design process was carried out by connecting both of the local control-loops to the thermal climate. Hence, two FB-controllers were used to control the room temperature; one that controlled the supply air flow rate and one that controlled the FCU cooling power supply. Both controllers had the same setpoint which means that they worked simultaneously. However, the cooling supplied by the FCU was prioritized but also limited. By limiting the supply from the FCU, the part of the total cooling energy used by the AHU was stepwise varied between 100 and 0 %.

Results

An intermediate result of the design process which originates from an OAT of 17 °C is shown in figure 8.2. The x-axis represents the mix, interpreted as the part of the total cooling supply that derives from the FCU. The y-axis represents the relative total energy usage regarding cooling energy as well as the electricity used by the central fan and the integrated fan of the FCU. As can be seen, the optimum is found for the mix of 30 % FCU and 70 % AHU.

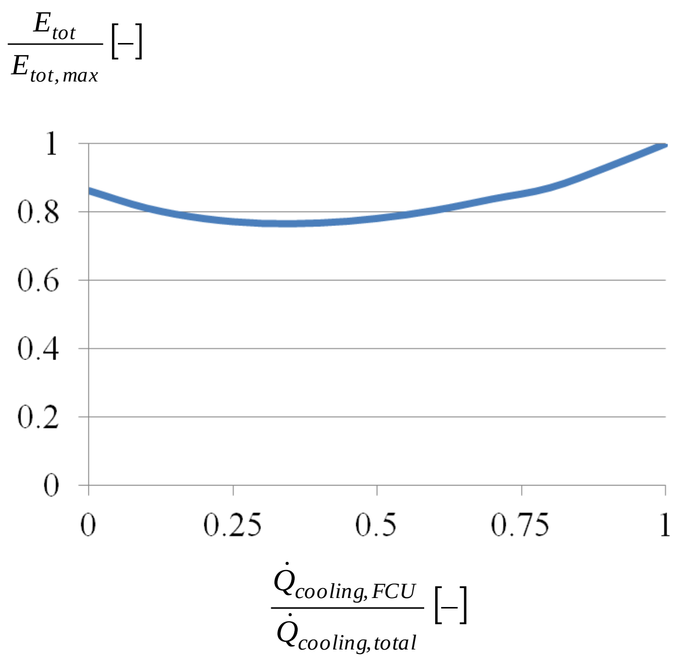


Figure 8.2 The effect of cooling supply allocation on total energy usage for an OAT of 17 °C. An optimum is found when about 30 % of the total cooling demand that is supplied by the AHU.

The final result of the design process is shown in figure 8.3. This function was derived by combining the results from all of the OAT specific simulations (e.g. figure 8.2) and consists of the mixes for which the total energy usage was minimized. As can be seen, above an OAT of 22 °C the entire cooling energy usage is preferable put on the FCU due to an energy intensive central air-handling. Below an OAT of 16 °C, 80 % of the cooling supply should derive from the AHU since a large part of the central cooling is free, but due to the electricity intensive operation of the central ventilation system, 20 % should derive to the FCU.

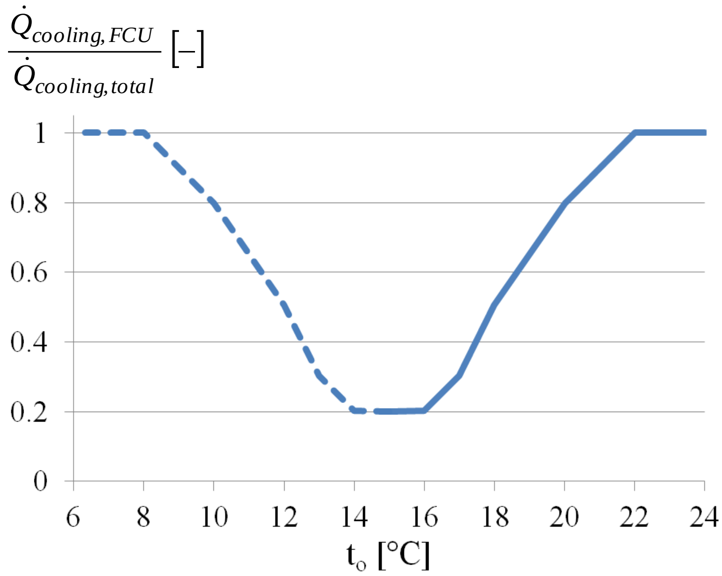


Figure 8.3 Optimal part of cooling demand supplied by AHU as a function of OAT

The optimal mix design process was only carried out for the outdoor temperatures that resulted in a demand for cooling energy in the AHU. The shift to heating demand occurs at an OAT of 15 °C, since below that temperature the AHU have to heat the outdoor air. In figure 8.3, the results below 15 °C were derived by mirroring the results from the cooling cases and this is indicated by the dashed line. This approach was considered as valid since only dry cooling is regarded in this work. Hence, the same amount of energy is required to achieve a certain temperature drop as is required for the corresponding temperature rise.

When it comes to the heating demand cases, the strategy determines the amount of heating that should be supplied by the AHU after the heat-recovery unit. Hence, recycled heat is not taken into account, which means that the temperature scale in figure 8.3 refers in the heating demand cases to the temperature at the inlet of the heating-coil.

8.2.3 Person counting strategy

The person counting strategy is based on a similar methodology as the conventional strategy but recognizes instead internal disturbances as the dominating influence on thermal climate.

The strategy determines the SAT by utilizing the output of local person counters as input to a function presented in figure 8.4. The x-axis denotes the occupancy factor (OF), i.e. the number of people present relative to the number of people that the HVAC-system and building is designed for, and the y-axis the SAT.

The function as well as the breaking temperatures in figure 8.4 was set more or less arbitrary in this work. However, some tuning was performed to avoid subcooling or overheating of the building.

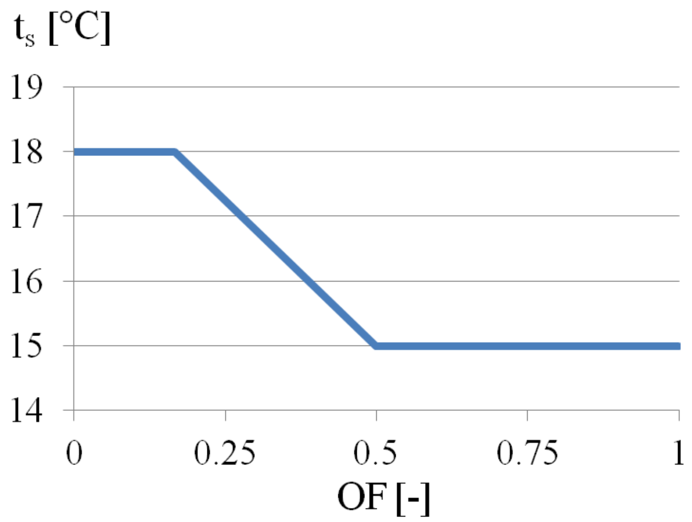


Figure 8.4 SAT as a function of the relative number of occupancy

8.2.4 Dynamic model

The last strategy is based on the dynamic FF-model for thermal control which was presented in chapter 5. Each room is equipped with its own model and the input is the complete set of thermal disturbances acting on the room. The model is used to calculate the SAT which would be needed to keep the temperature of the room above 21 °C without the help of the FCU. This is done for each room in the multi-zone platform and the room that returns the highest SAT is prioritized, i.e. this room determines the SAT for all rooms.

The purpose of this strategy is to minimize the SAT and hence utilize the hygienic flow for cooling as far as possible, but at the same time to avoid creating additional heating demands by subcooling so called disfavoured rooms. The term disfavoured refers to rooms in which a large proportion of the internal disturbances originate from people. Then, the supply air flow rate is large in relation to the cooling demand which means that there is a risk that the cooling power of the hygienic flow is larger than what's demanded. This does not apply to empty rooms since their supply air flow rate is zero in this work.

8.3 Results

In the text below, the main results of the SAT control study are presented. As in previous studies, the performance of the control systems are represented by the energy and peak power indicators presented in chapter 1.3.3. As mentioned, the indicators are calculated by comparing the novel strategies to the conventional OAT based strategy.

In addition, part-results derived from the design process of the optimal mix strategy as well as the total energy savings related to the different SAT strategies are presented.

8.3.1 Optimal mix part-results

Below the main results from the design process of the optimal mix strategy are presented. These results show the total energy savings potential by using the optimal mix strategy and they derive from the single-zone platform which was used in the design process. Hence, these results are not related to main results from this study that are derived from the multi-zone platform, but are included as a prolonging of the optimal mix design process.

Figure 8.5 shows the effect on the total energy usage by using the optimal mix strategy for SAT control instead of generating the entire cooling supply by either the AHU or the FCU. The y-axis represents the total energy usage presented as equation 1.2 in section 1.3.3 and the x-axis the OAT. For example, at an OAT of 17 °C, 10 or 25 % of the total energy is saved by utilizing the optimal mix instead of only using the FCU or the AHU respectively.

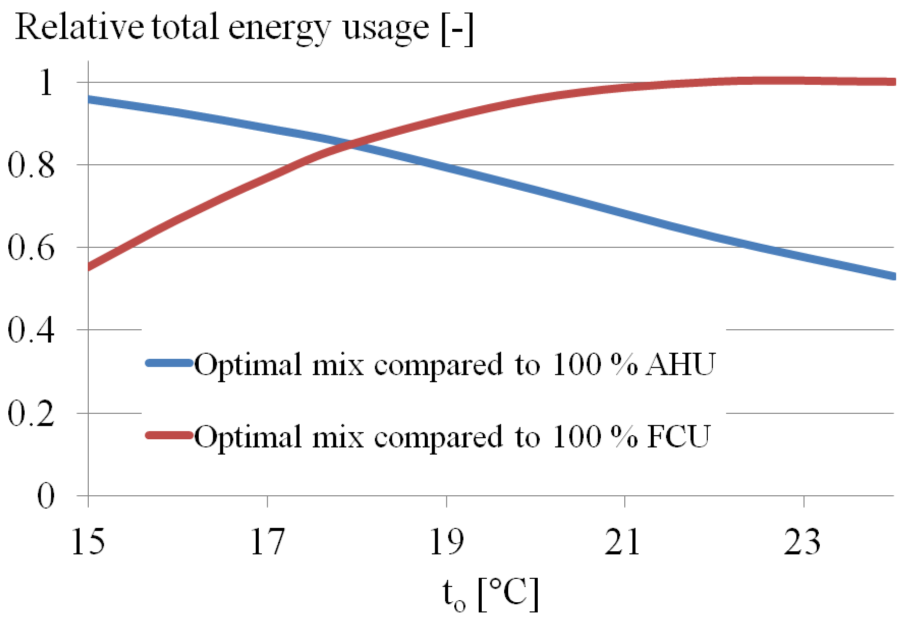


Figure 8.5 Difference in total energy usage by utilizing optimal mix instead of only AHU or FCU respectively

8.3.2 Main results

The strategy which was proven most favourable over the entire set of setups was the person counting strategy. This strategy was selected for the further testing in chapter 9, and below, the corresponding energy and peak power indicators are presented together with the total energy usages related to all of the evaluated strategies. The indicators for the rest of the strategies are presented in appendix E.

Figure 8.6 presents the SAT as a function of time when the selected person counting strategy is used. Since this strategy only is dependent on the number of people in the building and is independent on the ambient climate, the behaviour in figure 8.6 corresponds to all setups.

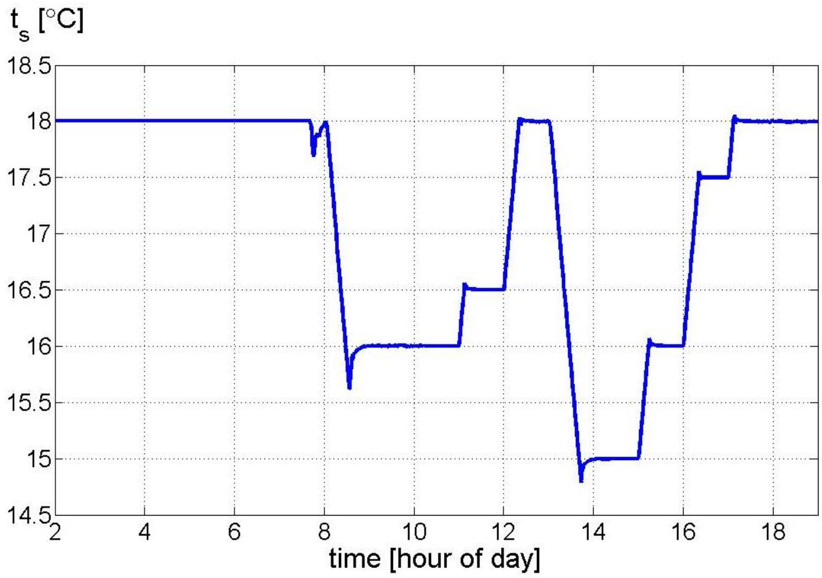


Figure 8.6 SAT as controlled by the people counter based approach

Heavy building structure setup

In figure 8.7 and 8.8, the total energy usages when the different SAT control strategies are used in the heavy building setup are presented. Figure 8.7 corresponds to the summer ambient climate and figure 8.8 to the winter ambient climate. Note that the absolute values should not be considered as relevant in this context. The purpose of the figures is instead to illustrate the difference between the strategies.

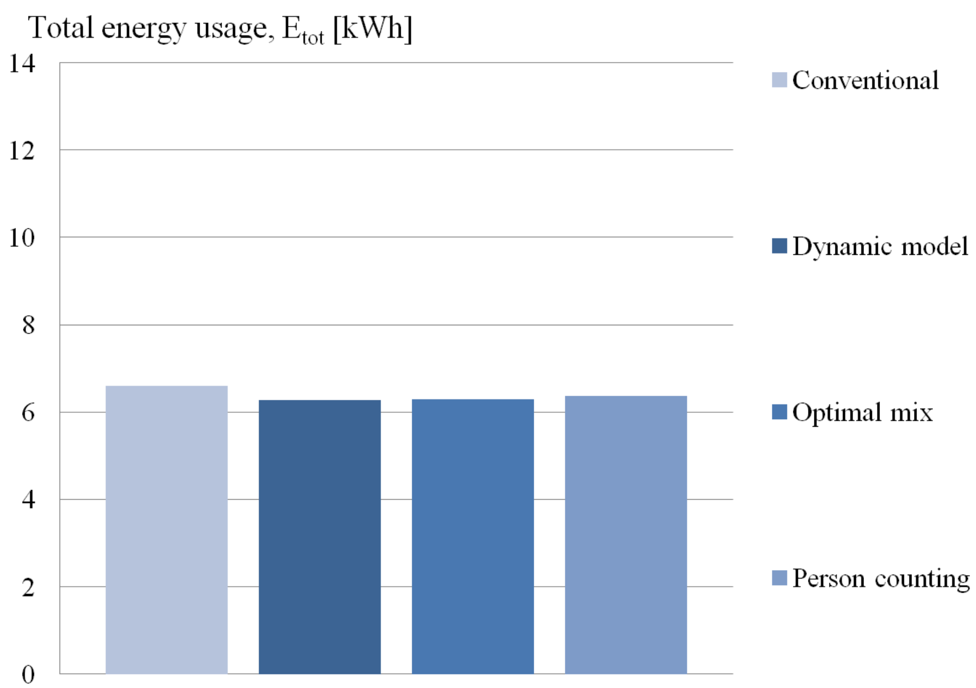


Figure 8.7 Total energy usage when the different SAT control strategies are used. Setup: heavy structure building, summer ambient climate

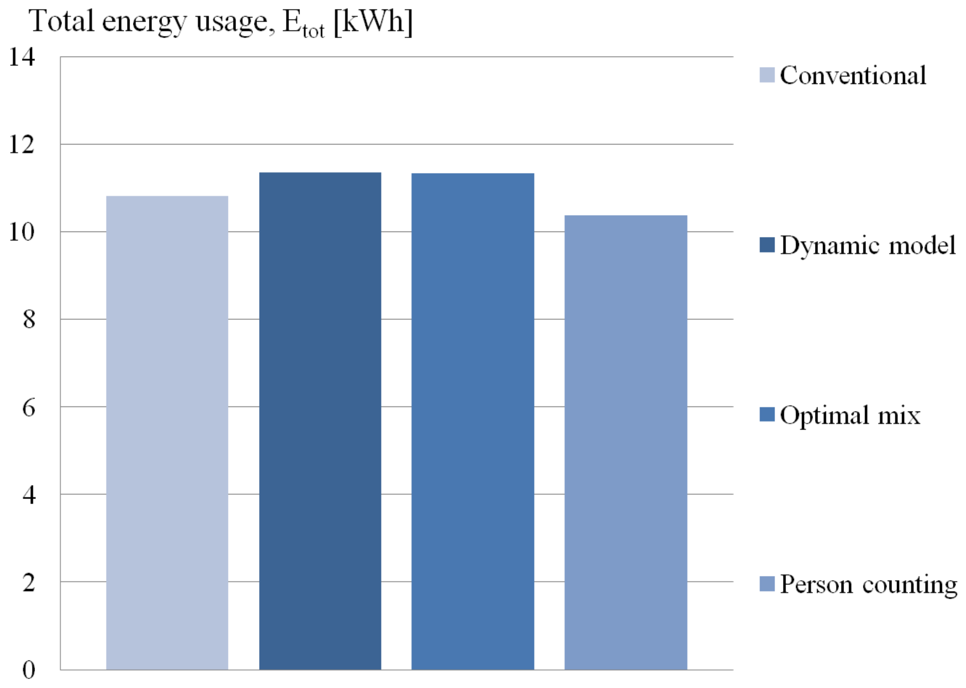


Figure 8.8 Total energy usage when the different SAT control strategies are used. Setup: heavy structure building, winter ambient climate

The resulting energy and peak power indicators when the personal counting strategy is used instead of the conventional OAT based strategy are presented in the tables below. The results correspond to the heavy building setup.

Table 8.1 Difference in energy usage by using the person counting SAT control strategy instead of conventional OAT based control. Test setup: Multi-zone, heavy building structure, winter and summer outside conditions

	Heating energy savings [%]	Cooling energy savings [%]	Electrical energy savings [%]	Total energy savings [%]
Person counters. Summer ambient conditions	1	4	0	4
Person counters. Winter ambient conditions	3	71	0	4

Table 8.2 Difference in required peak power by using the person counting SAT control strategy instead of conventional OAT based control. Test setup: Multi-zone, heavy building structure, winter and summer outside conditions

	FCU power savings [%]	AHU power savings [%]	Electrical power savings [%]
Person counters. Summer ambient conditions	-3	0	0
Person counters. Winter ambient conditions	0	12	0

Light building structure setup
In figure 8.9 and 8.10, the total energy usages when the different SAT control strategies are used in the light building setup are presented. Figure 8.9 corresponds to the summer ambient climate and figure 8.10 to the winter ambient climate. Note that the absolute values should not be considered as relevant in this context. The purpose of the figures is instead to illustrate the difference between the strategies.

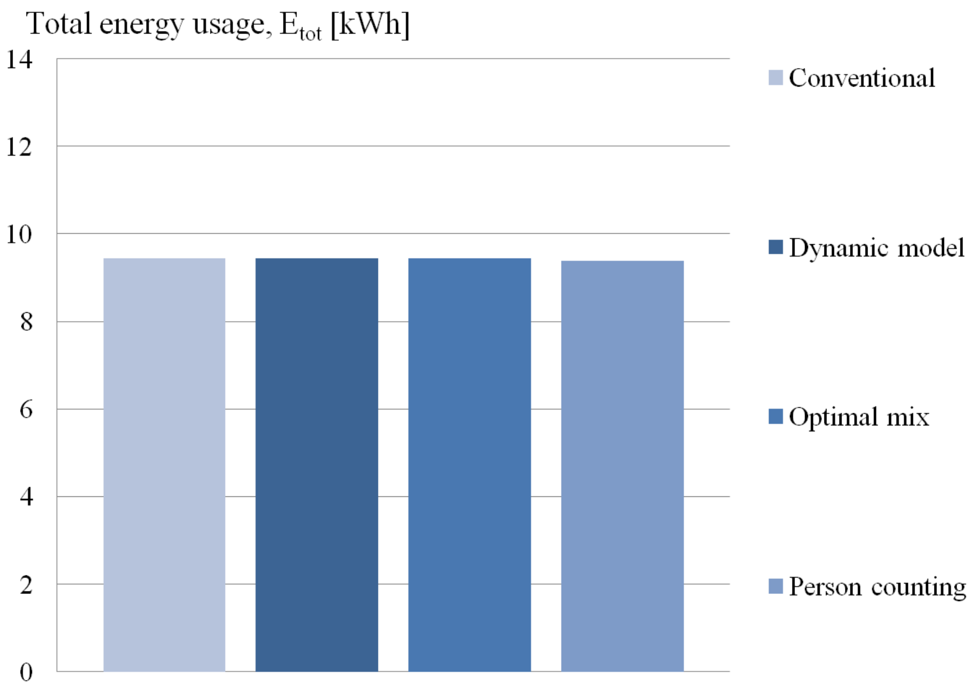


Figure 8.9 Total energy usage when the different SAT control strategies are used. Setup: light structure building, summer ambient climate

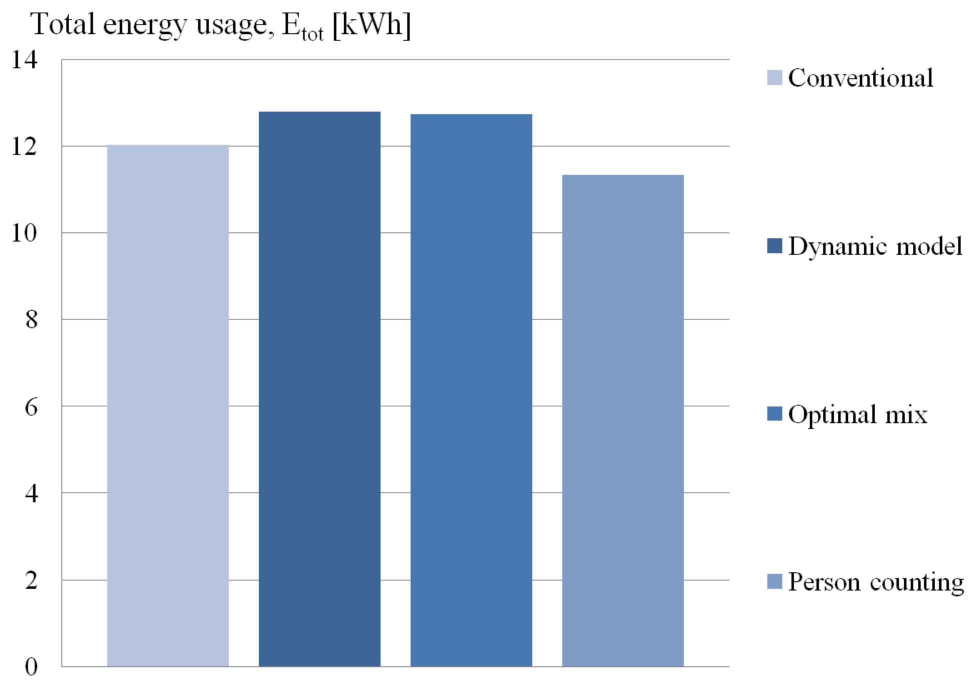


Figure 8.10 Total energy usage when the different SAT control strategies are used. Setup: light structure building, winter ambient climate

The resulting energy and peak power indicators when the personal counting strategy is used instead of the conventional OAT based strategy are presented in the tables below. The results correspond to the light building setup.

Table 8.3 Difference in energy usage by using the person counting SAT control strategy instead of conventional OAT based control. Test setup: Multi-zone, light building structure, winter and summer outside conditions

	Heating energy savings [%]	Cooling energy savings [%]	Electrical energy savings [%]	Total energy savings [%]
Person counters. Summer ambient conditions	0	1	0	1
Person counters. Winter ambient conditions	4	17	0	6

Table 8.4 Difference in required peak power by using the person counting SAT control strategy instead of conventional OAT based control. Test setup: Multi-zone, light building structure, winter and summer outside conditions

	FCU power savings [%]	AHU power savings [%]	Electrical power savings [%]
Person counters. Summer ambient conditions	-10	0	0
Person counters. Winter ambient conditions	0	12	0

8.4 Conclusion and discussion

In general, the saving potential shown in this study was scarce. On the other hand, an important conclusion is that the number of people inside an office building is a better indicator for SAT than the OAT.

Challenges of an SAT control strategy
The performance of a SAT control strategy is dependent on how well the resulting supplied cooling energy from the ventilation system is synched with the demands of the rooms. Since the magnitude of the internal disturbances is varying from one room to another the demands differ throughout the building. Hence, the performance of a SAT strategy is improved if the supplied cooling energy in average is better matched with the demands of the rooms.

The main challenge of SAT control is to fulfil a trade-off between serving rooms with large cooling demands and to avoid creating heating demands in disfavoured rooms, i.e. in rooms with large demands for fresh air but relatively low demands for cooling. Empty rooms are on the other hand never an issue in this work since the supply air is shut off when the room has been empty for a while.

8.4.1 Person counting strategy

The total energy savings in the heavy building setup when the person counting strategy was used was 4 % independent on ambient climate. In the light structure setup, the corresponding values were 1 and 6 % for summer and winter ambient conditions respectively. These savings derive from a reduced thermal energy supply while the electrical energy usage (excluding pump work) was unaffected by the SAT control strategy.

Below, the effects that derive from using the person counting strategy instead of the conventional strategy are discussed. The text is focusing on one setup at the time.

8.4.1.1 Heavy building structure

Summer ambient conditions
The savings indicated in the heavy structure summer setup derives from a reduced cooling energy usage of the AHU. But at the same time, the cooling energy used by the FCU was increased. This effect derives from that the SAT in general is higher by using the person counting strategy instead of the conventional SAT control.

This means that the person counting strategy has a similar effect on the allocation of cooling energy as the FF-control systems in section 4.3.2.2 for IAQ control. That is, part of the cooling energy supply is shifted from AHU to FCU. This shift is desirable in this work since the FCU is designed for thermal climate control while the ventilation system is designed for IAQ control.

Winter ambient conditions
The savings indicated in the heavy structure winter setup derives both from a reduced cooling energy usage by the FCUs and a reduced heating energy usage by the AHU. However, the heat supplied by the FCUs was increased a bit. The reduced heating energy usage of the AHU indicates that the SAT in general is lower when the person counting strategy is used instead of conventional SAT control.

The fact that the person counting strategy resulted in a total energy saving during winter indicates that the SAT determined by conventional control is too high compared to what's demanded. When the conventional control was used, both the cooling energy usage of the FCUs and the heating energy usage of the AHU were higher compared to when the person counting strategy was used. Hence, some of the heat that is supplied centrally has to be cooled off at room level. However, the increased heating energy usage of the FCUs when the person counting strategy was used indicates that the disfavoured rooms are affected negatively by the reduced SAT.

8.4.1.2 Light building structure

The results from the light structure setup show that the most critical situation for the person counting strategy is when a low occupancy period is preceded by a high occupancy period. In these situations, the synch between the SAT supplied by the person counting strategy and the demand is most out of phase. As discussed below, this effect explains the difference in performance between heavy and light structure setups since the impact was larger in the light case.

Summer ambient conditions
During summer, most of the room temperatures in the heavy building are evolving somewhere in the middle of the temperature dead-band since the structure provides cooling during most part of the day. However, when the disturbances become high during the day, the room temperatures in the light structure case are increased relatively fast up to the boundary of 22 °C. When a low occupancy period then is preceded by a high occupancy period, the SAT is instantly increased. In turn, the increased SAT results in an instantly increased heating supply from the ventilation system since the supply air flow rate still is high. If the temperature of the room is close to 22 °C, the result is an overshoot which in turn

is met by an increased cooling supply by the FCU. The corresponding effect in heavy structure setups are smaller, partly due to that few rooms temperatures are close to the upper limit of the dead-band and partly since the instant SAT increase is dampened by the structure.

Winter ambient conditions
The savings in the light structure winter setup are due to the same effects as in the heavy building winter setup discussed above. However, both the cooling and heating energy supply of the FCU were reduced more in the light structure case when the person counting strategy was used. The reason is that during winter, most of the rooms evolve somewhere in the middle of the temperature dead-band and not directly on the upper limit. Hence, the instant increase of the SAT when a low occupancy period is preceded by a high occupancy period does not result in over-shoots in the same way as during summer ambient conditions. The difference in performance between the light and heavy structures dependents on that the switch from high to low occupancy results in a heating demand in the light building relatively fast. This means that the instantly increased SAT, generated by the person counting strategy, is better synced with the demand in the light structure compared to the heavy structure.

8.4.1.3 Improvement suggestions

The person counting strategy for SAT control is not perfect and it is believed that further improvements of performance are possible. The most direct improvement would be to determine the breaking temperatures in the FF-model more accurate. In the current version, they are more or less set arbitrary.

A problematic situation was when a low occupancy period was preceded by a high occupancy period since the response of the person counting strategy was an instantly increased SAT which resulted in a heat supply peak from the ventilation system. However, the negative effects of this situation might be reduced by combining the person counting SAT control strategy with the FF- system for local IAQ control selected in chapter 5. The reason is that the response of the local FF-controller when a low occupancy period is preceded by a high occupancy period is to instantly decrease the supply air flow rate. Hence, the unwanted heating peak from the ventilation system might be avoided.

8.4.2 Rejected strategies

The optimal mix and the dynamic model strategies were rejected from the final results due to inconsistent results. Both were feasible during summer conditions, and both their energy and peak power indictors were slightly better than what was achieved by the person counting strategy. However, their performance during ambient winter conditions dropped below the performance of the conventional control strategy.

The reason to why the two rejected strategies performed well during summer and worse during winter is mainly due to an aggravated match between supply and demand during winter. During winter, both of the rejected strategies resulted in a higher SAT than what was returned from the conventional OAT based strategy.

The result was an increased heating energy usage of the AHU at the same time as the cooling energy usage of the FCUs was increased. This means that the increased SAT resulted in an increased local cooling demand. One positive effect of the two rejected strategies was that the heating supplied by the FCUs was slightly reduced. However, this reduction was not close to the increased heating energy usage of the AHU. This means that the increased SAT does not occur at times when the building has a heating demand but rather when there is a cooling demand.

Dynamic model strategy
The dynamic model strategy returns the highest SAT required in the building. In the study, it was indicated that this approach might be suboptimal and instead of avoiding heating in a few rooms the cooling demand is increased in many. This problem is on the other hand avoided during summer since none of the rooms requires heating during day-time.

Another problem with the dynamic model strategy is that the dynamics of the room air is approximated in the control-model, which was explained in section 4.3.2.4. The effect might be that the control strategy, which due to the approximation mainly considers the effect of the building structure, constantly is out of sync with the room air temperature. Hence, the supplied SAT might be even higher than what the most disfavoured room requires.

Optimal mix strategy
During winter, the optimal mix strategy returns that the entire cooling demand should be supplied by the FCUs. This means that the resulting SAT is set as high as possible, i.e. 18 °C, during the main part of the simulations. The main problem is that many of the rooms still has a cooling demand and the high SAT only aggravates the situation. Hence, this approach is not suitable for winter conditions.

During summer, the SAT is instead decreased since the strategy returns that a larger part of the cooling energy should be put on the AHU. However, if the OAT would increase further, the same problems which occurred during winter would also occur during summer. That is, the strategy returns that the entire cooling demand should be supplied by the FCU which means that the SAT is maximized. Hence, the optimal mix approach must be treated in a different way.

The main problem with this strategy is the implementation in HVAC-systems with hygienic ventilation since the supply air flow rate is locked by the CO_2 levels. When the strategy calls for a decreased cooling supply from the AHU, this is met by an increased SAT. If the HVAC-system instead was equipped with air-based heating and cooling, the situation would be different since then the supply air flow rate could instead be decreased. Further, since the maximum cooling power from the ventilation system is limited by the dimensioning hygienic flow rate and the minimum SAT, many of the mixes that the strategy implies are impossible to achieve.

9 Combined effects

In this chapter, the combined effect of the control strategies that were selected in the previous studies is presented. In the evaluation, the medium structure building presented in chapter 3.4.1.1 is also introduced along with the usual light and heavy structures.

Both control strategies for central and local control of the HVAC-system have so far been selected for further evaluation;

- In chapter 5, it was shown that the FF-controller consisting of person counters and non-dynamic signal processing had the largest potential of managing the control on room level, independent on the setup.
- In chapter 6, it was shown that this controller only should manage the IAQ control while the thermal part of the room should be managed by FB-controllers. Also this result was independent on the setup.
- Chapter 7 was dedicated to night-mode control and the only strategy that was selected was to shut off the HVAC-system during the night in heavy buildings during winter ambient conditions.
- In the preceding chapter, alternative method for control of SAT was evaluated. The results showed that by utilizing the number of people inside the building, the performance was increased compared to the other strategies. Also this conclusion was independent on test setup.

9.1 Method

The combined effect of the control strategies mentioned above is evaluated using the same multi-zone platform and disturbance profiles as were used in chapter 6. The strategies are also evaluated for the same condition parameters as in chapter 6, but in this case also the medium building structure is included in the test setups. This means that the combined strategies are tested in light, medium and heavy building structures during summer and winter ambient climate. The simulation time stretches over one working-week including 5 days and nights.

The strategies were combined in the most straightforward way; all the selected strategies were set out to manage the control task which they were designed to manage. The evaluation was primary focused on the combination of local FF-control of IAQ and the person counting SAT control strategy since the night-mode strategy was infeasible for most setups. Night-mode was only used in the heavy building winter setup and then the HVAC-system is shut off during night. In all summer setups, as well as the light and the medium structure winter setups, no distinguishing was made between night- and day-mode.

Just as before, the performance of the control system was measured by the now familiar indicators for energy usage and peak power. The evaluation was performed by comparing the combination of selected strategies to a combination of the conventional strategies that were introduced in the previous studies. These conventional strategies are;

- FB-controllers for IAQ and thermal control on room level (chapter 6)
- No distinguishing between night- and day-mode (chapter 7)

- OAT based central SAT control (chapter 8)

The only condition for comparability between the control strategies is that the CO_2 concentrations inside the rooms are not allowed to breach 1000 PPM.

9.2 Results

In the text below, the performance of the combined selected strategies for central and local control are presented. The results primary consists of the energy and peak power indicators presented in chapter 1.3.3, but the resulting total energy usage of the HVAC-system, when the different strategies are used, are also presented as a complement.

9.2.1 Total energy usage

In figure 9.1, 9.2 and 9.3 the total energy usage for the heavy, medium and light structure setup is presented respectively. Each figure presents the total energy usage during the summer and winter ambient conditions, both when the combined selected strategies and the conventional strategies are used. Note that the absolute values should not be considered as relevant in this context. The purpose of the figures is instead to illustrate the difference between the strategies.

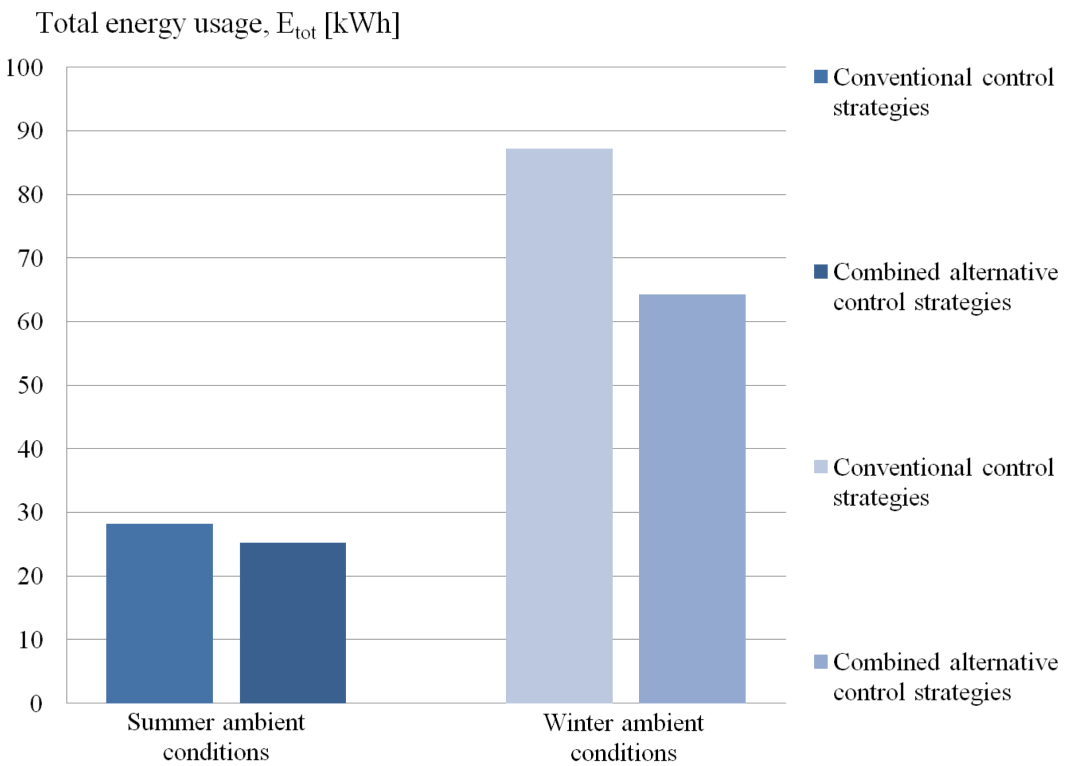


Figure 9.1 Total energy usage when the combined alternative control strategies as well as the conventional control strategies are used. Setup: heavy structure building, summer and winter ambient climate

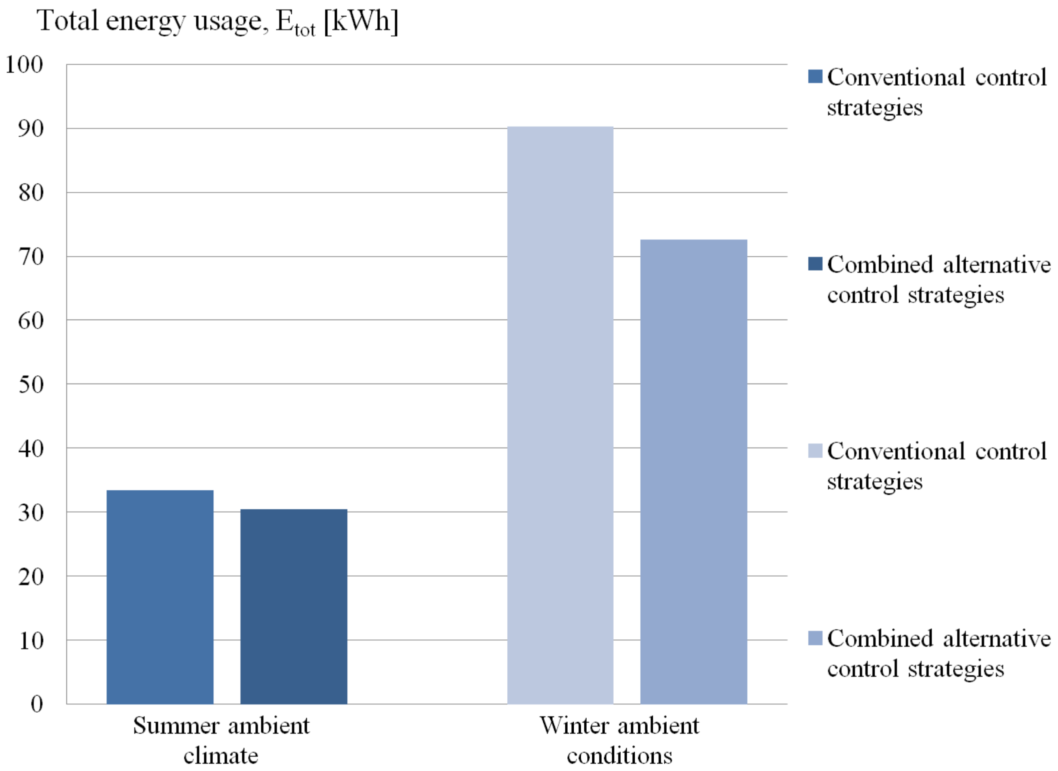


Figure 9.2 Total energy usage when the combined alternative control strategies as well as the conventional control strategies are used. Setup: medium structure building, summer and winter ambient climate

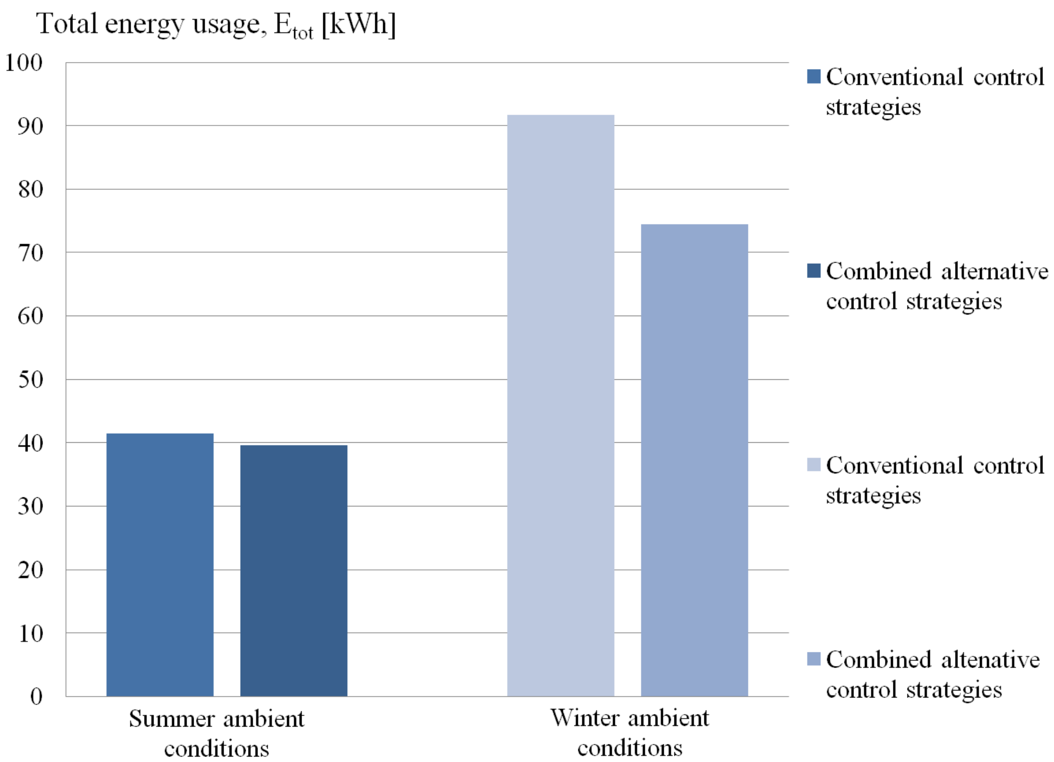


Figure 9.3 Total energy usage when the combined alternative control strategies as well as the conventional control strategies are used. Setup: light structure building, summer and winter ambient climate

9.2.2 Energy usage and peak power indicators

The performance indicators related to the control systems consisting of the combined selected control strategies are presented below. The results are presented in four tables. In the first and third, the energy indicators regarding heavy, medium and light structures are presented for summer and winter ambient conditions respectively. In the second and fourth the corresponding peak power indicators are presented.

Table 9.1 Difference in energy usage by using a combination of the selected strategies. Test setup: Multi-zone, summer outside conditions

	Heating energy savings [%]	Cooling energy savings [%]	Electrical energy savings [%]	Total energy savings [%]
Heavy structure	15	-2	57	11
Medium structure	13	-3	57	9
Light structure	9	-4	55	5

Table 9.2 Difference in required peak power by using a combination of the selected strategies. Test setup: Multi-zone, summer outside conditions

	FCU power savings [%]	AHU power savings [%]	Electrical power savings [%]
Heavy structure	9	27	54
Medium structure	9	27	54
Light structure	7	27	54

Table 9.3 Difference in energy usage by using a combination of the selected strategies. Test setup: Multi-zone, winter outside conditions

	Heating energy savings [%]	Cooling energy savings [%]	Electrical energy savings [%]	Total energy savings [%]
Heavy structure	25	1	54	26
Medium structure	19	-24	53	20
Light structure	20	-10	52	19

Table 9.4 Difference in required peak power by using a combination of the selected strategies. Test setup: Multi-zone, winter outside conditions

	FCU power savings [%]	AHU power savings [%]	Electrical power savings [%]
Heavy structure	-111	42	54
Medium structure	1	42	54
Light structure	0	43	54

9.3 Conclusions and discussion

The impact on the energy usage when different energy efficiency measures are a combined can be divided in three idealized scenarios;

1. The individual measures are enhancing each other impacts so that the total performance is larger than the sum of the individual contributions.
2. The total performance is the same as the individual contributions combined. This scenario might occur if the impacts of the individual measures are unaffected by each other.
3. The total performance becomes lower than the combined individual contributions since the impact of the individual measures are limited by each other.

9.3.1 Effects on energy savings

For the heavy building setups, the combination of the selected strategies resulted in a total energy savings of 26 and 11 % for winter and summer respectively. In the light building setups, the corresponding values were 19 and 5 % respectively.

The result from this study shows that a combination of the first and the third scenario above occur when the selected strategies are combined in this work. However, the interplay between these two scenarios resulted in that the performance of the combined strategies is close to fulfil the second scenario above.

The third scenario above occurs mainly since both the local FF-control of IAQ and the central control of SAT have an effect on the energy used by the AHU. Since the supply air flow rate is reduced by about one third when FF is used for IAQ control on room level, the potential energy savings of SAT control is reduced.

The first scenario above occurs due to the effect discussed in section 8.4.1.3 in connection to the SAT control strategies evaluation. It was determined that the main problems of the person counting SAT control strategy occurred when a low occupancy period was preceded by a high occupancy period. The results from the current chapter shows that the negative effects of this situation was reduced by combining the person counting SAT control strategy with the FF-control of local IAQ. The reason is that the instantly increased SAT, when a low occupancy period is preceded by a high occupancy period, is met by an instantly reduced supply air flow by the local FF-controller for IAQ. This effect especially has an impact on the energy savings in light building setup since the negative effects of the SAT control strategy was large in this case.

9.3.2 Effects on peak power savings

Both the peak power savings of the central fan and the AHU during summer was unaffected by the combination compared to when only the local FF-control system was used. But the power savings of the FCU was aggravated a bit during summer conditions. The AHU power savings during winter conditions, on the other hand, was increased by about 10 % for both light and heavy structures compared to when only FF for local control was used.

These effects on the peak power are explained by that constant values of SAT were used in chapter 7. The differences between the results shown in this chapter and the results shown in chapter 7 are either reflecting an increased or decreased SAT compared to these constant values.

- The reduction of AHU peak power during winter is due a decreased average SAT compared to the constant values used during the study in chapter 7.
- The reduction of AHU peak power during summer is due to an increased average SAT compared to the constant values used during the study in chapter 7.
- The increased peak power of the FCUs during summer condition is also due to the increased average value of the SAT. Hence, a slight shift in allocation of cooling energy occurs and a larger part is put on the FCUs.

As can be seen in table 9.4, the night-mode included in the heavy building setup increases the required peak power of the FCUs substantially. Even though this increase has been reduced compared to the results in chapter 7, the utilization of night-mode is questionable.

9.3.3 Influence of building thermal mass

In this study, also the medium building structure was introduced for the first time. The results showed that the value of the total energy indicator for the medium structure setup was in between the corresponding values for the heavy and light structure setups. Also the reduction of peak power coincides with the corresponding results of the light and the heavy building setups. Hence, the influence of the building structure on the potential of the selected strategies seems to be consistent.

10 Sensitivity analysis

The most sensitive control strategy selected in this work is the FF-control of IAQ on a local level. The reason is that this strategy requires the number of people inside the room (or rather the CO_2 emission per person) which is not commonly measured. It was assumed that a motion detector was used in the office room setups and that the response of a CO_2 sensor, preferable in combination with a motion sensor, was used in the meeting room setups. The CO_2 emission rate per person was set as a parameter in the FF-filter.

Even thought the exact number of people can be determined by the FF-sensor, the corresponding CO_2 emission can only be approximated. The reason is that the CO_2 emission from people is dependent on number variables, for example activity level, which are more or less unmeasurable[22]. The aim of this study is to determine how the performance of the selected IAQ FF-control strategy is influenced by these limitations.

Also the selected SAT control strategy is based on person counters but in this case the strategy only requires the number of people in the building and not the CO_2 emission rate. Since the strategy is based on a number of local sensors, presumable estimation errors might also be evened out to some extent. Never the less, also the sensitivity of this strategy is analysed indirectly in this chapter.

In this study, the sensitivity of an MPC controller is also studies in parallel. The structure of the MPC controller corresponds to the description in section 2.3 and it was designed according to appendix F. The MPC is based on the same FF-model as the selected FF-controller for local IAQ control. This means that the MPC and FF-controller that are tested in this study are very similar. The main difference is that the MPC is generating the control signals by minimizing the objective function while the control signals of the FF-controller are generated by solving the equations in the FF-filter.

10.1 Method

As mentioned, an estimation of the number of persons is only relevant in the meeting room since the corresponding occupancy factor to a response of a motion sensor is unknown. For that reason, the platform used in this study is the single-zone meeting room presented in section 3.4.1 and used in chapter 5.

The disturbances are simulated as a step of magnitude two in presence. Initially, one person is present and the period until the step occurs is used as stabilization. The initial CO_2 concentration before the step occurs is 1000 PPM. The thermal climate is controlled by a FB control loop and the IAQ by either the MPC or the selected FF-controller in combination with a FB-controller with a setpoint of 1000 PPM.

Parameters
The sensitivity analysis is studying the influence of three parameters on the performance of the selected FF for IAQ control. These parameters are;

- The estimation error of CO_2 emission rate per person

- The delay time until the person counter returns an accurate measurement
- The influence of un-modelled infiltration of outside air to the room

The study is conducted by evaluating the change of control performance with respect to stepwise perturbations of these parameters.

By including the CO_2 emission rate per person in the sensitivity analysis, both the influences of errors allocated to the FF-sensor and to the FF-filter are studied. By introducing small deviations between the value set in the FF-filter and the actual value used in the room model, the effect of FF modelling errors are studied. By increasing this deviation step-wise, the model error eventually reaches the same magnitude as would be a result of a measurement error of the person counter.

The infiltration rate has so far been left out in this work. This was due to the lack of accurate measurement methods, which means that it can't be assumed that the infiltration rate is known to the FF-filter in real applications. Instead, the influence on the selected FF for IAQ control is studied in this part of the work. In the sensitivity analysis, the infiltration rate through one of the long-sides of the room was varied step-wise between 0 – 0.4 $l/(s \cdot m^2)$, which corresponds to a total infiltration flow rate between 0 – 5 l/s. These values can be compared to the guidelines in Belok[10] which states that the infiltration should be < 0.3 $l/(s \cdot m^2)$ for a pressured difference of ±50 Pa.

MPC

The MPC is included in the analysis of the estimation error CO_2 emission rate per person and the delay-time of the person counter. As mentioned, the MPC is based on the same model as the FF-controller which means that these two controllers are similar from a performance point of view. The purpose of including the MPC in this part of the work is instead to compare the robustness between the MPC and the selected FF-controller. As described in section 2.3, the control signals of an MPC are based both on state measurements and measurable disturbances just like in the FF case. However, in the FF case, these two contributions derive from separated sub-controllers while they derive from the same algorithm in the MPC case. From that point of view, an MPC can be regarded as a merged FF- and FB-part which presumable would result in a larger robustness compared to the FF-controller.

10.1.1 Experimental study

The magnitude of the expected delay-time of the person counter is of course dependent on which measurement method that is used. In this work, it was assumed that the response of a CO_2 sensor is used. This method is considered as the most realistic approach and then the delay-time is a relevant parameter to include in the sensitivity analysis.

In connection to the evaluation of the delay-time sensitivity, an experimental study was performed to determine the actual delay-time until a strong correlation between the number of persons in a room and a CO_2 sensor response can be determined[34]. These results are then used to connect the sensitivity analysis to the experiment by using the measured delay-time as a bench-mark in the conclusion. This means that the experiment does not aim to determine the actual function between CO_2 level and number of people which would be required by the person counter in this work.

10.1.1.1 Method

The experimental study consists of a number of tests which were all performed in a laboratory of Building Services Engineering at Chalmers University of Technology. The tests were performed in a room about twice as large as the meeting room used previously in this work. The room in which the experimental study took place is supplied with fresh air of constant temperature using mixed ventilation via roof mounted diffusers. The supply air diffusers are used to control the flow rate according to a setpoint and the exhaust diffusers are controlled to achieve balanced ventilation. During the test, the following variables were measured with a sampling time of 10 s and the number of people participated in each test was recorded;

- The room CO_2 concentration
- The supply air CO_2 concentration
- The supply air flow rate
- The exhaust air flow rate

The procedure of the tests was to measure the response of the room CO_2 sensor located in the exhaust air duct when a various number of persons entered the room. The tests were also repeated for various setpoints of the supply air flow rate. Hence, the parameters in the test setups are the supply air flow rate and the number of people entering.

Test facility
The test facility consists of a room with 5.6 m in width and 6.2 meters in length according to figure 10.1. The height of the room is 2.4 m and the total volume is then 78.8 m^3.

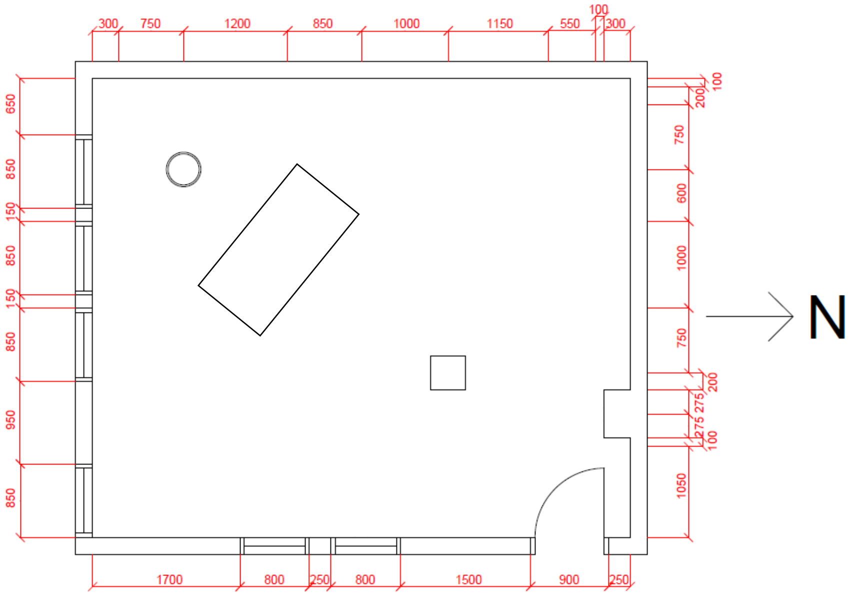


Figure 10.1 Layout of the test facility.

The circle in figure 10.1 denotes the supply air diffuser and the square the exhausted air diffuser. The supply air stream is guided in the opposite direction of the exhaust air to achieve mixed ventilation.

The participating people were of the same gender and about the same age and they were instructed to perform office related activities throughout the tests. The rectangle in between the diffusers marks the area in which people were allowed to dwell. The purpose was to reassemble the area which people is expected to occupy in normal office-rooms, relative to how the diffusers normally are located; a common procedure is to place the supply air diffusers close to the areas were people are expected to be situated and the exhaust air diffuser close to a wall. For example, if the people were allowed to occupy one of the corners the distance to the diffusers are much larger than what could be expected in a normal office room.

Due to the configuration of the test facility, the size of the test room approaches the size of the meeting room model from a ventilation point of view. The reason is that the area of the test facility that is not actively covered by the ventilation system is almost half of the total area. This was verified by mapping the flow distribution in a smoke-gas test and a sketch of the observations are presented in figure 10.2.

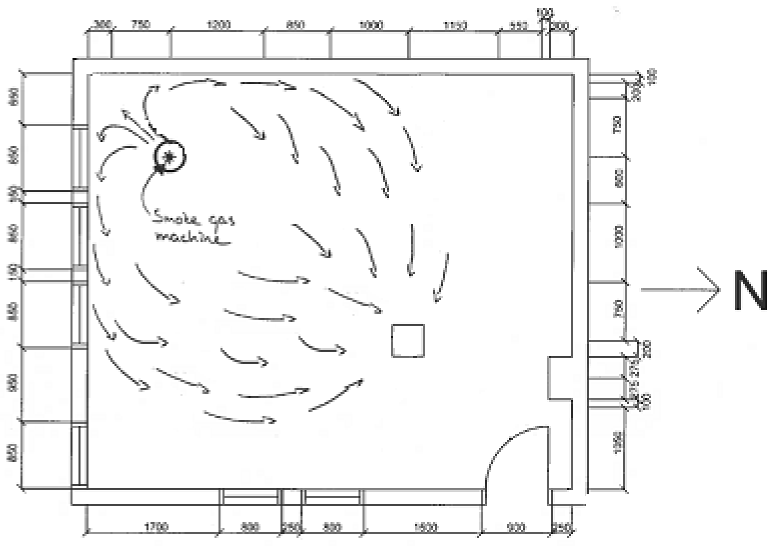


Figure 10.2 Distribution of ventilation air in the test facility

10.1.1.2 Verification

The experimental study began with a test to verify the measurement setup. Two persons entered the room and the measurements up to steady-state were compared to the corresponding analytical solution presented as equation 7.1 in section 7.2.3. By doing so the following variables were verified;

- The measured room CO_2 concentration
- The measured supply air CO_2 concentration
- The measured supply air flow rate
- The measured exhaust air flow rate
- The estimated CO_2 emission per person

The analytical solution is assuming complete mixing of room and supply air which means that there are no CO_2 concentration gradients in the room. This was emulated as far as possible in the verification part by placing portable fans in the test facility. The results are shown in figure 10.3 and as can be seen, the agreement, both related to transient and steady-state conditions, is satisfying.

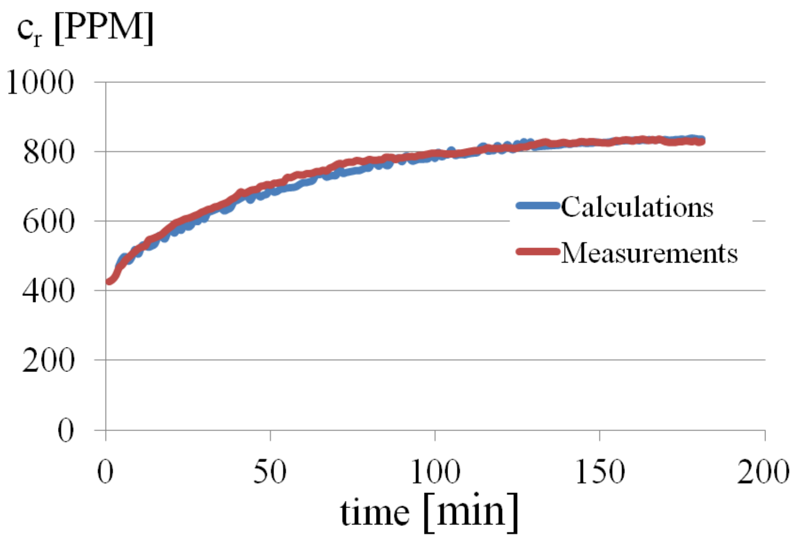


Figure 10.3 Verification of the measurement setup

10.1.1.3 Results

Each test setup was repeated several times to check the validity of the results. In all cases, the measured results corresponded to each other. In the text below, only one of the tests associated to each test setup is presented.

The results from the experimental study are presented in three figures below. Each figure is associated with a certain supply air flow rate. The first to 22 l/s, the second to 43 l/s and the third to 66 l/s. The figures contain the evolution of room CO_2 concentration for different number of people as a function of time. At time 0, people are entering the room and measurements were gathered for 4 minutes. All setups began with an initial CO_2 concentration of 410 PPM.

The intended interpretation is that the number of people can't be estimated until the plots in one figure have been separated from each other. Then, there exists a correlation between how much the CO_2 concentration has been increased and how many people that entered the room.

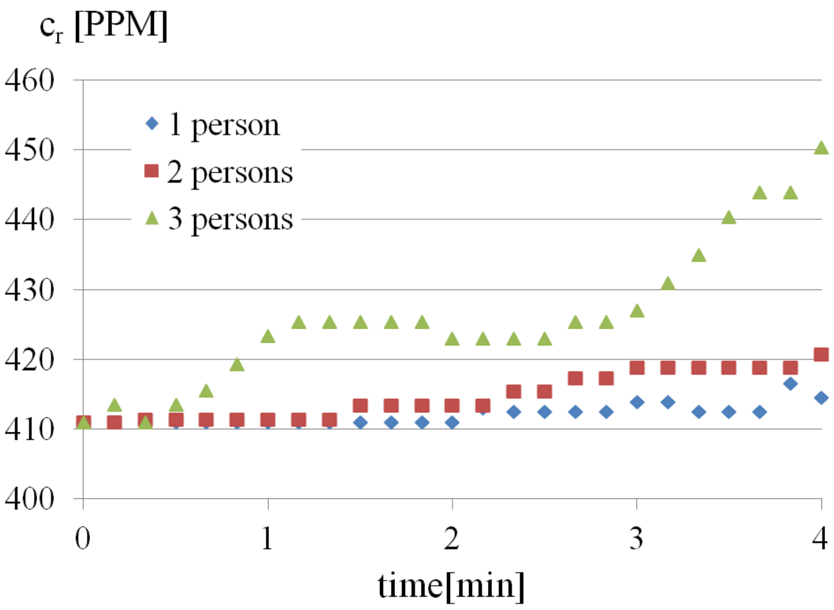


Figure 10.4 CO_2 sensor response as a function of time for a supply air flow rate of 22 l/s

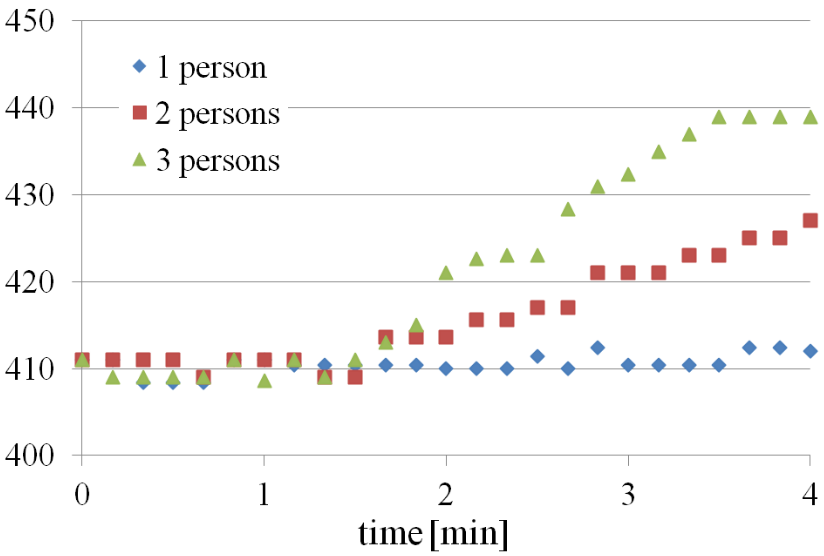


Figure 10.5 CO_2 sensor response as a function of time for a supply air flow rate of 42 l/s

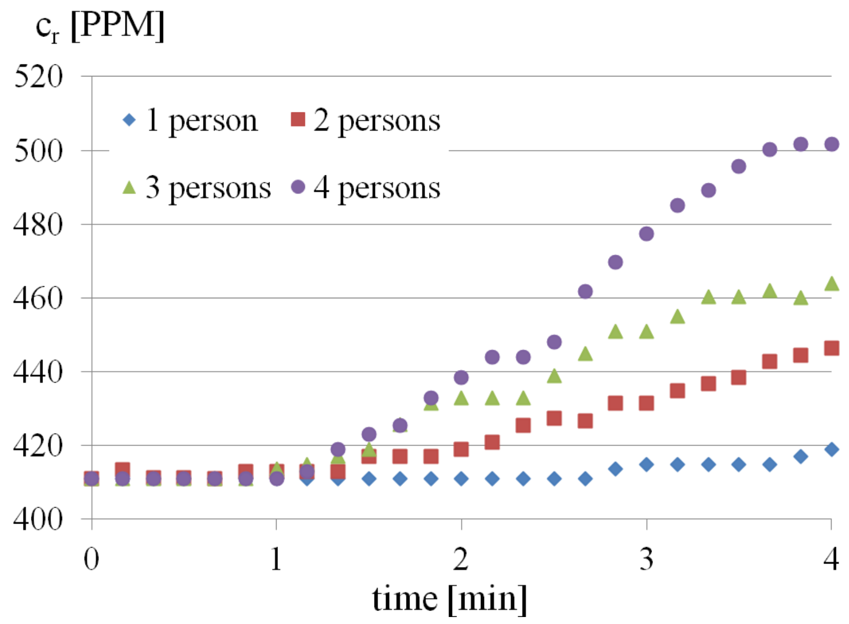


Figure 10.6 CO_2 sensor response as a function of time for a supply air flow rate of 66 l/s

As can be seen in the figure above, an estimation of the number of people using a CO_2 sensor is possible after about 3 minutes. The expected accuracy of this estimation is somewhat dependent on the supply air flow rate which can be seen by comparing figure 10.4 (22 l/s) with figure 10.5 (42 l/s) and 10.6 (66 l/s). In figure 10.4, the different occupancy cases are not very distinguished after 3 minutes while there is a clear difference in figure 10.5 and figure 10.6. In the 22 l/s case, the plots are more separated after 4 minutes, but the difference between 1 and 2 persons are also after that amount of time quite small.

Example of implementation

Below, parts of the results from the experimental study are presented in an alternative way. Figure 10.7 shows an example of how the measurements could be used to convert a CO_2 sensor in to a person counter.

The figure presents four different regression curves between the number of people in the room and the response of a CO_2 sensor. Each curve is connected to the CO_2 increase that was measured after a certain time from that the people entered the room. As can be seen, the correlation between the number of people and the CO_2 response are weak after one minute. After two minutes, the correlation is strong but since the CO_2 difference between 1 and 2 persons only is about 5 PPM, the measurements system must be very accurate to be able to estimate the number of persons. After 3 minutes, the regression is linear and an accuracy of 10 PPM is needed to distinguish between the different occupancy cases. Hence, also this example shows that a person counter delay-time of 3 minutes could be expected.

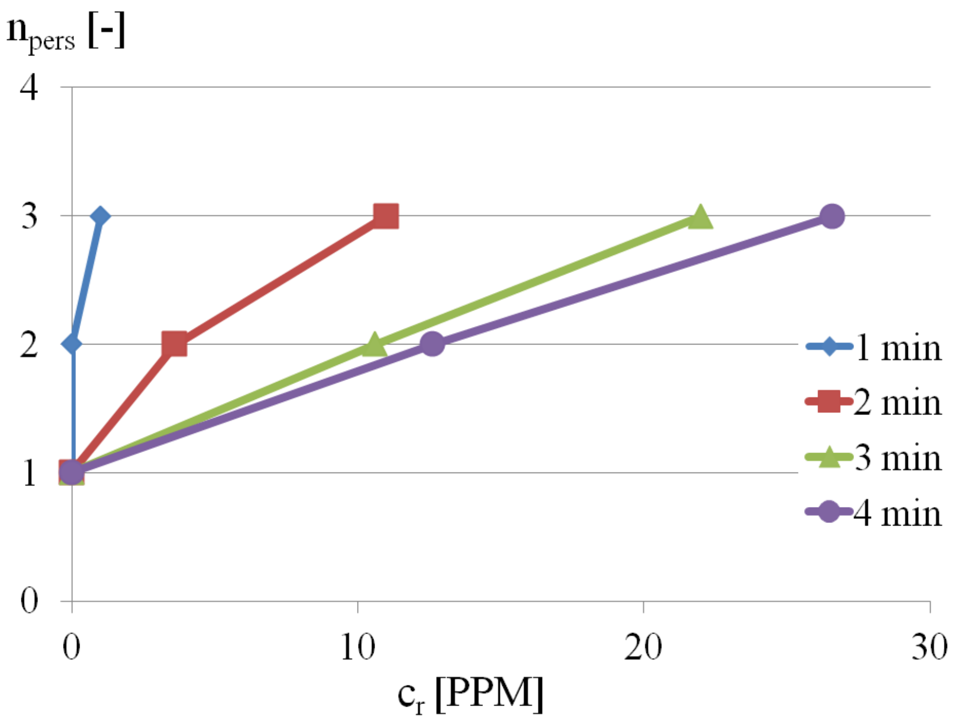


Figure 10.7 Number of people versus the CO_2 response after 1, 2, 3 and 4 minutes for a supply air flow rate of 66 l/s

10.2 Results

In the following section, the results from the sensitivity analysis of the selected FF-controller for IAQ control are presented. The results are presented in the same order as indicated in section 10.1; FF-filter modelling error regarding CO_2 emission rate which also includes FF-sensor measurement errors, person counter time-delay and un-modelled infiltration flow rate.

Modelled CO_2 emission rate and person counter measurement errors
In figure 10.8, the sensitivity of the IAQ FF-controller regarding modelling errors of CO_2 emissions rate per person is presented. Each square represents the results of one simulation. The results are illustrated as the control error (Δc_r) as a function of the total modelling error. The modelling error is presented as the difference between the actual CO_2 emission rate per person set in the room model $\left(\dot{M}\right)$ and the value set in the FF-filter $\left(\hat{\dot{M}}\right)$.

The CO_2 emission rate of the room model was set to 18 l/(h·pers). This means that a modelling error of magnitude ± 18 l/h, which is marked by dots in the figure below, corresponds to a person counter measurement error of ± 1 person. In section 10.1.1, it was shown that a measurement error of this magnitude was relevant to study since it could occur when low supply air flow rate was used (figure 10.4).

The positive control errors of the FF-controller in figure 10.8 are referring to overshoots of 1000 PPM. These control errors are dynamic and are decaying to zero rather quickly since the FB-controller steps in. All control errors of the MPC as well as the negative control errors of the FF-controller are referring to static errors. Hence, these are constant and not time-dependent.

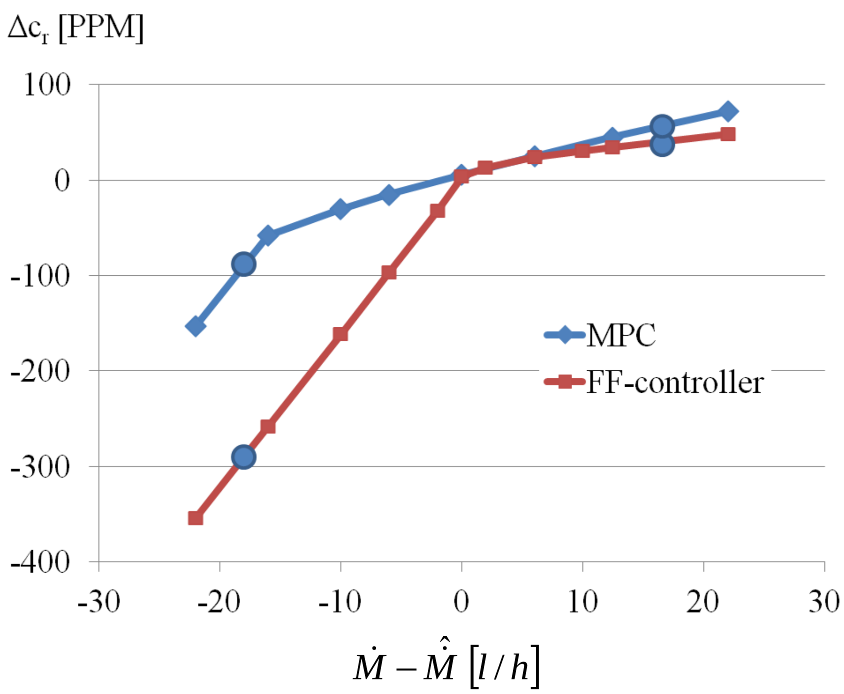


Figure 10.8 Control error as a function of CO2 emission rate per capita total model error

Delay-time of the person counter
In figure 10.9, the sensitivity of the IAQ FF-controller regarding the delay-time of the person counter is presented. Each polygon represents the results of one simulation.

The figure shows the dynamic control error as a function of the delay-time of the person counter. Hence, the error of the FF-controller and MPC will decay to zero as time after occupancy step passes by. Also, the overshoot of the FB-controller is included in the figure which of course is independent on the delay.

Most interesting is the delay between 3-4 minutes which is marked in the figure. According to the experimental study, this is the time required to return an accurate estimation when a CO_2 sensor is used as a person counter.

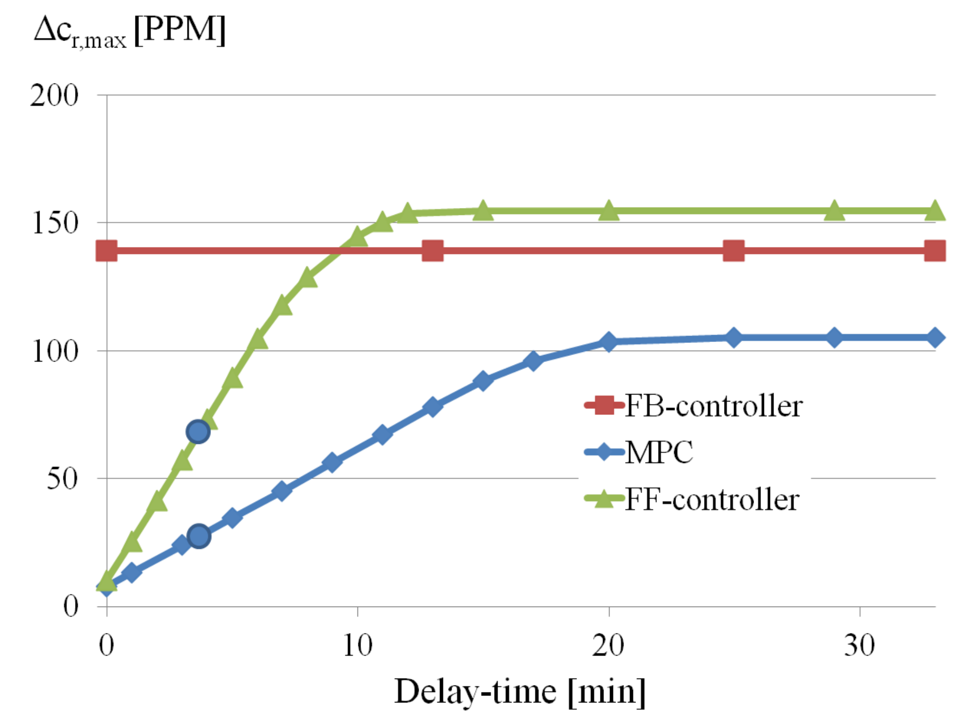


Figure 10.9 Dynamic control error as a function of delay-time related to person counter

Infiltration flow rate
The last parameter in the sensitivity analysis is the infiltration rate and the corresponding result is presented in figure 10.10. Each square represents the result of one simulation.

This figure shows the relation between the static control error (Δc_r) and the specific infiltration rate through one of the long-side walls. Since the infiltration rate is not taken into account by the FF-model, the resulting supply air flow rate will be larger than the demand in cases when the infiltration rate is larger than zero which results in the static control error.

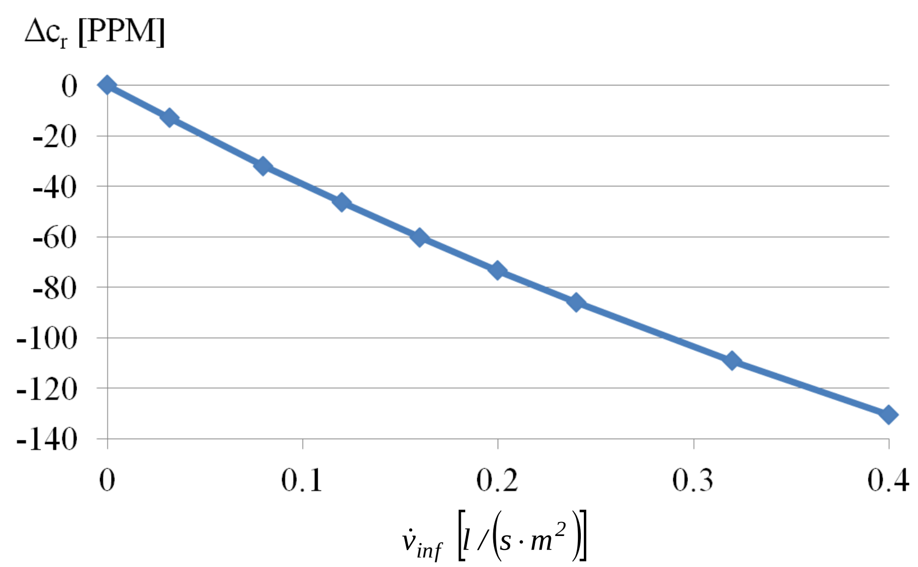


Figure 10.10 Static control error as a function of the specific infiltration rate through one of the long-side walls.

10.3 Conclusions and discussion

In general, it is obvious that the MPC is less sensitive to both model errors and person counter delay-time. But there are some considerations that preferable are taken into account. These are discussed below.

10.3.1 CO_2 emission rate modelling errors

The impact of the CO_2 emission rate modelling errors that results in positive control errors of the FF is relatively small. Instead, it is primary the model errors that results in negative control errors of the FF-controller that has a significant aggravating effect on the control performance. For example, the error that corresponds to a measurement error of -1 person results in that the system is stabilized at 710 PPM instead of 1000 PPM. This is about the same setpoint as needed for the FB-controller to avoid breaching 1000 PPM which means that after this point the control is aggravated by using FF-control.

In general, model errors have a small impact on the performance of the MPC. However, one important consideration is that all control errors of the MPC are static. For the FF-controller, the same is only true for negative errors. This aspect is of interest since static control errors are avoided by FB-controllers with I-action and hence, static errors produced by alternative control strategies should be considered as an aggravation.

Explanation to positive and negative control errors
The positive errors of the FF-controllers derive from that the supply air flow rate generated by the FF-part is too low. This means that the setpoint is breached, but after that, the FB-controller steps in and brings the system back to 1000 PPM. That is, the error is dynamic.

For negative errors, the FF-controller returns a supply air flow rate that is too large. This means that the CO_2 concentration never reaches the setpoint and the FB-controller is unable to correct the error. This explains why the breaking point of the curve associated with the FF-controller in figure 10.8 occurs at a control error of zero.

The explanation to why all of the control errors, associated with the MPC, are static is that this control algorithm lacks I-action. Hence, the MPC controller in this study can be compared to a P-controller with an optimal gain, which means that the MPC will bring the system exactly to the setpoint as long as the control model is correct. When errors in the model arise, control errors are also introduced. However, the performance of the MPC is still high since the control signal is in each time step based on measurements of the current CO_2 concentration. Hence, also negative errors are managed relatively well which is seen in figure 10.8 as a delayed breaking point compared to the corresponding breaking point associated to the FF-controller. The breaking point of the MPC is reached when the model error is larger than the evolution of the system during one time-step. Then, the control model does not contribute to the control anymore which means that the entire control is managed by the FB-part.

10.3.2 Delay-time of the person counter

As mentioned, all control errors related to the person counter delay-time are dynamic. Hence, both the FF-controller and the MPC will eventually reach the setpoint.

Since the initially CO_2 concentration before the occupancy step is 1000 PPM, the delayed person counter signal will result in an overshoot of this setpoint since also the response of the controllers are delayed. The MPC utilizes the current CO_2 concentration as well as the output from the person counter which means that the delayed measurement in general has a smaller impact. Further, the MPC uses the measured CO_2 concentration to determine the exact supply air flow rate which is needed to obtain the setpoint which means that it also performs better than the FB-controller.

In the MPC case, a delay of 20 minutes corresponds to a breaking point in figure 10.9 since the CO_2 concentration has already reached steady state when the number of persons is returned by the person counter. Hence, this additional information has no effect on the control performance. The same occurs for the FF-controller after about 9 min; the FB-controller has then managed to obtain the setpoint. However, after this point the effect of utilizing the occupancy aggravates the control. The reason is that the FF-part is independent on the actual setpoint and increases the supply air flow rate when the measurement is returned even though the system has stabilized at 1000 PPM. This result in an under-shoot of the setpoint which is reflected in the results as the additional dynamic control error compared to the FB-controller.

10.3.3 Infiltration flow rate

The sensitivity analysis also showed that the performance of the FF-controller can still be high even though the infiltration rate is not included among the disturbances. The static control error of the FF-controller is close to linearly dependent on the magnitude of the infiltration rate. At the highest value considered in the analysis, the control error corresponded to -135 PPM which means that instead of ending up at 1000 PPM the system will reside around 865 PPM during steady-state. The corresponding value for the FB-controller which was used for IAQ control in this study is the setpoint of 710 PPM which was required in order to avoid breaching 1000 PPM. This means that even though the infiltration rate is as large as the maximum value considered in this study, the actual CO_2 concentration will probably reside closer to the 1000 PPM limit if FF is used instead of FB.

10.3.4 Combined effects

The cumulative effect of the parameters discussed in the text above was not evaluated. The infiltration rate was dismissed since the corresponding sensitivity was small compared to the delay-time and modelling error. These two are on the other hand not evaluated together since it is unlikely that they would coexist when the current person counter is used. If a long delay-time is used, the estimation of the number of persons can be accurate, which means that the modelling error becomes small. Or, a short delay-time is used which means that the model error might be large.

11 Overall conclusions and discussion

In this work, a number of novel methods for local and central HVAC-control have been designed and evaluated. Most of these methods were based on FF and their overall purpose was to yield high comfort, low energy usage and low peak power.

In summary, a couple of conditions that are affecting the potential of FF-control have been identified. Among those, the thermal mass of the building and ambient climate were included. When the most feasible combination of these conditions occur, the control strategies developed in this work yield a total energy saving of 26 % compared to when conventional FB control systems, designed in the best possible way, are used. Included in this value is a reduction of electrical drive energy for fans of 54 % and a reduction of heat supply of 25 %. At the same time the required peak power of the central fan and AHU is reduced by 42 and 54 % respectively.

Whether or not the indoor climate is improved by FF-control can be argued. What can be concluded from this work is that with FF, the control of both temperature and IAQ becomes more stable compared to conventional FB-controllers. When FF is used, the temperature and IAQ can be kept closer to their setpoints and the deviations due to disturbances are reduced.

11.1 Summary of conclusions

In chapter 5 it was found that simple control algorithms can be used to achieve a control with high performance. The largest complexity is instead allocated to the FF-sensors and especially to personal counters in rooms designed for more than one person.

In chapter 6, it was shown that local FF-controllers should primary be used for IAQ control. The reason is that common FB-controller performed well when it came to temperature control. This means that the improvement of utilizing FF-controllers became low. In this chapter it was also shown that by using FF for local IAQ control, the total energy usage could be reduced between about 20 and 5 % dependent on the thermal mass of the building and ambient climate. In this value a reduction of electrical energy for fans of between about 60 and 50 % is included. At the same time, the peak power of the AHU is decreased by between about 30 and 35 % and the FCU of between about 16 and 0 % dependent on the ambient climate.

In chapter 7, different central night-mode control strategies were evaluate. The overall results pointed to small gains by distinguishing between night and day from a control perspective.

In chapter 8, different methods for central control of SAT were evaluated. It was found that the total number of persons inside the building is a better indicator for the SAT than the OAT which conventionally is used. The results pointed to a reduced total energy usage of about 4 %.

In chapter 9, the local FF-control methods from chapter 5 were used to control the HVAC-units indicated in chapter 6. This local control system was combined with the central methods from chapter 7 and 8. The results showed that during winter, this combined control system resulted in a saving of total energy between 26 and 19 %, dependent on building density, and compared to conventional control methods. During summer, the corresponding saving in total energy usage was between 11 and 5 %. At the same time, the required peak power of the AHU was reduced by 10 and 54 % dependent on the ambient climate and the peak power of the central fan was reduced by 54 %. The peak power of the FCU is not reduced during winter but with about 10 % during summer.

11.2 Discussion

In the following section, some important aspects on how this work was performed and their consequences on the results are discussed. All of these questions have been raised and answered before but they are worth mentioning again.

Simplifications
Two main simplifications have been done in this work;

1. Only dry cooling has been considered
2. Infiltration was not included

The first simplification above indicates that only the sensitive part is taken into account when air is cooled. This primary affects the energy use of the AHU since the cooling of outdoor air in many cases can result in condensation of water vapour. It is thought that this simplification affects the potential of FF negatively. The reason is that one of the main effects of using FF for local control is that a larger part of the cooling supply is allocated to the FCU. Hence, the energy savings due to the latent part in the AHU is not accounted for.

The second simplification above was introduced due to the complexity of assessing the infiltration flow rate which means that it is unrealistic to consider this disturbance as measurable. Instead, it was left out in the models. On the other hand, the effect of the infiltration on the potential of FF control was evaluated in chapter 10 and was there shown to be relative small.

System aspects
Five different aspects regarding how the systems in this work are designed needs some special attention;

1. The lower boundary of the ventilations flow rate was zero
2. CO_2 as an indicator of IAQ
3. FF was evaluated by comparing to best possible FB
4. The reference HVAC system was very flexible

The lower boundary of the ventilation flow rate was set to zero. In real systems, on the other hand, a non-zero boundary might be necessary to compensate for unmeasured emissions. However, in this work the CO_2 emission was treated as the only indicator for IAQ which motivates the choice of boundary. The relevance of this approach can be questioned even though this is a common procedure in real applications. A better approach would be to measure the complete set of emissions and to let the most critical one determine the flow rate. This is discussed by R.Kadribegovic[35] but is out of the scope of this work.

The effect of the zero flow rate boundary is thought to be positive for the FF-control systems. These systems have the possibility to quickly control the flow rate and the larger control range that is managed by the FF the larger is the gain.

The third aspect above refers to that the FB-controllers that were used in this study were designed with the best possible performance. It is thought that this approach is affecting the potential of FF controllers negatively. If FB-controllers with less performance instead would be used, the resulting control would be less stable. Hence, the breaching of the setpoint would be increased which means that the setpoints of the FB-controllers would be reduced even more to fulfil the criterions for comparability. In that case, also the comparability criterion for thermal climate might be relevant.

The fourth aspect above is refereeing to that the reference HVAC-system was designed not to limit the performance of the controllers regarding the match between supply and demand. Hence, the reference system was design with the largest flexibility for control as possible. This is motivated by that the HVAC-system first and foremost must be well designed. It is not a good idea to solve problems on a HVAC system level on a control system level. It is thought that this aspect is influencing the potential of FF-control positively. The FF-controllers are designed to supply what’s demanded and if this is not promoted by the HVAC-systems the potential will drop.

Evaluation design

1. Criterions for comparability

The aspect above is referring to the criterions that two control systems must fulfil in this work in order for to be considered as comparable. Regarding temperature, the criterion was equal degree hours outside the dead-band and regarding IAQ it was equal CO_2 peaks of 1000 PPM. These were introduced to incorporate the aspect of indoor climate in the evaluation and are necessary for the FF-controllers to be favourable. The reason is that the main feature of FF-controllers is disturbance rejection which means that the more important a stable control is, the more favourable is FF-control. Especially in the IAQ case, when the controller should avoid a threshold, the gain of FF-control is large. The reason is that the setpoint of the FB-controllers must be reduced to avoid breaching the limit which results in a larger energy usage.

It was shown that the criterion for temperature was not corresponding to what is considered as an acceptable thermal climate in office building. The resulting thermal climate when FB-controller was used was acceptable and for that reason the criterion was dropped. Without this criterion, FF was not favourable for temperature control. On the other hand, in applications where only small deviations from a temperature setpoint are acceptable FF might be a very good alternative to FB-controllers.

12 Future work

The results in this work derive almost entirely from simulations and the outcomes are the most favourable FF designs for local and central control. The next natural step is to test the selected control strategies in real applications. A laboratory environment is prepared for this task and it is primary the selected FF-control strategies from chapter 5 that will be evaluated.

Another important continuation of this research is to evaluate a set of measurement from real office buildings. These measurements derive from supply air diffusers with integrated sensors including;

- Motion sensors
- Duct temperature sensors
- Room air sensors
- Room CO_2 sensors

This data can be used to determine the potential of FF-control as well as normal operation regions in which the efficiency of the HVAC-components can be maximized. Also, load curves can be developed which can be used as input in simulations. Finally, the potential of recirculation by utilizing fresh conditioned air in empty offices can be evaluated.

Other areas of interest are;

- Optimization of air-to-water heat exchanger operation by using parallel connected units with air bypass
- Control strategies from parallel connected units with air bypass
- FF-control of recirculation system
- FF-control of all-air systems
- Control strategies for distributed fan and pump systems

Further, there is some work left in the studies presented in chapter 8 and 9 before any general conclusion can be drawn. For example;

- Replace the intermediate period of two hours in chapter 8 to a period with the length necessary to reach the day-mode setpoint just before the day begins
- Improve the strategies in chapter 9. Especially optimize the relation between number of persons and the SAT. Also, extend the optimal mix strategy.

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Appendix A

Below the main body of results from step 1 of the study in chapter 5 are presented. First the results from the meeting room setups are presented in A.1. These are followed by the results from the office room setups in A.2.

A.1

Table A.1 Result of step 1 in FF-control design selection process. Test setup: meeting room, heavy building structure. (1/2)

FF-sensor	FF-filter	CO_2 overshoot grading [-]	Temperature overshoot grading [-]	Electrical energy grading [-]	Cooling/ heating energy grading [-]	Total energy grading [-]	FCU power grading [-]	AHU power grading [-]	Electrical power grading [-]	**Sum [-]**
Motion sensor for IAQ control	Block	0.5	0.2	1.0	1.0	1.0	0.4	0.0	1.0	**5.2**
	Block and zero	1.0	1.0	0.0	0.7	0.0	1.0	0.0	0.0	3.7
	Init supply rate	0.7	0.8	0.0	0.0	0.0	0.6	0.0	0.0	2.1
Motion sensor for IAQ and thermal control	Block	0.5	1.0	1.0	1.0	1.0	0.4	0.0	1.0	**5.9**
	Block and zero	1.0	1.0	0.0	0.8	0.0	1.0	0.0	0.0	3.8
	Init supply rate	0.7	0.7	0.0	0.0	0.0	0.7	0.0	0.0	2.1

Table A.2 Result of step 1 in FF-control design selection process. Test setup: meeting room, heavy building structure. (2/2)

FF-sensor	FF-filter	CO_2 overshoot grading [-]	Temperature overshoot grading [-]	Electrical energy grading [-]	Cooling/ heating energy grading [-]	Total energy grading [-]	FCU power grading [-]	AHU power grading [-]	Electrical power grading [-]	**Sum [-]**
Electrical power sensor for thermal control	Direct	0.5	0.2	0.0	0.0	0.0	0.0	0.0	0.0	**0.7**
	Direct dynamic	0.5	0.2	0.0	0.0	0.0	0.0	0.0	0.0	0.7
	Static	0.5	1.0	0.0	0.0	0.0	0.3	0.0	0.0	1.8
	Dynamic	0.5	0.7	0.5	0.3	0.3	0.5	0.0	0.0	**2.8**
Person counter for IAQ control	Non-dynamic	1.0	1.0	0.5	1.0	1.0	1.0	1.0	1.0	**7.5**
	Dynamic	0.9	0.8	1.0	0.9	1.0	0.9	1.0	1.0	7.3

Table A.3 Result of step 1 in FF-control design selection process. Test setup: meeting room, light building structure.

FF-sensor	FF-filter	CO_2 overshoot grading [-]	Temperature overshoot grading [-]	Electrical energy grading [-]	Cooling/ heating energy grading [-]	Total energy grading [-]	FCU power grading [-]	AHU power grading [-]	Electrical power grading [-]	**Sum [-]**
Motion sensor for IAQ and thermal control	Block	0.5	1.0	1.0	1.0	1.0	0.5	0.0	1.0	**6.1**
	Block and zero	1.0	1.0	0.0	0.9	0.6	1.0	0.0	0.0	4.6
	Init supply rate	0.7	0.8	0.0	0.6	0.4	0.7	0.0	0.0	3.3
Electrical power sensor for thermal control	Direct	0.5	0.3	0.0	0.0	0.0	0.0	0.0	0.0	**0.8**
	Direct dynamic	0.5	0.3	0.0	0.0	0.0	0.0	0.0	0.0	0.8
	Static	0.5	0.9	0.0	0.1	0.1	0.6	0.0	0.0	2.2
	Dynamic	0.5	1.0	0.3	0.0	0.1	0.4	0.0	0.0	**2.3**

Table A.5 Result of step 1 in FF-control design selection process. Test setup: office room, heavy building structure, summer outside conditions.

FF-sensor	FF-filter	CO_2 overshoot grading [-]	Temperature overshoot grading [-]	Electrical energy grading [-]	Cooling/ heating energy grading [-]	Total energy grading [-]	FCU power grading [-]	AHU power grading [-]	Electrical power grading [-]	**Sum [-]**
Solar heating gain sensor for thermal control	Direct	0.0	1.0	0.6	1.0	1.0	0.2	0.0	0.2	**4.1**
	Direct dynamic	0.0	0.6	0.0	0.0	0.0	0.3	0.0	0.2	1.1
	Static	0.0	0.1	1.0	0.0	0.0	1.0	0.0	1.0	**3.1**
	Dynamic	0.0	0.1	0.6	0.0	0.0	1.0	0.0	0.9	2.6
Outside temperature sensor for thermal control	Direct	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
	Direct dynamic	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
	Static	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
	Dynamic	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Door opening sensor for IAQ control	Non-dynamic	1.0	0.7	1.0	1.0	1.0	1.0	0.9	1.0	**7.6**
	Dynamic	0.0	0.0	1.0	0.7	0.8	0.7	0.9	0.6	4.8
Person counter for IAQ control	Non-dynamic	1.0	1.0	0.0	0.0	0.0	1.0	0.0	1.0	**4.0**
	Dynamic	0.0	0.3	0.0	0.0	0.0	1.0	0.0	1.0	2.3

Table A.6 Result of step 1 in FF-control design selection process. Test setup: office room, light building structure, summer outside conditions.

FF-sensor	FF-filter	CO_2 overshoot grading [-]	Temperature overshoot grading [-]	Electrical energy grading [-]	Cooling/ heating energy grading [-]	Total energy grading [-]	FCU power grading [-]	AHU power grading [-]	Electrical power grading [-]	**Sum [-]**
Solar heating gain sensor for thermal control	Direct	0.0	0.8	1.0	1.0	1.0	0.0	0.0	0.0	**3.9**
	Direct dynamic	0.0	1.0	1.8	0.0	0.0	0.9	0.0	1.1	4.8
	Static	0.0	0.5	0.9	0.0	0.0	0.0	0.0	0.0	1.5
	Dynamic	0.0	0.0	1.8	0.0	0.0	1.0	0.0	1.0	**3.8**
Outside temperature sensor for thermal control	Direct	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
	Direct dynamic	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
	Static	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
	Dynamic	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0

Table A.7 Result of step 1 in FF-control design selection process. Test setup: office room, heavy building structure, winter outside conditions. (1/2)

FF-sensor	FF-filter	CO_2 overshoot grading [-]	Temperature overshoot grading [-]	Electrical energy grading [-]	Cooling/ heating energy grading [-]	Total energy grading [-]	FCU power grading [-]	AHU power grading [-]	Electrical power grading [-]	**Sum [-]**
Solar heating gain sensor for thermal control	Direct	0.0	0.0	0.0	0.0	0.0	1.0	0.0	0.7	**1.7**
	Direct dynamic	0.0	0.0	0.0	0.0	0.0	0.1	0.0	1.0	1.1
	Static	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
	Dynamic	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	**0.0**
Outside temperature sensor for thermal control	Direct	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
	Direct dynamic	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
	Static	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
	Dynamic	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0

Table A.8 Result of step 1 in FF-control design selection process. Test setup: office room, heavy building structure, winter outside conditions. (2/2)

FF-sensor	FF-filter	CO_2 overshoot grading [-]	Temperature overshoot grading [-]	Electrical energy grading [-]	Cooling/ heating energy grading [-]	Total energy grading [-]	FCU power grading [-]	AHU power grading [-]	Electrical power grading [-]	**Sum [-]**
Electrical power sensor for thermal control	Direct	0.0	1.0	1.0	1.0	1.0	1.0	0.0	1.0	**6.0**
	Direct dynamic	0.0	0.0	0.0	0.2	0.0	0.6	0.0	0.0	0.8
	Static	0.0	0.0	0.0	0.6	0.0	0.0	0.0	0.0	0.6
	Dynamic	0.0	0.0	0.0	0.7	0.0	0.0	0.0	0.0	**0.7**

Table A.9 Result of step 1 in FF-control design selection process. Test setup: office room, light building structure, winter outside conditions. (1/2)

FF-sensor	FF-filter	CO_2 overshoot grading [-]	Temperature overshoot grading [-]	Electrical energy grading [-]	Cooling/ heating energy grading [-]	Total energy grading [-]	FCU power grading [-]	AHU power grading [-]	Electrical power grading [-]	**Sum [-]**
Solar heating gain sensor for thermal control	Direct	0.0	0.0	0.0	0.0	0.0	0.1	0.0	0.7	**0.8**
	Direct dynamic	0.0	0.0	0.0	0.0	0.0	1.0	0.0	1.0	2.0
	Static	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
	Dynamic	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	**0.0**
Outside temperature sensor for thermal control	Direct	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
	Direct dynamic	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
	Static	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
	Dynamic	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0

Table A.10 Result of step 1 in FF-control design selection process. Test setup: office room, light building structure, winter outside conditions. (2/2)

FF-sensor	FF-filter	CO_2 overshoot grading [-]	Temperature overshoot grading [-]	Electrical energy grading [-]	Cooling/ heating energy grading [-]	Total energy grading [-]	FCU power grading [-]	AHU power grading [-]	Electrical power grading [-]	**Sum [-]**
Electrical power sensor for thermal control	Direct	0.0	1.0	0.0	0.0	0.0	0.1	0.0	1.0	**2.1**
	Direct dynamic	0.0	0.8	0.0	0.0	0.0	0.1	0.0	0.9	1.7
	Static	0.0	0.0	0.0	0.9	0.0	0.9	0.0	0.0	1.8
	Dynamic	0.0	0.0	0.0	1.0	0.0	1.0	0.0	0.0	**2.0**

Appendix B

Below the main body of results from step 2 of the study in chapter 5 are presented. In B.1, the results from the meeting room setups are presented. These are followed by the results from the office room setups in B.2.

B.1

Table B.1 Results of step 2 in FF-control design selection process for parameter-based methods. Potential of FF regarding energy usage compared to a corresponding FB control system. Test setup: meeting room, heavy building structure.

	Cooling energy savings [%]	Electrical energy savings [%]	Total energy savings [%]
Motion sensor (IAQ block)	14	28	17
Motion sensor (IAQ block and thermal direct)	12	27	16
Electrical power sensor (Thermal direct)	1	1	1
Person counter (IAQ non-dynamic and thermal direct)	13	70	26
Motion sensor and electrical power sensor (IAQ non-dynamic and thermal direct)	14	28	17

Table B.2 Results of step 2 in FF-control design selection process for parameter-based methods. Potential of FF regarding peak power reduction compared to a corresponding FB control system. Test setup: meeting room, heavy building structure

	FCU power savings [%]	AHU power savings [%]	Electrical power savings [%]
Motion sensor (IAQ block)	18	11	24
Motion sensor (IAQ block and thermal direct)	-8	11	24
Electrical power sensor (Thermal direct)	34	0	0
Person counter (IAQ non-dynamic and thermal direct)	38	44	73
Motion sensor and electrical power sensor (IAQ non-dynamic and thermal direct)	40	11	24

Table B.3 Results of step 2 in FF-control design selection process for model-based methods. Potential of FF regarding energy usage compared to a corresponding FB control system. Test setup: meeting room, heavy building structure

	Cooling energy savings [%]	Electrical energy savings [%]	Total energy savings [%]
Electrical power sensor (Thermal dynamic)	6	0	5
Motion and electrical power sensor (IAQ block and thermal dynamic)	20	28	22

Table B.4 Results of step 2 in FF-control design selection process for model-based methods. Potential of FF regarding peak power reduction compared to a corresponding FB control system. Test setup: meeting room, heavy building structure

	FCU power savings [%]	AHU power savings [%]	Electrical power savings [%]
Electrical power sensor (Thermal dynamic)	27	0	0
Motion and electrical power sensor (IAQ block and thermal dynamic)	36	11	24

Table B.5 Results of step 2 in FF-control design selection process for parameter-based methods. Potential of FF regarding energy usage compared to a corresponding FB control system. Test setup: meeting room, light building structure.

	Cooling energy savings [%]	Electrical energy savings [%]	Total energy savings [%]
Motion sensor (IAQ block)	3	27	8
Motion sensor (IAQ block and thermal direct)	1	24	6
Electrical power sensor (Thermal direct)	0	1	0
Person counter (IAQ non-dynamic and thermal direct)	0	67	14
Motion sensor and electrical power sensor (IAQ non-dynamic and thermal direct)	7	27	11

Table B.6 Results of step 2 in FF-control design selection process for parameter-based methods. Potential of FF regarding peak power reduction compared to a corresponding FB control system. Test setup: meeting room, light building structure

	FCU power savings [%]	AHU power savings [%]	Electrical power savings [%]
Motion sensor (IAQ block)	11	11	24
Motion sensor (IAQ block and thermal direct)	-47	11	24
Electrical power sensor (Thermal direct)	22	0	0
Person counter (IAQ non-dynamic and thermal direct)	12	44	73
Motion sensor and electrical power sensor (IAQ non-dynamic and thermal direct)	17	11	24

Table B.7 Results of step 2 in FF-control design selection process for model-based methods. Potential of FF regarding energy usage compared to a corresponding FB control system. Test setup: meeting room, light building structure

	Cooling energy savings [%]	Electrical energy savings [%]	Total energy savings [%]
Electrical power sensor (Thermal dynamic)	0	0	0
Motion and electrical power sensor (IAQ block and thermal dynamic)	7	27	11

Table B.8 Results of step 2 in FF-control design selection process for model-based methods. Potential of FF regarding peak power reduction compared to a corresponding FB control system. Test setup: meeting room, light building structure

	FCU power savings [%]	AHU power savings [%]	Electrical power savings [%]
Electrical power sensor (Thermal dynamic)	11	0	0
Motion and electrical power sensor (IAQ block and thermal dynamic)	31	11	24

B.2

Table B.9 Results of step 2 in FF-control design selection process for parameter-based methods. Potential of FF regarding energy usage compared to a corresponding FB control system. Test setup: office room, heavy building structure, summer outside conditions

	Cooling energy savings [%]	Electrical energy savings [%]	Total energy savings [%]
Solar heat gain sensor (Thermal direct)	0	0	0
Door opening sensor (IAQ non-dynamic and thermal direct)	1	3	1
Door opening sensor (IAQ vent off)	1	3	1
Person counter (IAQ non-dynamic)	1	5	2
Person counter and door opening sensor (IAQ non-dynamic)	3	17	5
Person counter and electrical power sensor (IAQ non-dynamic and thermal direct)	1	5	2

Table B.10 Results of step 2 in FF-control design selection process for parameter-based methods. Potential of FF regarding peak power reduction compared to a corresponding FB control system. Test setup: office room, heavy building structure, summer outside conditions

	FCU power savings [%]	AHU power savings [%]	Electrical power savings [%]
Solar heat gain sensor (Thermal direct)	0	0	0
Door opening sensor (IAQ non-dynamic and thermal direct)	13	0	0
Door opening sensor (IAQ vent off)	13	0	0
Person counter (IAQ non-dynamic)	16	15	35
Person counter and door opening sensor (IAQ non-dynamic)	14	16	35
Person counter and electrical power sensor (IAQ non-dynamic and thermal direct)	13	15	35

Table B.11 Results of step 2 in FF-control design selection process for model-based methods. Potential of FF regarding energy usage compared to a corresponding FB control system. Test setup: office room, heavy building structure, summer outside conditions

	Cooling energy savings [%]	Electrical energy savings [%]	Total energy savings [%]
Solar heat gain sensor (Thermal static)	1	0	1
Door opening sensor (IAQ non-dynamic and thermal static)	1	3	1
Person counter and electrical power sensor (IAQ non-dynamic and thermal dynamic)	3	5	3

Table B.12 Results of step 2 in FF-control design selection process for model-based methods. Potential of FF regarding peak power reduction compared to a corresponding FB control system. Test setup: office room, heavy building structure, summer outside conditions

	FCU power savings [%]	AHU power savings [%]	Electrical power savings [%]
Solar heat gain sensor (Thermal static)	1	0	0
Door opening sensor (IAQ non-dynamic and thermal static)	13	0	0
Person counter and electrical power sensor (IAQ non-dynamic and thermal dynamic)	16	15	35

Table B.13 Results of step 2 in FF-control design selection process for parameter-based methods. Potential of FF regarding energy usage compared to a corresponding FB control system. Test setup: office room, heavy building structure, winter outside conditions

	Heating energy savings [%]	Electrical energy savings [%]	Total energy savings [%]
Solar heat gain sensor (Thermal direct)	0	0	0
Door opening sensor (IAQ non-dynamic and thermal direct)	2	3	2
Electrical power sensor (Thermal direct)	2	0	1
Person counter (IAQ non-dynamic and thermal direct)	-1	15	-1
Person counter, electrical power sensor and door opening sensor (IAQ non-dynamic and thermal direct)	4	18	9

Table B.14 Results of step 2 in FF-control design selection process for parameter-based methods. Potential of FF regarding peak power reduction compared to a corresponding FB control system. Test setup: office room, heavy building structure, winter outside conditions

	FCU power savings [%]	AHU power savings [%]	Electrical power savings [%]
Solar heat gain sensor (Thermal direct)	-1	0	0
Door opening sensor (IAQ non-dynamic and thermal direct)	0	0	0
Electrical power sensor (Thermal direct)	0	0	0
Person counter (IAQ non-dynamic and thermal direct)	0	19	47
Person counter, electrical power sensor and door opening sensor (IAQ non-dynamic and thermal direct)	0	19	35

Table B.15 Results of step 2 in FF-control design selection process for model-based methods. Potential of FF regarding energy usage compared to a corresponding FB control system. Test setup: office room, heavy building structure, winter outside conditions

	Heating energy savings [%]	Electrical energy savings [%]	Total energy savings [%]
Solar heat gain sensor (Thermal dynamic)	0	0	0
Door opening sensor (IAQ non-dynamic and thermal dynamic)	2	3	2
Electrical power sensor (Thermal dynamic)	-1	0	0

Table B.16 Results of step 2 in FF-control design selection process for model-based methods. Potential of FF regarding peak power reduction compared to a corresponding FB control system. Test setup: office room, heavy building structure, winter outside conditions

	FCU power savings [%]	AHU power savings [%]	Electrical power savings [%]
Solar heat gain sensor (Thermal dynamic)	0	0	0
Door opening sensor (IAQ non-dynamic and thermal dynamic)	0	0	0
Electrical power sensor (Thermal dynamic)	0	0	0

Table B.17 Results of step 2 in FF-control design selection process for parameter-based methods. Potential of FF regarding energy usage compared to a corresponding FB control system. Test setup: office room, light building structure, summer outside conditions

	Cooling energy savings [%]	Electrical energy savings [%]	Total energy savings [%]
Solar heat gain sensor (Thermal direct)	0	0	0
Door opening sensor (IAQ non-dynamic and thermal direct)	0	2	1
Door opening sensor (IAQ vent off)	0	2	1
Person counter (IAQ non-dynamic)	1	5	1
Person counter and door opening sensor (IAQ non-dynamic)	2	16	3
Person counter and electrical power sensor (IAQ non-dynamic and thermal direct)	1	5	1

Table B.18 Results of step 2 in FF-control design selection process for parameter-based methods. Potential of FF regarding peak power reduction compared to a corresponding FB control system. Test setup: office room, light building structure, summer outside conditions

	FCU power savings [%]	AHU power savings [%]	Electrical power savings [%]
Solar heat gain sensor (Thermal direct)	0	0	0
Door opening sensor (IAQ non-dynamic and thermal direct)	9	0	0
Door opening sensor (IAQ vent off)	9	0	0
Person counter (IAQ non-dynamic)	12	15	35
Person counter and door opening sensor (IAQ non-dynamic)	9	16	35
Person counter and electrical power sensor (IAQ non-dynamic and thermal direct)	12	15	35

Table B.19 Results of step 2 in FF-control design selection process for model-based methods. Potential of FF regarding energy usage compared to a corresponding FB control system. Test setup: office room, light building structure, summer outside conditions

	Cooling energy savings [%]	Electrical energy savings [%]	Total energy savings [%]
Solar heat gain sensor (Thermal static)	1	0	1
Door opening sensor (IAQ non-dynamic and thermal static)	0	2	1
Person counter and electrical power sensor (IAQ non-dynamic and thermal dynamic)	2	6	2

Table B.20 Results of step 2 in FF-control design selection process for model-based methods. Potential of FF regarding peak power reduction compared to a corresponding FB control system. Test setup: office room, light building structure, summer outside conditions

	FCU power savings [%]	AHU power savings [%]	Electrical power savings [%]
Solar heat gain sensor (Thermal static)	0	0	0
Door opening sensor (IAQ non-dynamic and thermal static)	9	0	0
Person counter and electrical power sensor (IAQ non-dynamic and thermal dynamic)	13	15	35

Table B.21 Results of step 2 in FF-control design selection process for parameter-based methods. Potential of FF regarding energy usage compared to a corresponding FB control system. Test setup: office room, light building structure, winter outside conditions

	Heating energy savings [%]	Electrical energy savings [%]	Total energy savings [%]
Solar heat gain sensor (Thermal direct)	0	0	0
Door opening sensor (IAQ non-dynamic and thermal direct)	1	3	2
Electrical power sensor (Thermal direct)	2	0	1
Person counter (IAQ non-dynamic and thermal direct)	0	15	0
Person counter, electrical power sensor and door opening sensor (IAQ non-dynamic and thermal direct)	3	17	8

Table B.22 Results of step 2 in FF-control design selection process for parameter-based methods. Potential of FF regarding peak power reduction compared to a corresponding FB control system. Test setup: office room, light building structure, winter outside conditions

	FCU power savings [%]	AHU power savings [%]	Electrical power savings [%]
Solar heat gain sensor (Thermal direct)	0	0	0
Door opening sensor (IAQ non-dynamic and thermal direct)	1	0	0
Electrical power sensor (Thermal direct)	0	0	0
Person counter (IAQ non-dynamic and thermal direct)	-1	19	47
Person counter, electrical power sensor and door opening sensor (IAQ non-dynamic and thermal direct)	0	19	35

Table B.23 Results of step 2 in FF-control design selection process for model-based methods. Potential of FF regarding energy usage compared to a corresponding FB control system. Test setup: office room, light building structure, winter outside conditions

	Heating energy savings [%]	Electrical energy savings [%]	Total energy savings [%]
Solar heat gain sensor (Thermal dynamic)	0	0	0
Door opening sensor (IAQ non-dynamic and thermal dynamic)	1	3	2
Electrical power sensor (Thermal dynamic)	1	0	1

Table B.24 Results of step 2 in FF-control design selection process for model-based methods. Potential of FF regarding peak power reduction compared to a corresponding FB control system. Test setup: office room, light building structure, winter outside conditions

	FCU power savings [%]	AHU power savings [%]	Electrical power savings [%]
Solar heat gain sensor (Thermal dynamic)	-1	0	0
Door opening sensor (IAQ non-dynamic and thermal dynamic)	0	0	0
Electrical power sensor (Thermal dynamic)	0	0	0

Appendix C

Below the main body of results from the study in chapter 6 are presented. First the results for the setups consisting of heavy building structures under summer and winter ambient conditions are presented respectively. These are followed by the corresponding results related to light structures.

Table C.1 Results of multi-zone setup. Energy usage related to FF-control, compared to a corresponding FB control system. Test setup: Multi-zone system, heavy building structure, summer ambient conditions

	Heating energy savings [%]	Cooling energy savings [%]	Electrical energy savings [%]	Total energy savings [%]
Model-based. FF-control of heating, cooling and air supply	12	-3	58	9
Model-based. FF-control of heating and air supply	15	-1	58	11
Parameter-based. FF-control of heating, cooling and air supply	12	-3	58	9
Parameter-based. FF-control of heating and air supply	14	-1	58	11

Table C.2 Results of multi-zone setup. Required peak power related to FF-control, compared to a corresponding FB control system. Test setup: Multi-zone system, heavy building structure, winter ambient conditions

	FCU power savings [%]	AHU power savings [%]	Electrical power savings [%]
Model-based. FF-control of heating, cooling and air supply	14	28	54
Model-based. FF-control of heating and air supply	16	28	54
Parameter-based. FF-control of heating, cooling and air supply	13	28	54
Parameter-based. FF-control of heating and air supply	16	28	54

Table C.3 Results of multi-zone setup. Energy usage related to FF-control, compared to a corresponding FB control system. Test setup: Multi-zone system, heavy building structure, winter ambient conditions

	Heating energy savings [%]	Cooling energy savings [%]	Electrical energy savings [%]	Total energy savings [%]
Model-based. FF-control of heating, cooling and air supply	18	-327	53	18
Model-based. FF-control of heating and air supply	19	-229	53	19
Parameter-based. FF-control of heating, cooling and air supply	18	-404	53	17
Parameter-based. FF-control of heating and air supply	19	-234	53	19

Table C.4 Results of multi-zone setup. Required peak power related to FF-control, compared to a corresponding FB control system. Test setup: Multi-zone system, heavy building structure, winter ambient conditions

	FCU power savings [%]	AHU power savings [%]	Electrical power savings [%]
Model-based. FF-control of heating, cooling and air supply	-1	35	54
Model-based. FF-control of heating and air supply	-1	35	54
Parameter-based. FF-control of heating, cooling and air supply	0	35	54
Parameter-based. FF-control of heating and air supply	0	35	54

Table C.5 Results of multi-zone setup. Energy usage related to FF-control, compared to a corresponding FB control system. Test setup: Multi-zone system, light building structure, summer ambient conditions

	Heating energy savings [%]	Cooling energy savings [%]	Electrical energy savings [%]	Total energy savings [%]
Model-based. FF-control of heating, cooling and air supply	12	-3	56	6
Model-based. FF-control of heating and air supply	11	-3	56	6
Parameter-based. FF-control of heating, cooling and air supply	10	-3	56	5
Parameter-based. FF-control of heating and air supply	10	-3	56	5

Table C.6 Results of multi-zone setup. Required peak power related to FF-control, compared to a corresponding FB control system. Test setup: Multi-zone system, light building structure, summer ambient conditions

	FCU power savings [%]	AHU power savings [%]	Electrical power savings [%]
Model-based. FF-control of heating, cooling and air supply	15	28	54
Model-based. FF-control of heating and air supply	10	28	54
Parameter-based. FF-control of heating, cooling and air supply	10	28	54
Parameter-based. FF-control of heating and air supply	10	28	54

Table C.7 Results of multi-zone setup. Energy usage related to FF-control, compared to a corresponding FB control system. Test setup: Multi-zone system, light building structure, winter ambient conditions

	Heating energy savings [%]	Cooling energy savings [%]	Electrical energy savings [%]	Total energy savings [%]
Model-based. FF-control of heating, cooling and air supply	17	-85	52	13
Model-based. FF-control of heating and air supply	18	-65	53	16
Parameter-based. FF-control of heating, cooling and air supply	17	-82	52	14
Parameter-based. FF-control of heating and air supply	18	-72	52	15

Table C.8 Results of multi-zone setup. Required peak power related to FF-control, compared to a corresponding FB control system. Test setup: Multi-zone system, light building structure, winter ambient conditions

	FCU power savings [%]	AHU power savings [%]	Electrical power savings [%]
Model-based. FF-control of heating, cooling and air supply	1	34	54
Model-based. FF-control of heating and air supply	2	35	54
Parameter-based. FF-control of heating, cooling and air supply	0	35	54
Parameter-based. FF-control of heating and air supply	0	35	54

Appendix D

Below the main body of results from the study in chapter 7 are presented. First the results for the setups consisting of heavy structures under summer presented respectively. These are followed by the corresponding results related to light structures.

Table D.1 Difference in energy usage by using night-mode compared to using the same mode during night and day. Test setup: Office-room, heavy building structure, summer outside conditions

	Heating energy savings [%]	Cooling energy savings [%]	Electrical energy savings [%]	Total energy savings [%]
IAQ focusing	-79	2	0	1
Thermal focusing	-163	13	-84	-5
Constant flow rate	-373	11	-27	1
Shut-off	-66	0	0	-1

Table D.2 Difference in required peak power by using night-mode compared to using the same mode during night and day. Test setup: Office-room, heavy building structure, summer outside conditions

	FCU power savings [%]	AHU power savings [%]	Electrical power savings [%]
IAQ focusing	1	0	0
Thermal focusing	2	0	0
Constant flow rate	2	0	0
Shut-off	0	0	0

Table D.3 Difference in energy usage by using night-mode compared to using the same mode during night and day. Test setup: Office-room, light building structure, summer outside conditions

	Heating energy savings [%]	Cooling energy savings [%]	Electrical energy savings [%]	Total energy savings [%]
IAQ focusing	-18	0	0	-2
Thermal focusing	-38	0	-52	-11
Constant flow rate	-80	1	-28	-12
Shut-off	-8	0	0	-1

Table D.4 Difference in required peak power by using night-mode compared to using the same mode during night and day. Test setup: Office-room, light building structure, summer outside conditions

	FCU power savings [%]	AHU power savings [%]	Electrical power savings [%]
IAQ focusing	0	1	0
Thermal focusing	0	0	0
Constant flow rate	0	0	0
Shut-off	0	0	0

Appendix E

Below the main body of results from the study in chapter 8 are presented. First the results for the setups consisting of heavy structures under summer and winter ambient conditions are presented. These are followed by the corresponding results related to light structures.

Table E.1 Results of SAT control study. Relative energy usage related to alternative SAT control instead of conventional OAT based control. Test setup: Multi-zone, heavy building structure, winter and summer ambient conditions

	Heating energy savings [%]	Cooling energy savings [%]	Electrical energy savings [%]	Total energy savings [%]
Dynamic thermal FF-model. Summer ambient conditions	3	6	0	5
Dynamic thermal FF-model. Winter ambient conditions	-4	-69	0	-5
Optimal mix. Summer ambient conditions	3	6	0	5
Optimal mix. Winter ambient conditions	-3	-69	0	-5

Table E.2 Results of SAT control study. Relative required peak power related to alternative SAT control instead of conventional OAT based control. Test setup: Multi-zone, heavy building structure, winter and summer ambient conditions

	FCU power savings [%]	AHU power savings [%]	Electrical power savings [%]
Dynamic thermal FF-model. Summer ambient conditions	-6	36	0
Dynamic thermal FF-model. Winter ambient conditions	0	-2	0
Optimal mix. Summer ambient conditions	-6	36	0
Optimal mix. Winter ambient conditions	0	-1	0

Table E.3 Results of SAT control study. Relative energy usage related to alternative SAT control instead of conventional OAT based control. Test setup: Multi-zone, light building structure, winter and summer ambient conditions

	Heating energy savings [%]	Cooling energy savings [%]	Electrical energy savings [%]	Total energy savings [%]
Dynamic thermal FF-model. Summer ambient conditions	1	0	0	0
Dynamic thermal FF-model. Winter ambient conditions	-4	-17	0	-6
Optimal mix. Summer ambient conditions	1	0	0	0
Optimal mix. Winter ambient conditions	-4	-16	0	-6

Table E.4 Results of SAT control study. Relative required peak power related to alternative SAT control instead of conventional OAT based control. Test setup: Multi-zone, light building structure, winter and summer ambient conditions

	FCU power savings [%]	AHU power savings [%]	Electrical power savings [%]
Dynamic thermal FF-model. Summer ambient conditions	0	36	0
Dynamic thermal FF-model. Winter ambient conditions	0	-2	0
Optimal mix. Summer ambient conditions	0	36	0
Optimal mix. Winter ambient conditions	0	-2	0

Appendix F

Below the theory regarding MPC controllers presented in chapter 2.3 is extended.

Feasibility of the RHC

The main difference between a closed-loop and an open-loop control strategy is quite obvious. In a closed-loop, the controller computes the entire control signal sequence based on the initial state. This sequence is then kept during the entire control task and in each time-step the next control signal in the sequence is subjected to the process. In order for this strategy to perform, a perfect model of the system within the current operational mode is required. This also requires a process without any disturbances or changes of other uncontrolled inputs during the control action. For the closed-loop case, the control signal in each time-step is based on the current status of the process. However, closed-loop strategies most often lacks of deeper insight of the current process which leads to a control strategy which is quite spontaneous.

The RHC is something in the middle since a closed-loop control sequence is determined in each time-step. Even though the prediction horizon is long, the RHC prediction is quite short-sighted compared to “real” infinite horizon setups which in some cases can lead to problems. The controller might lead the process along a trajectory which ends at infeasible results beyond the current prediction horizon. If that is the case, the process will end up in a region where it can’t be steered away from without violating the constraints. An example of this issue is shown in figure 6.2. The admissible initial values of the states x_1 and x_2 consist of both the grey and the white region. However, after evaluating the control strategies for different initial states it became evident that only the grey region leads to feasible results. Another related problem is that the controller might steer the system along a trajectory which on one hand is feasible but one the other doesn’t convergence to the origin.

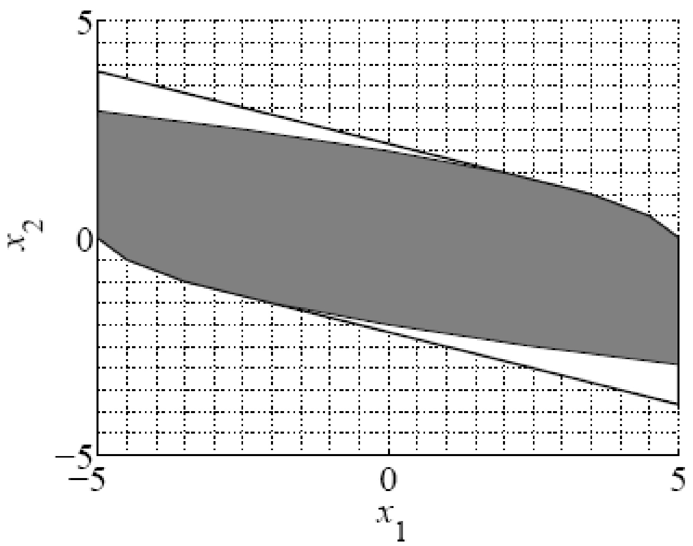


Figure F.1 Initial states (grey and white) and states which results in feasible solutions (grey) [28]

A desired property of the RHC is that feasibility is guaranteed for all future times. This property is called persistent feasibility and can be achieved if the controller is designed properly. The theory which is used to determine conditions for persistent feasibility of explicit RHC is based on sets. In the area of control theory different sets of states are defined. A specific set possess some specific features and it contains all states which has these features. In this content, the set called Control Invariant Set is of interest. This set is the result of the assignment "Find the set of initial states for which there exists a controller such that the system constraints are never violated". This means that if the state values of a certain process is contained in the corresponding invariant set the process can evolve freely in a feasible manner. Such sets can be defined for a certain process given the constraints on input, output and states.

By treating the different states values as set the control assignment can be more structured. The controller actions can be divided into two steps. First the controller should steer the system from any initial state into the control invariant set by solving a constrained multiparametric program. In the next step, an unconstrained LQR with infinite horizon since inside the control invariant set the system constraints will never be violated. Hence, by defining *N* as the minimum number of steps which is needed for the controller to drive the system from any admissible initial state to the control invariant set, the controller becomes persistent feasible. The minimum number of steps required for persistent feasibility is usually referred to the determinedness index and is usually solved numerically off-line.

Figure 6.3 shows an example of how a control invariant set can look like. In this example, it is denoted as C_∞ and is a closed set of two variables, namely x_1 and x_2. It is surrounded by other regions which constitutes of different combinations of initial values of state x_1 and x_2. These regions are characterized by the number of step that a certain controller needs in order to steer the system from one of the regions into the critical invariant set. Starting from the control invariant set, each time a boundary is breached the controller needs yet another step to go back to the control invariant set. If figures 6.3 represent the total space of admissible initial states, the determinedness index is four since the controller needs four steps to drive the system from the most outer set of initial states the control invariant set.

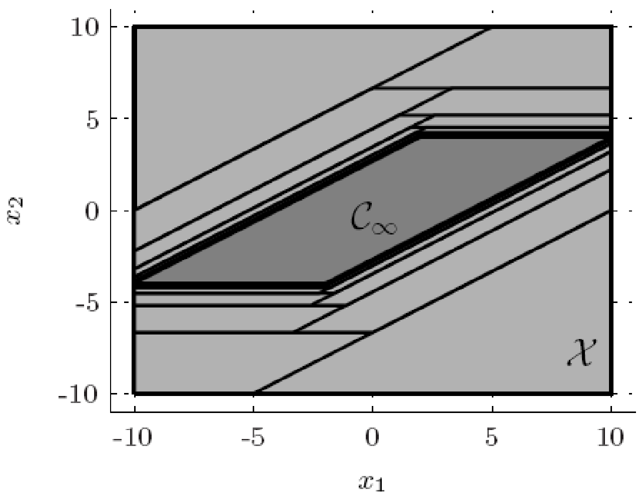


Figure F.2 Control invariant set denoted by C_∞ and initial states denoted X. [13]

Persistent feasibility does not necessarily solve the other issue which previously was addressed. Namely that the controller might steer the system along a trajectory which on one hand is feasible but one the other doesn't convergence to the origin. This problem will however be solved in order to guarantee stability of the controller. The constraint might also prohibit the controller to steer the system to the origin. This problem must however be solved by relaxing the constraints.

Stability of the RHC

Yet we haven't discussed the stability of any MPC controller. The concept of stability can be evaluated in many different ways and the one used in this project is the so called Lyapunov stability. This concept is based on that if a stable system is disturbed just a little bit, it should stay in a small neighbourhood of its origin. According to Lyapunov stability, the system is unstable if it is not stable. The usual way to show Lyapunov stability is to construct a so called Lyapunov function for the system in question. This function has some specific properties and if it is found the system is stable. This means that starting from any initial state, the system will eventually settle to the origin. However, if a Lyapunov function isn't found, this doesn't mean that the system is unstable. It just means that we can't say anything about its stability.

Lyapunov stability can also be compared to other criterions of stability. For example, if the system is stable according to Lyapunov stability, its eigenvalues are inside the unit disc. This is a common stability criterion for discrete systems.

If the RHC controller is designed according to some criterions, its cost function becomes a Lyapunov function. Since such a function then has been defined for the system, it is stable according to Lyapunov. These criterions are given by Borelli[13]. The one which must be taken into account in the design process states that

1. $Q = Q' > 0,\ R = R' > 0,\ P > 0$
2. The states, the final states and the admissible control signals contain the corresponding origin.
3. The final states are a control invariant set

The first criterion states that two of the weights must be symmetrical matrices and all of them must be definite positive. The second state that the constraints on the states, final states and control signals must be designed so that an equilibrium point is admissible. The third one requires the controller to be persistent feasible which according to previous discussion is retrieved by setting the prediction horizon to the determinedness index. If the Lyapunov stability is retrieved the state of the system will converge to the origin. Hence, the second issue stated above is overcome.

Appendix G

Below the final form of the equations used to model building and HVAC-components are presented. For each model, the input, output and model parameters are presented. The parameters have either been treated as varying from one model to another or as constant throughout the study. First, the constant reappearing model parameters are presented.

Reappearing constant model parameters: $c_{p,a}$ (specific heat capacity of air, 1000 J/(kg·K)), $c_{p,i}$ (specific heat capacity of air, 1200 J/(kg·K), ρ_a (density of air), ρ_w (density of water), $c_{p,w}$ (specific heat capacity of water)

Room model: thermal and CO2 concentration

Input variables: $\dot{V}_s$ (supply air flow rate), t_s (temperature of supply air), c_s (CO2 concentration of the supply air), $\dot{Q}_{FCU,heat}$ (heat supply rate), $\dot{Q}_{FCU,cooling}$ (heat extraction rate)

Output variables: t_r (room air temperature), c_r (room air CO2 concentration), $t_{surf,ie}$ (surface temperature of interior building elements, i.e. walls, floor and roof), $t_{surf,ow}$ (surface temperature of exterior wall)

Disturbances: t_{adj} (air temperature of adjacent rooms), t_o (outdoor air temperature), $\dot{V}_{door}$ (air flow rate through door), $\dot{V}_{\inf}$ (infiltration flow rate), $\dot{Q}_{light}$ (heat emitted by lighting), n_{people} (number of people), $\dot{Q}_{equip}$ (heat emitted by equipment), $\dot{Q}_{sun}$ (solar heat gain)

Variable model parameters: V_r (room volume), m_i (mass of interior), U_{win} (overall heat transfer coefficient of window), A (area of building element), $\dot{q}_{people}$ (specific heat emitted by people), $c_{p,ie}$ (specific heat capacity of internal building element), ρ_{ie} (density of internal building element), ρ_{ow} (density of exterior wall), $c_{p,ow}$ (specific heat capacity of exterior wall), V_{ie} (thermally active volume of interior building element), V_{ow} (thermally active volume of exterior wall), U'_{ie} (convective and conductive heat transfer coefficient of interior building element), U'_{ow} (convective and conductive heat transfer coefficient of exterior wall)

Constant model parameters: α_i (room-side convective heat transfer coefficient, 6.9 W/(m^2·K)), α_{adj} (convective heat transfer coefficient of adjacent room 6.9 W/(m^2·K)), α_o (convective heat transfer coefficient of external wall 6.9 W/(m^2·K)), R_{conv} (part of heat transferred by convection, values presented in table 4.1), $c_{p,i}$ (specific heat capacity of interior, 1255 J/(kg·K)

$$\frac{dt_r}{d\tau}\cdot(V_r\cdot\rho_a\cdot c_{p,a}+m_i\cdot c_{p,i})=\dot{V}_s\cdot\rho_a\cdot c_{p,a}\cdot(t_s-t_r)$$

$$+\dot{V}_{inf}\cdot\rho_a\cdot c_{p,a}\cdot(t_0-t_r)+\dot{V}_{door}\cdot\rho_a\cdot c_{p,a}\cdot(t_{adj}-t_r)$$

$$+U_{win}\cdot A_{win}\cdot(t_0-t_r)+\alpha_i\cdot A_{ow}\cdot(t_{surf,ow}-t_r)$$

$$+\sum_{n=1}^{4}\alpha_{i,n}\cdot A_{surf,ie,n}\cdot(t_{surf,ie,n}-t_r)+R_{conv,sun}\cdot\dot{Q}_{sun}$$

$$+n_{people}\cdot R_{conv,people}\cdot\dot{q}_{people}+R_{conv,equip}\cdot\dot{Q}_{equip}$$

$$+R_{conv,light}\cdot\dot{Q}_{light}+R_{conv,FCU}\cdot\dot{Q}_{FCU,heat}+R_{cr,FCU}\cdot\dot{Q}_{FCU,cooling} \tag{eq. G.1}$$

$$\frac{dt_{surf,ie,1}}{d\tau}\cdot(V_{ie,1}\cdot\rho_{ie,1}\cdot c_{p,ie,1})=A_{ie,1}\cdot U'_{ie,1}\cdot(t_{surf,ie,1}-t_{adj,1})$$

$$+\alpha_i\cdot A_{surf,ie,,1}\cdot(t_r-t_{surf,ie,1})+\frac{1}{R_{conv,sun,ie,1}}\cdot\dot{Q}_{sun}$$

$$+n_{people}\cdot\frac{1}{R_{conv,people,ie,1}}\cdot\dot{q}_{people}+\frac{1}{R_{conv,equip,ie,1}}\cdot\dot{Q}_{equip}$$

$$+\frac{1}{R_{conv,light,ie,1}}\cdot\dot{Q}_{light}+\frac{1}{R_{conv,FCU,ie,1}}\cdot\dot{Q}_{FCU,heat}$$

$$+\frac{1}{R_{cr,FCU,ie,1}}\cdot\dot{Q}_{FCU,cooling}$$

$$\vdots$$

$$\frac{dt_{surf,ie,4}}{d\tau}\cdot(V_{ie,4}\cdot\rho_{ie,4}\cdot c_{p,ie,4})=A_{ie,4}\cdot U'_{ie,4}\cdot(t_{surf,ie,4}-t_{adj,4})$$

$$+\alpha_i\cdot A_{surf,ie,4}\cdot(t_r-t_{surf,ie,4})+\frac{1}{R_{conv,sun,ie,4}}\cdot\dot{Q}_{sun}$$

$$+n_{people}\cdot\frac{1}{R_{conv,people,ie,4}}\cdot\dot{q}_{people}+\frac{1}{R_{conv,equip,ie,4}}\cdot\dot{Q}_{equip}$$

$$+\frac{1}{R_{conv,light,ie,4}}\cdot\dot{Q}_{light}+\frac{1}{R_{conv,FCU,ie,4}}\cdot\dot{Q}_{FCU,heat}$$

$$+\frac{1}{R_{cr,FCU,ie,4}}\cdot\dot{Q}_{FCU,cooling}$$

$$\frac{dt_{surf,ow}}{d\tau}\cdot(V_{ow}\cdot\rho_{ow}\cdot c_{p,ow})=A_{ow}\cdot U'_{ow}\cdot(t_{surf,ow}-t_o)$$

$$+\alpha_i\cdot A_{surf,ow}\cdot(t_r-t_{surf,ow})+\frac{1}{R_{conv,sun,ow}}\cdot\dot{Q}_{sun}$$

$$+n_{people}\cdot\frac{1}{R_{conv,people,ow}}\cdot\dot{q}_{people}+\frac{1}{R_{conv,equip,ow}}\cdot\dot{Q}_{equip}$$

$$+\frac{1}{R_{conv,light,ow}}\cdot\dot{Q}_{light}+\frac{1}{R_{conv,FCU,ow}}\cdot\dot{Q}_{FCU,heat}$$

$$+\frac{1}{R_{cr,FCU,ow}}\cdot\dot{Q}_{FCU,cooling}$$

$$\frac{dc_r}{d\tau}\cdot(V_r)=\dot{V}_s\cdot(c_s-c_r)+\dot{V}_{inf}\cdot(c_o-c_r)+\dot{V}_{door}\cdot(c_{adj}-c_r)$$

$$+n_{people}\cdot\dot{m}_{CO_2} \tag{eq. G.2}$$

where

$$U'_{ie} = \frac{1}{\frac{1}{\alpha_{adj}} + \sum \frac{d_{ie}}{\lambda_{ie}}} \qquad U'_{ow} = \frac{1}{\frac{1}{\alpha_o} + \sum \frac{d_{ow}}{\lambda_{ow}}}$$

Flow rate through door

Input variables: h_{door} (door opening), t_{adj} (adjacent room temperature), t_r (room temperature)

Output variables: $\dot{V}_{door}$ (flow rate through door)

Model parameters: A_{door}(area of door), H_{door}(height of door)

$$\dot{V}_{door} = 0.03 \cdot h_{door} \cdot A_{door} \cdot \left(H_{door} \cdot \left|t_{adj} - t_r\right|\right)^{0.5} \qquad \text{(eq. G.3)}$$

General pressure drop; grate, filter, heat recovery exchanger, balancing damper

Input variables: $\dot{V}$ (flow rate)

Output variables: Δp (pressure drop)

Model parameters: Δp_{design}(design pressure drop), $\dot{V}_{design}$(design flow rate), n_{turb} (turbulence exponent)

$$\Delta p = \Delta p_{design} \cdot \left(\frac{\dot{V}}{\dot{V}_{design}}\right)^{n_{turb}} \qquad \text{(eq. G.4)}$$

Water pipe and air duct with transport delay

Input variables: $\dot{V}_i$ (inlet flow rate), v_i (inlet flow velocity), t_i (inlet temperature), c_i (inlet concentration)

Output variables: $\dot{V}_o$ (outlet flow rate), t_o (outlet temperature), c_o (outlet concentration), Δp (pressure drop)

Internal variables: f (friction factor)

Model parameters: ρ (density of air/water), L (duct/pipe length), d_h (hydraulic diameter of duct/pipe)

$$\begin{aligned} c_i &= c_o \cdot e^{\tau_d \cdot s} \\ \dot{V}_i &= \dot{V}_o \cdot e^{\tau_d \cdot s} \\ t_i &= t_o \cdot e^{\tau_d \cdot s} \end{aligned} \qquad \text{(eq. G.5)}$$

$$\tau_d = \frac{L}{v_i}$$

$$\Delta p = f \cdot \frac{\rho \cdot v}{2} \cdot \frac{L}{d_h} \qquad \text{(eq. G.6)}$$

$$f = funct(Re)$$

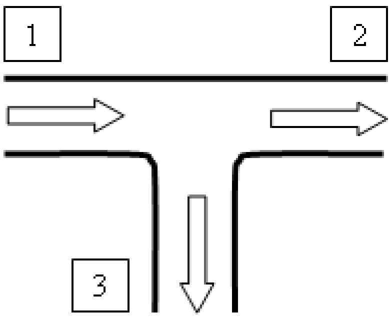
Duct branch (index according to the picture below)

Input variables: $\dot{V}_1$ (air flow rate through main duct before the branch), $p_{tot,1}$ (total pressure before in main duct before the branch), $\dot{V}_3$ (diverted air flow)

Output variables: $\dot{V}_2$ (air flow rate through main duct after the branch), $\Delta p_{1,2}$ (pressure drop between branch 1 and 2), $\Delta p_{1,3}$ (pressure drop between branch 1 and 3)

Internal variables: f_2 (pressure loss factor between branch 1 and 2), f_3 (pressure loss factor between branch 1 and 3)

Model parameters: A (area of inlet/outlet)



$$p_{tot,1} = p_{tot,2} + \Delta p_{1,2} = p_{tot,2} + f_2 \cdot \left(\frac{\dot{V}_2}{A_2}\right)^2 \cdot \frac{\rho_a}{2}$$

$$p_{tot,1} = p_{tot,3} + \Delta p_{1,3} = p_{tot,3} + f_3 \cdot \left(\frac{\dot{V}_3}{A_3}\right)^2 \cdot \frac{\rho_a}{2} \qquad \text{(eq. G.7)}$$

$$\dot{V}_1 = \dot{V}_2 + \dot{V}_3$$

$$f_2 = funct\left(\frac{\dot{V}_2}{\dot{V}_1}\right)$$

$$f_3 = funct\left(\frac{\dot{V}_3}{\dot{V}_1}\right)$$

Duct and pipe bend

Input variables: $\dot{V}$ (volume flow rate)

Output variables: Δp (pressure drop)

Model parameters: ρ (density of media), d_h (hydraulic diameter of bend), φ (angle of bend), r (radius of bend)

$$\Delta p = \frac{2 \cdot \rho \cdot \dot{V}^2 \cdot \varphi}{45 \cdot \pi^2 \cdot d_h{}^4} \cdot \left(\pi \cdot \frac{r}{d_h} \cdot f_f + 2 \cdot f_{other} \right) \qquad \text{(eq. G.8)}$$

$$f_2 = funct\left(\frac{\dot{V}_2}{\dot{V}_1} \right)$$

$$f_{other} = funct\left(\frac{r}{d_h} \right)$$

Diffuser and control valve with exponential characteristics
Input variables: $\dot{V}$ (flow rate), z (opening of valve/diffuser)

Output variables: Δp (pressure drop)

Model parameters: Δp_{design}(design pressure drop), $\dot{V}_{design}$(design flow rate), $\left(\frac{\dot{V}_{max}}{\dot{V}_{min}} \right)$ (ratio of maximum and minimum flow rate)

$$\Delta p = \Delta p_{design} \cdot \left(\frac{\dot{V}}{\left(\frac{\dot{V}_{max}}{\dot{V}_{min}} \right)^{z-1} \cdot \dot{V}_{design}} \right)^2 \qquad \text{(eq. G.9)}$$

Pumps and fans
Input variables: $\dot{V}$ (flow rate), Δp (pressure rise)

Output variables: $\dot{W}$ (electrical energy usage)

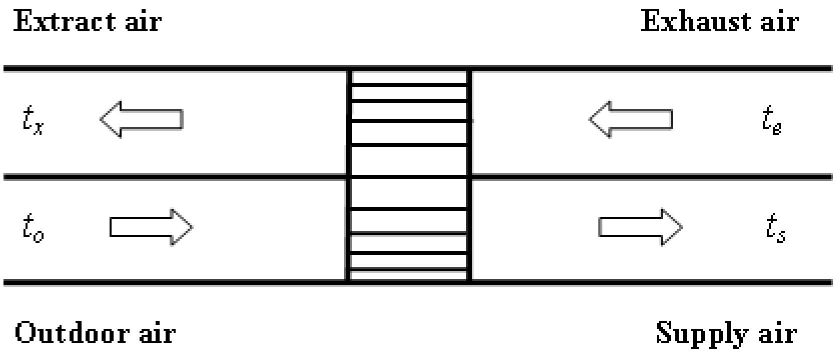
Model parameters: η (total electrical efficiency)

$$\dot{W} = \frac{\dot{V} \cdot \Delta p}{\eta} \qquad \text{(eq. G.10)}$$

Heat recovery exchanger (index according to the picture below)
Input variables: $\dot{V}_s$ (supply air flow rate), n (rotational speed of heat recovery wheel), t_0 (outdoor air temperature), t_e (exhaust air temperature)

Output variables: t_s (supply air temperature), η_t (temperature efficiency)

Model parameters: m_{HRX} (mass of heat recovery wheel), $c_{p,HRX}$ (specific heat capacity of heat recovery wheel), $\eta_{t,max}$ (maximum temperature efficiency), $\eta_{t,1}$ (temperature efficiency at the arbitrary operational point 1), n_1 (rotational speed of heat recovery wheel at the arbitrary operational point 1), $\dot{V}_{s,1}$ (supply air flow rate at the arbitrary operational point 1)



$$\frac{dt_s}{d\tau}\cdot\left(\frac{m_{HRX}\cdot c_{p,HRX}}{2\cdot\dot{V}_s\cdot\rho_a\cdot c_{p,a}}\right)+t_s=(1-\eta_t)\cdot t_o+\eta_t\cdot t_e \qquad \text{(eq. G.11)}$$

$$\eta_t=\left(\eta_{t,max}\right)^{\frac{\dot{V}_S}{\dot{V}_{S,1}}}-\left(\eta_{t,max}\right)^{\frac{\dot{V}_S}{\dot{V}_{S,1}}}\cdot\left(1-\left(\frac{\eta_{t,1}}{\eta_{t,max}}\right)^{\frac{\dot{V}_S}{\dot{V}_{S,1}}}\right)^{\frac{n}{n_1}}$$

Air heater and dry cooler

Input variables: $\dot{V}_a$ (flow rate on air-side), $\dot{V}_w$ (flow rate on water-side), v_a (velocity on air-side), t_{ai} (air-side inlet temperature), t_{wi} (water-side inlet temperature)

Output variables: t_{ao} (air-side outlet temperature), t_{wo} (water-side outlet temperature), Δp_a (air-side pressure drop), Δp_w (water-side pressure drop), U_a (overall heat transfer coefficient on air-side)

Internal variables: f_a (air-side friction factor)

Model parameters: A_a (heat transfer area on air-side), $U_{a,design}$ (design overall heat transfer coefficient on air-side), $\dot{V}_{a,design}$ (design flow rate on air-side), L_a (flow length on air-side), L_w (flow length on water-side), $d_{h,a}$ (hydraulic diameter on air-side), $d_{h,w}$ (hydraulic diameter on water-side)

$$\dot{V}_w\cdot\rho_w\cdot c_{p,w}\cdot(t_{wi}-t_{wo})+U_a\cdot A_a\cdot(t_{ai}-t_{wo})=0$$

(eq. G.12)

$$t_{ao} = \frac{U_a \cdot A_a}{\dot{V}_a \cdot \rho_a \cdot c_{p,a}} \cdot t_{wo} + \left(1 - \frac{U_a \cdot A_a}{\dot{V}_a \cdot \rho_a \cdot c_{p,a}}\right) \cdot t_{ai}$$

$$U_a \cdot A_a = U_{a,design} \cdot A_a \cdot \left(\frac{\dot{V}_a}{\dot{V}_{a,design}}\right)^{0.65}$$ (eq. G.14)

$$t_{ao} = \frac{e^{-10s}}{65s+1} \cdot \dot{V}_a + \frac{e^{-20s}}{55s+1} \cdot \dot{V}_w + \frac{e^{-20s}}{4s+1} \cdot t_{ai} + \frac{e^{-20s}}{50s+1} \cdot t_{wi}$$ (eq. G.13)

$$\Delta p_a = f_a \cdot \frac{\rho_a \cdot v_a}{2} \cdot \frac{L_a}{d_{h,a}}$$ (eq. G.15)

$$f_a = funct(Re_a)$$

$$\Delta p_w = f_w \cdot \frac{\rho_w \cdot v_w}{2} \cdot \frac{L_w}{d_w}$$

$$f_w = funct(Re_w)$$ (eq. G.16)

Sensors
Input variables: temperature, CO2 concentration, flow rate etc.

Output variables: temperature, CO2 concentration, flow rate etc.

Model parameters: τ_d (transport-delay), $\tau_{0.65}$ (time-constant)

$$y = \frac{e^{\tau_d \cdot s}}{\tau_{0.65} s + 1} \cdot u$$ (eq. G.17)