Doctoral Thesis

Advanced building ventilation system based on the paradigms of decentralization and exergy minimization using a highly interlaced network structure for the supply and a sensor-based local control approach for the exhaust

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ABHANDLUNG
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Nomenclature

AC       Air Conditioning
ADN      Air Distribution Network
ADS      Air Distribution System
AFUE     Annual Fuel Utilization Efficiency
AHU      Air Handling Unit
BSG      Building Systems Group
CAV      Constant Air Volume
DCV      Demand Controlled Ventilation
DHW      Domestic Hot Water
GHG      Greenhouse Gas
HVAC     Heating Ventilation Air Conditioning
IAQ      Indoor Air Quality
LEV      Local Exhaust Ventilation
OOM      Object Oriented Modeling
OP       Operation Point
PV       Photovoltaic
RPM      Revolutions Per Minute
UFAD  Underfloor Air Distribution
VAV  Variable Air Volume
VOC  Volatile Organic Compounds
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Abstract

There is a large potential in the building sector for reducing greenhouse gas (GHG) emissions using improved concepts along with state-of-the-art technology. In contrast to the mobility sector where high grade energy sources are absolutely required for locomotion, in the building sector the amount of high grade energy actually used could be greatly reduced. A quantity that defines the quality of a given energy form is the exergy. It is a combination of the energy conservation described by the first law of thermodynamics and the principle of entropy generation formulated in the second law of thermodynamics. In every energy conversion process there are losses involved that lead to entropy production and hence exergy destruction, i.e. degradation of the grade of energy. In order to minimize conversion losses in a system, the quality of the energy source has to be matched to the quality required by the sink. Unfortunately the awareness of the concept of exergy is not yet very high in the building industries and hence only scarcely regarded in the design of buildings. For this reason also, buildings are predominantly heated with high grade fossil fuels although the heat required to maintain its temperature during cold outside temperatures is of very low quality.

This thesis is aimed at contributing to the ongoing research in the building sector with the goal of designing buildings that can be operated as Zero Emission Buildings. Instead of relying only on the improvement of existing solutions, new solutions are researched to get closer to this goal.

For an improvement of the thermodynamic processes in buildings to reduce GHG emissions, a paradigm shift is needed in the way buildings are operated. One paradigm that is treated in this work is the one of exergy minimization or the so called lowEx paradigm (Chapter 2). The primary goal is to free buildings from GHG emissions during operation but the concept of exergy is the mean to achieve the optimization of the processes such the realization of the ”zero carb” building becomes economically feasible. Another paradigm being crucial for this work is the one of decentralization (Chapter 2). Focused
on these paradigms, this work is concerned about the analysis and validation of new concepts in the field of HVAC to achieve the goals of zero GHG emissions.

A first step towards more sustainable buildings is to choose an integrated system approach. This involves the integration of technical components into an over-all system concept as well as the integration of the technical system into the planning process. When both integrations are realized a more sustainable building can be obtained (Chapter 4).

With the focus on ventilation a new system is proposed that uses a local and demand controlled approach to effectively remove contaminants from the building together with a decentralized supply concept.

The exhaust system (Chapter 5) comprises a large amount of individually controlled suction points that are distributed over the space but connected with each other through ducting. The negative pressure in the ducting is maintained by a centralized exhaust fan and the distribution of the suction power among the different points is dependent on their status, whether they are activated or not. The activation of a single point depends on the local contaminant concentration being measured. The ventilation pattern that evolves in such a system is not predictable and too complex to be handled by a central control. Because of the autonomous function of the individual points only a very simple control for each point is necessary. An advanced operation of the exhaust is achieved with this control approach because it always automatically adapts to the current needs in the building. The major benefits from this approach are the increased efficiency as well as the improved effectiveness. The demand controlled nature of the system with a precise detection through distributed sensing leads to large reductions in the ventilation rate according to the occupancy patterns. Using local extraction of contaminants with high concentrations ensures lower contaminant levels in the occupied zone with lower ventilation rates when compared to classical systems. When assuming a typical occupancy level in office buildings of 76% a perfectly demand controlled system leads to ventilation rate reductions of 24%. The effect of local detection and extraction of the contaminants allows further reduction of ventilation rates by 9% as was confirmed by measurements. Lower ventilation rates greatly reduce the demand for conditioning outside air to room conditions. It also reduces the fan power significantly, i.e. by a factor greater than 3, taking the ventilation rate reduction estimates from above.

The supply system (Chapter 6) ensures continuity in that it replaces the exhausted air with conditioned outside air. The exhaust system with its sensing intelligence plays the role of a master and determines ventilation activity while the supply system acts as a slave. The application of the local exhaust principle leads to a spatially varying demand of supply air. In order to avoid
pressure differences to build up between neighboring zones, the air supply has to work with very low pressure drops. Ideally it requires an infinite volume from where supply air can be withdrawn without inducing any flow in the volume, i.e. a “lake of fresh air”. This lake situation was approximated by using a decentralized supply approach with a large number of air handling units (AHU) feeding a plenum from the periphery of the building. The higher the number of sources, given a total amount of air to be supplied, the smaller is the air flow velocity per source and the more likely it is to obtain a homogeneous static pressure distribution. Instead of using a raised floor construction as a pressure plenum, a ducting network is used to connect the various AHUs to each other. In such a multi-source arrangement, when connecting the sources with each other through ducting, a networked, closed-loop supply structure automatically arises. The supply infrastructure regarded in this work can be fully integrated into the primary building structure. This is due to the fact that many devices are used to solve the task of air supply which leads to small units and a slim, distribution network. In typical, centralized single-source supply systems, a branching topology has to be used. This means that an integration into the building structure is impossible because of the size of the plant and the ducting. As a consequence, suspended ceilings or raised floors have to be used, and consequently the thermal mass of the building is decoupled from the room. With the decentralized system capable of being integrated in the building structure, the thermal mass of the building can be used as a thermal store to reduce the peak power required for heating or cooling. This allows as a consequence systems with moderate water temperatures to be used which greatly reduces exergy destruction on the emission side and lead to a significant improvement of heat pump performance.

The goal of achieving suitable characteristics of the supply system with a ducting network was pursued by looking at different network topologies. It was shown that closed-loop networks have a favorable behavior in terms of air distribution similar to purely parallel arrangements. A 1D mathematical model was set up and implemented to assess the distribution properties of closed-loop structures. Variable boundary conditions, as they appear because of the influence of wind on buildings, were covered in the model. In one study a classical decentralized supply system with an unconnected supply structure was compared to a chosen closed-loop network arrangement. It was shown that under influence of wind a much more uniform distribution of air flow rates in space can be achieved with a networked supply structure. Consequently, for a networked supply system, a lower over-all sensitivity to wind induced disturbances is obtained. The networking of the ducting allows for a flexible rerouting of air flows in the system such that a shortage of air
flow in one zone can be compensated by a neighboring zone. For the system design considered it could be shown that the routing can be influenced with very small changes to the pressure boundaries in the room. With a negative pressure of only 1 Pa applied to openings located in a critical zone in terms of air supply, the flow rate could be restored up to a very high percentage of the undisturbed case. Introducing simple junctions in addition to the openings being used to connect the ducts in the network, can further improve the networks distribution properties. Especially for the supply of the building core, far away from the facade, an improvement can be achieved. Still, the increased complexity of the network introducing junctions, does though not always pay off. Using junctions the pressure characteristics in the network can become more rigid such that the rerouting of air flows also becomes more difficult and less effective. It was found for the chosen setup that a simple rectangular grid with only opening-type nodes being used, can perform very well in terms of air distribution. A general sensitivity of the network performance to size and complexity of the network was identified and hence makes it difficult to come up with a universal solution for the ideal topology.

Thanks to the decentralized supply system, with its redistribution capabilities of the network, a hybrid operation can be realized. An over-capacity of fans installed at the periphery of the building allows for a flexible allocation of fan power such that together, with the wind forces acting on the building, an optimization of the plant performance can be achieved. With a 40% overcapacity and a wind speed of 10 m/s, a 25% reduction of total fan power i.e. exergy demand was reached. When comparing the decentralized supply to a typical centralized supply system in the studied case, the fan power was reduced by a factor of 4. Including the benefits from running the system in hybrid mode even a reduction by a factor 5 is possible. Besides a wind-based hybrid mode, a temperature-based mode is also possible. Again with a 40% fan over-capacity, assuming the outside air in the temperature boundary layer along a single facade of a building to be 12 K above freestream air temperature, led to a 6% and a 40% reduction of exergy load for heating and cooling respectively. This reduction in exergy load for conditioning of the outside air leads to commensurate savings of electricity for the heat pump.

The combined ventilation system with its local, sensor-controlled exhaust and its decentralized supply was studied as well under the control aspect. Different control strategies where theoretically tested for selected scenarios. The variable exhaust acts as a disturbance to the system similar to the wind influence which also has to be balanced out by the supply. The pure flow rate control of both the exhaust and the supply system was found to be the most robust approach. A constant pressure control of the supply system on an
AHU level was found to be particularly interesting also in combination with
the operation under influence of wind. When used together with a networked
supply structure showing a flat pressure loss characteristics, an non-uniform
supply through the large number of AHUs can be effectively handled. AHUs
delivering air from the pressure side can only deliver an increased amount
of air if the back pressure of the distribution system remains small. With
the constant pressure control of the AHUs, the supply system automatically
optimizes fan power by maximizing the air supply from the pressure side of
the building while minimizing the air supply from the negative side. For an
optimization in the thermal hybrid mode, the constant pressure control is not
suitable but a constant flow rate control is required.

The analysis of a new ventilation system integrated together with a heat
pump system revealed a large potential for reducing the use of high grade
energy, i.e. exergy. On the exhaust side as well as on the supply side reductions
factors of 3 to 4 or higher are possible. Similar or even larger potentials exist
when looking at the thermal systems including the heating and cooling of
buildings. When these available potentials are actually used the lowEx, zero
carbon building can be built and operated with minimal economic costs.
Zusammenfassung


Für eine Verbesserung der thermodynamischen Prozesse und die Ver-

Ein erster Schritt in Richtung nachhaltige Gebäude wird durch einen integrierten Systemansatz erreicht. Dieser Schritt beinhaltet die Integration der technischen Komponenten in ein übergeordnetes Systemkonzept sowie die Integration des technischen Systems in den gesamten Planungsprozess. Wenn beide Integrationformen realisiert werden, sind die Vorbedingungen für nachhaltigere Gebäude gegeben (Chapter 4).

Mit dem Fokus auf der Lüftung wird ein neuer Ansatz für ein lokales, bedarfsgesteuertes Abluftsystem vorgeschlagen, welches ein effektives Wegführen der Schadstoffe aus dem Raum ermöglicht. Das Abluftsystem wird ergänzt durch ein dezentrales Zuluftkonzept mit einer Vielzahl von vernetzten Komponenten.

entsprechend dem Belegungsmuster im Vergleich zu klassischen Systemen zu erheblichen Verringerungen der Luftwechselzahlen.


Die in dieser Arbeit betrachtete Infrastruktur für die Zuluft kann vollständig in die Primärstruktur des Gebäudes integriert werden. Dies ist aus dem Grund, dass eine Vielzahl von Geräten die Zuluftaufbereitung und -Verteilung übernehmen und sich dadurch kleine Einzelgeräte und ein schlankes Verteilungssystem ergibt. Im Vergleich dazu sind klassische zentrale, Raumluftsysteme sehr groß und verteilen die Luft von einem Punkt ausgehend über Rohrsysteme mit Baumstruktur im Gebäude. Sowohl die einzelne An-
lage aber auch das resultierende Rohrsystem ist dadurch zu gross, um in
der Gebaendestructur integriert werden zu koennen. In der Folge muessen
abgehaengte Decken- oder aufgestaenderte Bodenkonstruktionen verwendet
werden, um die Installation zu verstecken, so dass die thermische Gebaude-
masse vom Raum entkoppelt wird. Mit dem dezentralen System und dessen
Moeglichkeit der Integration kann die Gebaudemasse thermisch aktiviert und
als Speicher verwendet werden, wodurch sich die Lastspitzen fuer Heizen oder
Kuehlen reduzieren lassen. Es ergibt sich somit die Moeglichkeit der Verwen-
dung von Heiz- und Kuehlsystemen, welche mit Wassertemperaturen nahe
der Raumtemperatur betrieben werden koennen. Die Verwendung moderaten
Systemtemperaturen hat eine erhebliche Reduktion der Exergievernichtung
auf der Abgabeseite zur Folge und fuehrt zudem dazu, dass die Effizienz der
Waermepeumpe signifikant gesteigert wird.

Die Charakteristik des Zuluftsystems mit vermaschtem Rohrnetzwerk wurde
untersucht indem verschiedene Netzwerktopologien betrachtet wurden. Es
konnte gezeigt werden, dass vermaschte, closed loop-Netzwerke die gewuen-
schtes Eigenschaften im Hinblick auf die Lufverteilung, aehnlich einer rein par-
alleler Anordnung zeigen. Ein mathematisches 1D Modell wurde aufgestellt
und implementiert fuer die Berechnung der Lufverteilung in solchen closed-
loop Netzwerken. Variable Randbedingungen wie sie vom Windeinfluss aufs
Gebaude herruehren wurden im Modell beruecksichtigt. In einer Studie wurde
ein klassisches dezentrales Zuluftsystem ohne Vernetzung der Zuluftleitungen
mit einem ausgewahlten, vernetztem closed-loop System verglichen. Es hat
sich gezeigt, dass durch die Vernetzung das Zuluftsystem weniger sensitiv auf
Stoerungen durch den Wind reagiert und somit ein vernetztes System unter
Windeinfluss eine regelmaessigere Luftverteilung ermoeglicht als ein unvernet-
ztem System. In einem vernetzten Rohrsystem ist eine flexible Umleitung der
Stromungspfade moeglich, so dass eine Unterversorgung einer Zone durch die
benachbarte Zone kompensiert werden kann. Es konnte auch gezeigt werden
dass die beobachteten Pfadaenderungen bereits mit sehr geringen Druckun-
terschieden erreicht werden koennen. So konnte unter Windeinfluss mit dem
untersuchten, vermaschten Netzwerk mit 1 Pa Unterdruck in einem unter-
versorgten Raum die Luftmenge nahezu vollstaendig auf den urspruenglichen
Sollwert gebracht werden.

Das Einfuehren von geschlossenen Verbindungsknoten zusatzlich zu den
sonst verwendeten Zuluftoeffnungen als Knotenpunkte zwischen Rohren, kann
die Luftverteilungseigenschaft des Netzwerkes guenstig beeinflussen. Dies trifft
vor allem zu fuer die Versorgung der Kernzonen im Gebaude, welche weit weg
von der Fassade liegen. Trotz diesen Vorteilen der geschlossenen Knoten kann
die Erhoehung der Netzwerkkomplexitaet durch das Einfuehren solcher Knoten

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auch kontraproduktiv sein. Es wurde beobachtet, dass sich durch die Verwendung dieser Knoten die Druckcharakteristik im Netz dahingehend verändert, dass eine Pfadaenderung der Strömung unwahrscheinlicher und somit die gewünschte Umverteilung der Zuluft weniger effektiv wird. Das einfache Beispiel eines Netzwerkes mit orthogonaler Gitterstruktur und Zuluftöffnungen als einziger Knotentyp zeigte im Vergleich zu komplexeren Strukturen ein aehnlich gutes Verhalten. Es wurde aber generell eine relativ starke Abhängigkeit der Verteilungseigenschaften von Parametern wie Netzwerkgröße und komplexität erkannt, so dass es schwierig ist universelle Kriterien für das ideale Netzwerk aufzustellen.


Chapter 1

Introduction

1.1 Motivation

The man made climate change calls for action to reduce anthropogenic greenhouse gas (GHG) emissions. More than a third of the global primary energy usage and the respective amount of the global CO2 emissions are assigned to the building sector [7], [1], [5], [4]. In the years from 1973 and 2003 the energy demand in buildings has increased by 39% in IEA countries [4]. According to [3] worlds primary energy demand between 2005 and 2030 will further rise by 55% assuming a “business as usual” scenario. Fossil fuels will cover 84% of this rise. The biggest increase in demand is found for natural coal with 73%. Because of the high amount of fossil fuels used for energy generation a significant increase of GHG emissions comes along with this increase of the global energy demand. This energy related jump in GHG emissions is estimated at 57% [3]. The ”business as usual” scenario is no valid option for the future and hence drastic measures are required to stop any further increase of GHG emissions.

In buildings, about a quarter of the total GHG reductions would result from inexpensive measures such as better insulation of buildings, better HVAC technologies, etc. [2], [6]. These measures carry no net life cycle costs. When adopted at an early design phase they could even reduce construction costs because the need for heating, etc. decreases [4].

A large potential for reductions however is still unexploited. In a new construction, energy savings up to 75% could be achieved designing and operating buildings as complete systems and using advanced equipment. The
largest reductions over the entire building stock by 2030 are achieved through retrofitting existing buildings and replacing energy intensive equipment [6].

In the design of new buildings it is often neglected that similar to the way it was used in the past in old buildings, the thermal mass can be used for climate control. At least 50% of the inner surfaces of a building could be made available for thermal activation of the buildings mass and consequently lower the heating and the cooling power during usage. Because of the type of equipment used especially in large office buildings, the thermal mass in the building is separated from the room air by raised floor constructions or suspended ceilings. These cavities are needed to host the HVAC equipment such as huge ducts for the ventilation and hence disable the buildings mass to be activated. By splitting up large, centralized units into many small units the amount of ducting can be reduced and different ways of integration of technology into the building structure is possible. The decentralization of certain functionality and its delegation to a large number of small units that can be integrated in the building structure paves the way for better performing buildings with thermally activated building mass.

Well known equipments such as combustion heaters are optimized and have more or less reached their limits with a thermal efficiency of 95%. Obviously, from this technology no salvation is to be expected. New concepts are needed to make another step forward and dramatically reduce GHG emissions of buildings during operation. A paradigm shift in the way buildings are operated is necessary. Extending the common way the performance of buildings is evaluated is the needed step. Instead of solely relying on the energy perspective which allows to quantify the fluxes coming in and out of the buildings a measure to determine the quality of different forms of energy is needed. In real world, in any process losses occur; external but also internal due to entropy being produced. The entropy production is a law of nature and a driving force for all real processes. It is described by the 2nd law of thermodynamics. A combination of the easy to apply energy concept and the law of entropy is found in the concept of exergy. The exergy concept can lead to further optimization of building service systems. It is the next step into a new generation of building systems.

What do the buildings of the future look like, how are they going to be operated? From a physics perspective it is clear they are going to be lowEx buildings which follow the proposition of exergy minimization.

Typical heat generation in nowadays buildings rely on combustion heaters which are "exergy destruction machines" if used to supply heat at room temperature. Exergy is lost in the course of one transformation to another and is zero in the end when a system comes to equilibrium with the environment.
For this reason the quality of an energy source has to be matched to the quality required by the sink. By respecting these rules in the design of technical components and systems, lowEx buildings with a minimum input of high grade energy can be realized. Also, the exergy concept identifies processes which allow the utilization of low grade, environmental, i.e. renewable energy sources.

Even more ambitious than the lowEx building is the goal of the "zero-Carb" building which does not emit any CO2 during operation. The lowEx paradigm helps, though to reach the goal of zero CO2 emissions in that it excludes combustion processes in buildings completely and generally minimizes the necessary input of other high grade energy such as electricity. If the electricity supplied is fully generated from renewable sources and free of CO2 emissions the building becomes a "zeroCarb" building.

All the concepts and components analyzed in this work are strictly serving the purpose of the realization of the "zeroCarb" building. An integrative project "B35" of an "zeroCarb" apartment house in Zurich will be realized by Professor Leibundgut by end of 2010. This building will make use of the result of this work and from the research carried out by the Building Systems Group (BSG) at the ETH Zurich.

1.2 Aim of the Work

The general aim of this work is to contribute to the ongoing research in the field of new energy and exergy efficient building service systems (HVAC Systems). For this purpose new concepts were analyzed and validated. Beside the theoretical work it is aimed at developing and further improving new technical components that will help to realize the concepts under consideration. A decentralized air handling unit (AHU) as well as a multifunctional ceiling panel are part of the aimed development. The theoretical work is concerned with the in depth analysis of a new, sensor-controlled exhaust ventilation and a decentralized supply and distribution system.

For the sensor-controlled, local exhaust ventilation (LEV) concept based on the demand controlled ventilation principle it is the goal to validate its increased ventilation effectiveness and energy efficiency compared to classical systems using different models and measurements. Also, the working principles and behavior of this new exhaust system have to be explained in order to identify its potential for future energy and exergy savings.

Concerning the decentralized air supply it is the aim to find an appropriate layout of the air distribution system (ADS) to satisfy the requirements from
the local exhaust system with temporal and spatial variations of the ventilation rates. Other influences such as wind forces acting on the building have to be accounted for in the layout of this concept. Based on the analysis of the air distribution it is tried to identify the potential energy and exergy savings that are expected from such a decentralized approach.

Taking the elements discussed in this work together to comprise an integrated building system it is further the aim to show that this integrated system is suitable to be used in lowEx buildings and will allow for a reduction in the use of high grade energy of a factor 10 to 15 when compared to classical building systems.

1.3 About this Thesis

This thesis has been carried out in the BSG of the Swiss Federal Institute of Technology, ETH Zurich. This group is situated within the Institute of Building Technology being part of the Department of Architecture. The Building Systems Group with its head, Prof. Dr. Hansjuerg Leibundgut has devoted its research to the lowEx building design. This covers new concepts for the design and planning of buildings, for technical systems as well as the development of new components. The BSG is the official representative for Switzerland in the program ECBCS Annex49: Low Exergy Systems for High-Performance Buildings and Communities from the International Energy Agency (IEA). The main objective of Annex 49 is to develop concepts for reducing the exergy demand in the built environment, thus reducing CO2 emissions from the building stock. Specifically, exergy analysis shall be used to develop tools, guidelines, best practice examples and background materials to designers and decision makers in the building sector. It helps in disseminating the concept of exergy and through this, in achieving a better performance of future buildings with less or no emissions of greenhouse gases.

The work presented herein is concerned about the efficient utilization of energy in the operation of buildings. In this context new concepts of a sensor-controlled exhaust and a decentralized supply system have been analyzed. This analysis lead to several published scientific papers that are collected and presented in this thesis. Beside these publications more background information and extended research results to certain aspects are provided.
Bibliography


2.1 HVAC

The term HVAC stands for heating, ventilation and air conditioning. HVAC is responsible for climate control meaning the control of temperature, humidity and indoor air quality. While in some systems all three functions are combined they can also be separate, handled by independent systems. The term HVAC originates from the U.S. where systems with combined air heating and conditioning are predominantly used.

2.1.1 Short Historical Background on HVAC with a Focus on Ventilation

For a very long time open fires in housing were the predominant way for heating and cooking. In early times no effective means for the removal of smoke were available such that a serious health problem caused by the open fires was there. This problem in some places like western Europe in 1558 became even more severe when the usage of hardwood for heating purposes was forbidden [31]. The deforestation during the urbanization at that time lead to a scarcity of hardwood such that this resource had to be reserved for construction. Instead, soft wood of lower quality had to be burned for heating purposes, giving rise to even more smoke during combustion. Later, charcoal became popular because of its smokelessness. But large carbon monoxide concentrations in closed rooms arising from this fuel caused many people to die and subsequently triggered significant improvements in fire place design.
at the end of 18th century. First fireplace designs that allowed heating of a room without contamination of smoke, followed. The problem of smoke contamination was resolved and soon overshadowed by the contamination of indoor air from other organic compounds. Lavoisier doing research on the composition of air in public spaces recorded a carbon dioxide concentration in the air much above average in crowded confined spaces. Our own breath was newly considered as our greatest enemy [17] and it was concluded that ventilation with outside air was the only way to avoid people to suffer from a intoxication by carbon dioxide. 

Along with the progress made in the field of thermodynamics and the theory of heat came the discovery of the motive force of convection. The observation that air rises above a heat source lead to new natural ventilation designs which made use of the effect of buoyancy. In 1863 Henry Gouge invented what was called ”Gouges’s atmospheric ventilator” [10]. This was a device with a kerosene lamp mounted in a duct. The air in the duct was pulled up by the installed heat source leading to suction at the end of the duct which allowed room air to be actively removed. Other designs with steam instead of a kerosene lamp followed. An improved design introduced by Billings allowed the steam not only to be used for aspiration of air but at the same time for heating of the delivery air. Billings design was able to heat and to provide ventilation to the building and made a separate radiative heating system unnecessary. The development of compact steam engines gave a boost to this combined heating and ventilation system. Again Billings proposed a minimum ventilation rate of 30 cfm corresponding to about 50 $m^3/h$ per person in buildings. He has shown before that insufficient ventilation facilitated the spread of diseases such as tuberculosis. This ventilation rate requirement could only be satisfied by mechanical ventilation and was therefore very welcome by manufacturing industries of mechanical devices. Further improvements on mechanical ventilation systems and health concerns made the air handler system to become standard for conditioning of the interior environment.

The use of new construction methods and materials lead to more impermeable buildings. Internal heat gains due to lighting exploded with the start of the 20th century such that cooling was regularly integrated in air handling systems since 1920. The conceptual design of the air handling systems otherwise, has almost remained unchanged from its origin in 1890s [31]. The oil crises in 1973 made energy prices to go up such that energy consumption for building services became an issue. New standards were established to promote the design of more energy efficient plants. Unchanged was the conflict between energy issues and minimum ventilation rate requirements for indoor air quality (IAQ) reasons. It was searched for new control approaches to further reduce
energy usage. The variable air volume (VAV) systems replaced the commonly used constant air volume (CAV) systems. These VAV systems are used until now, especially in the U.S. In Europe air handlers have been less widespread. Heating and ventilation rather stayed separate such that other heating and ventilation designs and principles were used. Different standards also distinguished the development of building services in Europe from the one in the U.S.

2.1.2 A Classification of HVAC Systems

The classification of HVAC systems depends on the understanding of the term HVAC. When the HVAC system is synonym to an air handling unit (AHU) that combines all functions of heating, cooling and ventilation a different classification has to be made compared to the situation where HVAC is understood as a general term for building services with separate systems for each service. HVAC systems in the sense of AHU are not further treated in this work. Instead a classification only for separate heating and ventilation systems is given. In that sense air condition (AC) is not covered at all because the ventilation is not assumed to serve the purpose of climate control but solely the purpose of maintaining a good indoor air quality (IAQ).

Heating System

In the generation system the heat required to heat up the building and to provide the domestic hot water (DHW) is made available. Different ways in providing heat can be differentiated. Fossil fuels are most commonly used to gain heat in a combustion process. A simple burner or an absorption heat pump are two possibilities. In contrast to a burner a heat pump allows environmental heat to be used such that less high grade energy is required to satisfy the task of heating. An alternative to combustion is a heat pump with a vapor compression cycle driven by electricity. Compression cycle heat pumps can be distinguished depending on the heat source and the heat transfer media they use. Air to air, air to water or geothermal heat pumps with an open or closed water/brine-water cycle are commonly used. As a second alternative to combustion systems direct systems are to be mentioned. These systems are characterized by directly using a heat source with a sufficiently high temperature level such that no energy conversion process is needed no more. Examples for this type are solar heating systems using solar collectors or high temperature geothermal systems. If district heat is available, it can be used in the same way. Another type of direct heating since no complicated conversion
system is needed is the resistance heating. This type simply works through
dissipation of electrical current in a resistive element. Similar to combustion
systems it converts high grade energy into heat with a large loss of quality.

For the distribution of the heat most systems are closed loop water sys-
tems. A piping system is used to distribute the heat within the building.
An exception to this are air heating systems. For these system instead of a
piping system, an air ducting system has to be installed. The air is either
from the outside, fully recirculated or a combination of both. For the water
piping system several types of circuits are possible. The four general types
of circuits are full series, diverting series, parallel direct return and parallel
reverse return (Tichelmann circuit) [14]. These circuits differ in controllabil-
ity, in system pressure drop and temperature distribution. The selection of a
circuit mainly depends on the size of the installation and the characteristics
of the terminal loads, i.e. the emissive elements.

The emission systems use the physical principles of convection and radia-
tion to transfer heat to the air and to surrounding surfaces respectively. The
temperatures used in the heating system mainly determines the design of the
emission system. Radiators, convectors or panels are typical emission systems.
They mostly differ in the supply temperature and hence the surface area used
and the way the routing of the airflows induced by natural convection is han-
dled. Classical radiators work with supply temperatures around 60 degree
Celsius. Panel heating is designed for lower supply temperatures. As an alter-
native to panel heating, slab heating can be named. Low heating temperature
systems generally require large surface areas to achieve sufficiently high heat-
ing capacity. In case of slab heating supply temperatures close to the room
temperatures can be used. In heating mode a supply temperature around 30
degree Celsius and in cooling mode a supply temperature around 18 degree
Celsius are sufficient if the building envelope is adequately insulated. The use
of supply temperatures close to the room temperature allows for a very effi-
cient operation of the heating system in conjunction with a heat pump. In old
buildings, built before 1970, that are usually poorly insulated high tempera-
ture radiator based systems are predominantly used. Air heating systems in
contrast to water-bearing systems only rely on the mechanism of convection
to heat up the room air. The system can be connected to an air distribu-
tion system or work locally. Air heating systems either run with outside air,
recirculated air or a combination of both.
Ventilation System

Concerning ventilation it can be most generally differentiated between natural, mechanical or hybrid ventilation. Natural ventilation is if no mechanical power input but only natural forces are used to drive air currents through the building. The two driving forces for natural ventilation are wind or buoyancy forces. Wind induced ventilation can be either single-sided or cross ventilation. It is driven by pressure differences between the inside of the building and the outside where the air intake is located or between opposite sides of the building. The dynamic head of the flow on the stagnant windward side of the building is converted into static pressure. The opposite is true for the lee side where a small static pressure situation arises due to large flow speeds. A qualitative pressure distribution around a flat-roofed building that is exposed to the wind with one side is shown in Figure 2.1. Buoyancy induced ventilation, also called stack ventilation is driven by density difference caused by a difference in temperature between the inside and the outside of the building.

Figure 2.1: Pressure distribution around a flat-roofed building under influence of wind. Source: [3]
Cold air has a larger density than warm air such that a pressure difference over the building envelope is created depending on the height of the columns of air of different temperatures. Wind-induced and buoyancy-induced ventilation can be combined together to form a more reliable system. Stack ventilation requires a temperature difference between inside and the outside to be present while the wind-induced ventilation necessarily requires the presence of wind in order to work.

In contrast to natural ventilation, mechanical ventilation relies on the input of mechanical work to drive the air through the building. Hybrid ventilation combines the concept of natural with the one of mechanical ventilation. A hybrid system can be viewed as a two-mode system which is controlled to minimize energy consumption. The operation mode of a hybrid system varies according to the changing environment taking maximum advantage of ambient conditions at any point in time [12],[13]. Mechanical ventilation systems can be divided into supply-only, exhaust-only or balanced system [23]. As the naming suggests supply-only systems use one or more fans to supply air to the inside of the building relying on leakage to remove the air from the building. Exhaust-only systems use one or more fans to extract the indoor air relying on infiltration to replace the extracted air. The balanced system mechanically supplies the same amount of air as it is mechanically extracted.

Supply-only similar to exhaust-only systems are a relatively cheap ways to ventilate a building. Supply-only systems however show several disadvantages that make this approach unsuitable. It pressurizes the building such that moist inside air can enter cavities in the walls and the roof and lead to condensation problems. Because the indoor air is lost through leakage no heat recovery is possible. Exhaust only systems are often found in apartment houses with extraction points in the kitchen and/or in the bathroom and toilets. This makes sure that air contaminated with odors does not further spread into other rooms such as living or sleeping rooms. Exhaust-only systems allow heat recovery to be implemented but the infiltration of cold outside air may cause problems of draught. Balanced systems are most advanced and commonly used in larger office buildings and in apartment houses of the new generation. These systems allow for a good control over the ventilation rate and the air distribution. Heat can be recovered from the exhaust air and the preconditioned supply air allows for maximum comfort. This approach also enables large ventilation rates to be used without any feeling of discomfort. The space requirement for the machinery is largest for balanced systems and smallest for a exhaust-only system. When a purely centralized system is used space requirement is significant. The space requirements for different options can be evaluated using Figure 2.2. Taking an example of centralized air handling with full climate control for an
air flow rate of 16'000 m³/h the area required is 82 m² and the room height is 2.8 m. This results in a volume of almost 230 m³ in the building that has to be reserved just to install the air handling equipment.

A further criterion in the classification of ventilation is the air flow pattern that is created in the ventilated space. These patterns are referred to as short-circuit, completely mixed, displacement or piston flow [5], [20], [9]. Sometimes local exhaust ventilation (LEV) is also considered as a separate ventilation strategy [3]. All strategies differ in the guidance of the air flow and hence in the way they dilute and remove contaminants from the indoor environment. Also they differ concerning comfort aspects such as temperature and velocity distribution in the space. Different ventilation strategies with its air flow pattern is shown in Figure 2.3.

Short circuit flow is an unwanted effect and not an aim of design. Mixing ventilation is based on the dilution principle. Fresh air is injected into the room with large momentum and a high degree of turbulence with the goal to mix with the room air. This principle suits well the purpose of temperature control because through the effect of mixing induced temperature gradients decay

Figure 2.2: Space requirement in the service room for centralized air handling depending on the air flow rate. The curves in the graph represent from the top to the bottom the room height required by the equipment, the space requirement for full climate control, for climate control excluding humidification, ventilation including supply and exhaust, pure exhaust. Source: [7]
quickly. Fairly large temperature gradients can therefore be used. In a mixing system, room air is stirred such that temperatures and contaminants are equally distributed over the entire room. The concentration at which contaminants are removed from the room is hence equal to the average concentration in the room air. Building materials, furniture, carpets but also humans are sources of contaminants and odors. Most of these contaminant sources are disperse while humans are local sources. For this reason complete mixing can not be the most efficient approach to remove contaminants.

The primary goal of ventilation is to extract contaminants as efficiently as possible and to supply clean air to the space [20]. Various measures are available to express the quality or the appropriateness of a ventilation system and to compare different air distribution strategies with each other. Earliest contributions to efficiency definitions for the characterization of the ventilation performance were made in the 1980’s by [24]. An short overview of efficiency definitions introduced by different authors is given by [5]. The performance of ventilation can be quantified on a local as well as on a global scale. Global indices give information about how a room as a whole performs ventilation wise. Local indices give the same information but for a specific location within a room. It can be evaluated for any point within a room. A workgroup that has led to the publication of a guidebook on ventilation effectiveness [20] suggests in agreement with the general opinion a classification of ventilation effectiveness indices into two groups. On one side an index representing the ability of a system to exchange the air in the room expressed by the so called air change efficiency. On the other side an index representing the ability of a system to remove air-borne contaminant, the contaminant removal effectiveness.

Displacement ventilation in contrast to mixing ventilation relies on the stratification of temperature and contaminants in the room. Depending on the temperature of the air and its respective density an equilibrium layer exists
at a certain height above ground. Warm air moves towards the ceiling while cooler air stays at the bottom. In a displacement system fresh and cool air is introduced into the room with low momentum at the floor level. Driven by buoyancy the fresh air from the floor rises around heat sources such as humans but also computers or similar devices towards the ceiling. The contaminants in the occupied zone are disposed to the upper, warmer zone by the convective streams around the heat sources. A separation in the room is achieved with a layer of fresh air in the occupied zone and a highly contaminated upper zone where the exhaust is located. This air flow pattern in contrast to mixing leads to lower contaminant concentrations in the occupied zone. Because of the dependency of this approach on natural convective forces it can not be used for air heating. The air supplied has to be below the average room temperature. Theoretically, displacement ventilation works properly if the air supply rate matches the flow rate of the convective streams above the heat sources. Since the convective heat flux around a single person is around 60 to 100 m$^3$/h [32] the resulting ventilation rate would become too high compared to typical recommendations from building codes. CO2 measurements in practice however showed that even with an air supply considerably lower than the theoretical feed air demand for the convective streams an improvement of the air quality in the occupied zone can be achieved when compared to a mixing situation [30].

Similar to displacement ventilation is the piston ventilation where the fresh air moves like a piston through the room and pushes the contaminated air out. This strategy is primarily used in clean rooms. The design requires a nearly laminar flow in order to minimize the dispersion of contaminants. Unlike the displacement ventilation where buoyancy forces are the driver for air displacement the piston flow mainly relies on the momentum of the supply air [3]. The most effective way of ventilation with the least contaminant concentration in the room is the local exhaust ventilation (LEV). This approach removes the contaminants at the source and avoids the further spread of contaminant in the room. The effectiveness of this ventilation depends on the capture efficiency of the extractor hood. LEV is most commonly applied in industries where large and discrete contaminant concentrations, i.e. strong sources are present. Also common is LEV in kitchens where beside the removal of contaminants the removal of heat and moisture plays an important role.
2.1.3 Classical Paradigms for Building Service Systems

First Law Systems

Since the early days of operation of buildings with mechanical systems, huge improvements have been made in terms of energy efficiency of these systems. Mechanical and electrical components have been refined and new control strategies have been implemented but beside the control the concept of the systems have not significantly changed. Efficiency in the process of heat generation using a boiler has been improved mainly by optimization of the combustion process and by recovering the heat from the flue gases through the condensation of the water vapor by preheating the entering stream of water. The thermal efficiency often expressed as the annual fuel utilization efficiency (AFUE) of state of the art condensing boilers goes up to 95% and hence there is no large potential left for further improvements. In moderate climates as found in central Europe the largest impact on the energy performance of the buildings has the quality of the envelope. By insulating buildings also using new materials the heat losses of new buildings compared to old, uninsulated buildings are tremendously decreased. But also here, there is a limit to the amount of insulation installed for economical but also architectural reasons. The strategy of mere energy saving is in future not sufficient to achieve the goal of cost effective but sustainable, zero carbon buildings. The restriction of the designer’s perspective to the first law of thermodynamics, i.e. the energy conservation principle keeps them away from finding new concepts and new potentials for further improvements. To overcome this situation not solely the quantity but also the quality of different forms of energy has to be considered. Attempts to capture these quality aspects are undertaken by the building codes considering the primary energy usage related to the usage of secondary energy. This is achieved by introducing primary energy factors for different energy carriers and countries. Although this approach is helpful to catch the inefficiencies in the energy conversion process on the way to the building it does not help to understand where losses in quality take place within the building. The second law of thermodynamics has to be introduced in order to understand the dispersion of energy, i.e. the loss of quality within a thermodynamic process.

Pressurized, Hierarchical Systems

Looking at classical heating and ventilation systems they have in common that they usually work as pressurized hierarchical systems. Heat generation or the treatment of the outdoor air happens at a single spot within the building from
where it is distributed to the space. Systems that follow this approach can be termed "centralized systems". Centralized systems rely on hierarchical distribution structures with tree topologies. For the transportation of the media such as water or air the entire system is pressurized. Because of the uneven transport distances and their related flow resistances between different paths, throttling is required to allow for an even supply. This type of pressurized system with large terminal resistances is very stable but at the same time very inefficient. Throttling from high to low pressure is the process of dissipating potential energy contained by a fluid into waste heat. In large buildings the complexity of the distribution system and the transport distances from a centralized source to the actual locations of supply get very large and so do pressure losses. In contrast to ventilation the heat distribution system for heating is a fully closed system. This offers the possibility for choosing more appropriate topologies to minimize pressure losses. The Tichelmann circuit for instance insures that the total flow resistance of the connection to every terminal load is the same and no throttling for transportation is needed. Unlike the water distribution system the air distribution system relies on large ductwork that significantly influences the structure of a building. Ductwork for centralized air distribution is usually too large to be integrated in the building structure such that suspended ceilings and other cavities have to be created for hiding those installations. This of course results in space being lost for use and also it reduces the flexibility of the architect for spatial design.

2.1.4 Changing Boundary Conditions: New Prerequisites for HVAC

Like in the past history of HVAC, all technical systems have to adapt to changing boundary conditions and new requirements. The built environment although it seems to be a very conservative sector was and is subject to continuous change. In the recent past, significant changes are taking place because of the necessity of a more sustainable operation of buildings for the future. The theoretical base and technology is available to design advanced building systems. What is needed are new concepts, paradigm shifts in the way buildings are operated and supplied with services. As an example, air handling units that have functions of heating and ventilation combined may be substituted with systems specialized to a single task. Decoupling ventilation from the heating demand such that it is not used for climate control but only to assure good IAQ offers a large variability for new system designs and optimization. The amount of air exchanged with the environment can be controlled independently from the heating system. Also, splitting up large centralized units
into many smaller units operated in parallel can result in a larger variance for new concepts.

Ongoing improvements of insulation materials, construction methods transform buildings into systems with an indoor climate distinct from the outdoor climate. The combination of air tightness, large resistances to heat fluxes over the boundaries and growing cooling loads due to the spread of information technologies further increase the need of these buildings for control of IAQ, temperature and humidity by machinery. Energy flows over the boundaries are minimized and buildings are approximated to the "ideal" of closed systems. In that sense, heat recovery from ventilation was a huge step towards energy conservation on a buildings level and is now commonly used for larger buildings with mechanical ventilation. Improvements like heat recovery from the exhaust air make other heat fluxes leaving the building to become more significant and hence triggers further improvements. Although hardly implemented until now heat recovery from waste water will soon establish to further decrease heat losses from buildings.

The optimization of mechanical equipment merely based on the energy conservation principle at some point is exhausted. No fundamental improvements can be found no more only adopting this perspective. Introducing the second law of thermodynamics allows to pinpoint new potential for further improvements. While in times of the oil crisis in 1973 the primary issue was energy and its related prices, today the issue of climate change caused by the emission of greenhouse gases has added to it. The HVAC systems of today have to account for the actual situation and to operate buildings under the premise of minimum greenhouse gas emissions.

2.2 New Paradigms in the Field of Building Services

Building services gained much more attention in the last few years because of the issue of climate change. More and more, fossil fuels are being substituted with alternative renewable fuels or with electricity. The importance of electricity in the energy mix in near future will rise and with it the problem of bringing sufficient renewably generated electricity to the grid. Many countries debate about their future electricity generation; whether it will be nuclear, fossil-fuel based or renewable. There are projects like the Trans-Mediterranean Renewable Energy Cooperation (TREC) which works toward a fully renewable electricity supply for Europe. The cooperation is an initiative of the German
association of the Club of Rome and the Hamburg Climate Protection Foundation. It envisions wind and solar power stations to be installed in the Middle East and North Africa where solar irradiation is high and to transport electricity over high-voltage direct current (HVDC) lines to Europe. Many studies carried out by the German Aerospace Center (DLR) are available and approve the feasibility of this project [36], [37]. Critics argue that the project embodies large political risks and uncertainties although technically feasible.

As long as buildings are heated with fossil fuels or use electricity that originates from non-renewable sources they are large emitters of greenhouse gases and are therefore subject to further optimization in the future. Existing building service systems today often reached their performance limit and for a further optimization new systems and methods are needed. A suitable method to judge the thermodynamic performance is the exergy method based on the second law of thermodynamics including the aspect of "quality" of energy. Using the concept of exergy allows to select appropriate energy sources to perform a certain task and to minimize the losses of quality.

On a system level, a change from hierarchical and centralized systems to decentralized and networked systems offers new opportunities for further optimization. This change from a classical energy perspective to an exergy perspective and the transition from centralized to decentralized systems is the focal point for the research carried out in the BSG at ETH. The two paradigms of exergy minimization (lowEx) and the decentralization can be seen as two pillars building the fundament for this work.

2.3 Exergy and the lowEx Paradigm

Applying the concept of exergy to the building sector to further improve the thermodynamic performance of buildings and to pave the way for reductions of the greenhouse gas emissions is termed here the lowEx paradigm.

2.3.1 Exergy: A Definition

The energy in a closed system undergoing some thermodynamic process is conserved but as stated by the second law of thermodynamics the potential of the system to do work decreases during the process due to irreversibilities. Combining the conservation principle of the 1st law with the quality aspects of the 2nd, leads to a new description of the usefulness of energy in a system. This new concept is called exergy and is very useful in identifying the loss of work potential in a thermodynamic process. The name exergy was first
introduced by Zoran Rant [22]. His choice was made such that it reflects the principal properties of this concept. First it was looked for a name similar to other thermodynamic state variables such as enthalpy or entropy. Then, most important it was looked for a decent prefix, a "differentia specifica" which reflects the fact that exergy is the work that can be extracted in a process, i.e. the available work. For this reason "Ex" was chosen.

There are many definitions of exergy. Most of them differ only slightly and describe the same concept in different words. A modern restatement of Gibbs’ formulation is:

*Exergy can be defined as the maximum theoretical useful work obtained if a System S is brought into thermodynamic equilibrium with the environment by means of processes in which the S interacts only with the environment.*

Exergy is hence defined as the work available in a mass as a result of its nonequilibrium condition relative to some reference condition [1]. It is the portion of energy in a system that can be transformed into work through a reversible process until the system is in equilibrium with its environment [21]. The portion of energy contained in a system after it has reached its equilibrium state is called anergy [21]. This part of energy has no potential left to do work and can also be seen as the part of energy that has fully dispersed. Conversely is the exergy the part of energy that still has the possibility to disperse [28]. Exergy can also be looked at as "ordered energy" and e.g. internal energy of matter as "disordered energy" [16]. This idea of order is closely related to the concept of entropy which is a measure for the randomness in a system. As already stated before, there is a limit for the conversion from a disordered to an ordered form of energy reflected by the concept of exergy.

### 2.3.2 A Short History of Exergy

The thermodynamic concept of exergy was first introduced in 19th century and has been applied since then in various fields and is in recent years also finding its way into the building sector. A short historical overview on the exergy concept is subsequently presented. This overview is a summary of the extensive literature review presented by [26].

Sadi Carnot in 1868 was the first one to define an "availability" of energy for the conversion to work. In 1873 Gibbs introduced an expression for the available work including a diffusion term based on entropy. About ten years later Gouy and Stodola derived independently from each other an expression for the useful energy and identified the lost work potential as the product
of the entropy produced and the temperature of the environment. This result is known as the Gouy-Stodola-Theorem found in several textbooks as for example [6]. In 1953 Zoran Rant introduced the term exergy and defined it to denote the “technical working capacity” (technische Arbeitsfähigkeit) [22]. Alternatively to the above used definition of exergy, Baehr in 1962 [4] introduced another widely used definition which is: Exergy is the portion of energy which is entirely convertible in all other forms of energy. Baehr also used a definition for anergy which can be seen as the fully dispersed part of energy without any potential left to do work. The concept of exergy and the method of exergy analysis is introduced in almost every serious thermodynamics textbook. There are also textbooks available that are fully devoted to the exergy topic. Two fundamental books written in German by [25] and [4] have strongly influenced generations of European researchers in that field. More modern books are written by [16] in 1985, by [1] and [34], [33]. The recently most referenced book on exergy analysis is the one written by Bejan [6]. Since 2000 there is also EXERGY - An International Journal by Elsevier which was then replaced in 2004 by the International Journal of Exergy (IJEx) teaming up with another prestigious publisher, Inderscience Publishers Inc..

2.3.3 The lowEx Paradigm

The lowEx paradigm is deduced from the exergy concept and simply means that little exergy in the process shall be consumed. In contrast to the trend toward zero energy buildings (ZEB) the lowEx paradigm focuses on the exergy demand and destruction. Several different definitions of ZEB are available [35]. The most common definition for ZEB in the U.S. requires the building to have a zero off-site net energy demand. This enforces the installation of on site capacity to supply sufficient energy to the building originating from renewable sources. Relying on renewable sources only, the concept of ZEB is synonymous to the one of a zero carbon building. The lowEx paradigm aims to minimize the supply of high valued energy but does not rule out the possibility of supplying energy that has been harvested off-site. It rules out the use of combustion processes to supply low temperature heat for thermodynamical reasons and promotes the import of renewably generated electricity in that it minimizes the capacity required. The generation of electricity though does not necessarily have to be performed on site but rather where conditions are ideal. As an example: Producing electricity from solar irradiation using solar cells in Switzerland needs about a 1.5 times larger capacity than generating the same amount of electricity in the south of Spain [15] as it can easily be verified by looking at Figure 2.4. In that case it is more cost effective to produce in Spain.
and import the electricity to Switzerland. Having all electricity produced renewably makes the lowEx building being a zero carbon building as well.

Introducing the exergy concept into the building sector leads to a paradigm shift in the conception and the design of buildings. This shift is necessary in order to make further improvements of the performance of buildings possible and cost effectively free buildings completely from CO2 emissions during operation by allowing a more efficient use of high grade energy.

Figure 2.4: Photovoltaic Solar Electricity Potential in European Countries. 
Source:[15]

The Concept of Exergy Applied to Buildings

The concept of exergy has only recently been applied to the built environment. Among the first researchers in that field with a publication about the application of the exergy analysis to the built environment [27] was professor Shukuya from Japan. Several publications (parts 3-9) followed. In 1999 the IEA ECBCS research project Annex 37 Low Exergy Systems for Heating and Cooling was started. The ECBCS is the Implementing Agreement on Energy Conservation in Buildings and Community Systems established by the International Energy Agency (IEA) and it focuses its work on ways to improve
energy efficiency in buildings. The Annex 37, lasting five years, had the aim
to promote the rational use of energy and to accelerate the use of low valued
energy sources for heating and cooling. This research project resulted in a
guidebook [2] providing theoretical background, analysis tools, examples and
strategies for the design of low exergy heating and cooling systems. As a con-
sequence of the work of Annex 37 the LowEx.Net has been founded. It is a
network of International Society for Low Exergy Systems in Buildings. The
objective of the network is to assess existing technology for low exergy heat-
ing, cooling, etc. and to enhance the development of new technologies and
concepts for low exergy systems for buildings. As a continuation of Annex
37 the Annex 49 Low Exergy Systems for High-Performance Buildings and
Communities started in 2006. It aims at improving on a community and on a
building level general processes where energy / exergy is used. It also aims at
promoting new, innovative technologies and at disseminating the exergy con-
cept for the evaluation of system performance. Another meaningful project is
CostExergy which is about the Analysis and Design of Innovative Systems for
Low Exergy in the Built Environment. The main goals of this project is to
disseminate new knowledge and to design tools that facilitate the application
of the exergy concept to the built environment. CostExergy ist funded by the
European Cost program. Cost stands for Cooperation in the Field of Scientific
and Technical Research. In the U.S., the American Society of Heating, Re-
frigerating and Air-Conditioning Engineers (ASHRAE) has a technical group
that is concerned about the exergy analysis for sustainable buildings. Like
the other projects focusing on the exergy concept also this technical group
continues the work initiated by Annex 37 and will help to make the exergy
concept to be more commonly applied in the design of new buildings.

2.4 Paradigm of Decentralization and Minia-
turization

Any system may be divided into subsystems which may be again subdivided
until the limit of resolution, i.e. the level of ”elementary particles” is reached.
By dividing systems into subsystems the system and its constituent elements
are put into a hierarchical order. This structural hierarchy does not say any-
thing about the interaction between the subsystems and hence not necessarily
mean that there is a hierarchy in control. If there is a hierarchy in control
we talk about master/slave or centralized systems. Conversely we can define
decentralized or networked system that do not adhere to this hierarchical way
of control. Moving from hierarchical, centralized to networked, decentralized systems in the building service technology is therefore a paradigm shift that opens the possibility to realize a huge amount of new concepts and designs that have not been exploited so far.

2.4.1 Definitions

A centralized system is a system in which one element plays a major role and controls the operation of the system [11]. This implies a master/slave relation between subsystem whereas a small change on the master level affects the behavior of the over-all system. An example for such a system is a totalitarian regime where decisions made at the root of the system affect all other participants within the system.

Decentralization is the process of dispersion of decision making governance [38]. Crucial in the concept of decentralization is the difference between hierarchy based on authority or on an interface. Authority is the unequal power relationship while the interface describes the relationship between two or more participants of equal power. The more decentralized a system the more it relies on lateral relationships (interface) and the less it acts on command (authority).

In systems theory decentralized systems are naturally occurring, usually self regulating systems that function without an organized center or authority. Decentralized systems lack a center of control and are commonly composed of several components that work in unison and that form together a stable structure. In a networked system where decentralized units interact with each other an over-all system behavior may occur which can not simply be deduced from the behavior of a single unit. This phenomena of the ”creation of a new property” is known as emergence [8] and is a feature of complex systems. A truly complex system is completely unredudant in its structure or behavior such that no aspects of its structure or behavior can be inferred from any other and hence no further simplification of the system is possible [29].

Miniaturization is the process of continuous down scaling of any kind of device. Most prominent is the result of miniaturization in electronics where this process was witnessed by an empirical observation known as the Moore’s law. Moore published in 1965 a prediction of how silicon components would behave over the next ten years. From the experience at Intel where he was director of the R&D Laboratories in the semiconductor division he concluded that the complexity for minimum component costs increases roughly at a rate of factor two per year [18]. This observation and the prediction for future development of integrated circuits was later corrected to what today is used as the Moore’s law. It states that the number of transistors that can be
inexpensively placed on a single chip approximately doubles every two years [19]. A graphical representation of Moore’s law is shown in Figure 2.5.

![Figure 2.5: Original Graph of Moore’s Law. Source:[18]](image)

### 2.4.2 Hierarchical vs. Networked Systems

Typically, building services such as freshwater supply, heat generation or ventilation are provided by hierarchical and centralized structures. Hierarchical structure’s complexity over-proportionally increases with increasing size and hence there is a limit to growth of these systems. In very large buildings it can be observed, that services are split into several subunits in order to be able to handle complexity. Still, these centralized systems are significant structural elements and largely influence the actual structure of the architecture. Building services, being an ”infrastructure” however are not meant to determine the architecture but rather to provide services to it. For this reason a transition from centrally organized to decentralized systems seems very interesting and promising. A recent trend in the last ten to twenty years toward decentralized system also confirms the potential. In contrast to large, centralized systems, small, decentralized units can easily be integrated into the building structure.
and also allow for a simple installation or upgrading of the capacity. With the ongoing miniaturization of electrical components, space savings coming along with the decentralized technology becomes even more significant. The decentralized approach differs in its philosophy from hierarchical systems in that it generates and provides the service on the spot where it is demanded. This means that decentralized units can act independently of each other and hence allow for a maximum flexibility. This increased flexibility with the on-demand principle and the local service provision allows for significant energy savings. On the other hand the decentralized units acting based on simple, local rules build together a swarm and are able to solve a complex, higher-ranking task. This phenomena, called "emergence" is a non-trivial behavior of a swarm that cannot simply be inferred from the behavior of a single participant of the swarm. Examples in nature are the formation of fish swarms in order to increase the probability of each individual to survive a predator’s attack. For building services it can be an opportunity to be able to solve a complex, superordinate task based on a very simple set of rules for the individual to solve its subordinate task. Using this approach eventually leads to better performing buildings.
Bibliography


Chapter 3

General Theory - Mathematical Formulations

3.1 The 2nd Law of Thermodynamics and the Definition of Entropy

3.1.1 Sadi Carnot and Thermodynamic Cycles

Before the definition of entropy has been established and mathematically formulated, there was a worded statement of the 2\textsuperscript{nd} law of thermodynamics. All the statements made by different people (Clausius, Kelvin, Planck) were closely related to all-day observations and to experience gained from experiments performed on thermodynamic cycles. Both, the statement of the 1\textsuperscript{st} and the 2\textsuperscript{nd} law of thermodynamics started with Sadi Carnot's analysis of thermodynamic cycles. Carnot (1796-1832) was concerned about the question of how far heat engines (Figure 3.1) can be improved in terms of efficiency and whether there is a limit for improvements. Carnot found out that it is favourable for any heat engine to work between two reservoirs of the temperature $T_h$ and $T_c$ and exchange its heat only at these temperatures. He also found out, that the efficiency of a heat engine that works between two temperature levels (Carnot machine) is independent of the working agent employed. This latter finding can be verified with a simple thought experiment. If one imagines two competing heat engines (actually one working as a pump) working with different agents (water, ammonia, etc.) which exchange the same amounts of heat and at the same temperatures their work output and input
respectively must be of the same magnitude. If not so it would be possible to create motive power without consumption of either heat or any agent. This of course is contradictory to sound physics which deny the existence of perpetual motion.

![Carnot Heat Engine Diagram]

Figure 3.1: Carnot heat engine with two infinite thermal reservoirs of constant temperatures $T_C$ and $T_H$.

3.1.2 Clausius’ Statement of the $1^{st}$ and the $2^{nd}$ Law of Thermodynamics

Based on Carnot’s observations Clausius (1822-1888) formulated the $1^{st}$ law of thermodynamics (he defined the state function $U$ as internal energy) and expressed mathematically the efficiency of a Carnot cycle for an ideal gas for any range of temperature.

$$\eta = 1 - \frac{T_c}{T_h} < 1 \quad \text{for} \quad T_c = 0 \text{ K which is impractical} \quad (3.1)$$

Clausius then devoted his time to more general cases; away from ideal gases, away from Carnot cycles away from cycles of whatever type away from reversible processes. Criticizing Carnot’s opinion about transformation of heat in cycles (Carnot did not know about the $1^{st}$ law of thermodynamics), Clausius made a statement which became known as the $2^{nd}$ law of thermodynamics.
Clausius statement of 2nd law: Heat cannot pass by itself from a colder to a warmer body or more precise: It is impossible for any system to operate in such a way that the sole result would be an energy transfer by heat from a cooler to a hotter body.

This statement does not rule out the possibility of heat transfer from a lower to a higher temperature like refrigerators or heat pumps do. But the sole result implies that there must be some other effects involved to make this transfer possible, i.e. the interaction with the environment by an input of work. It is impossible to construct a refrigeration cycle that operates without input of work. Clausius tried to prove this intuitively correct statement and considered again the Carnot cycle with its universal efficiency definition for cycles. Given the 1st law of thermodynamics he now knew, that \( W_0 = Q_h - |Q_c| \). From this, the general definition of the efficiency is:

\[
\eta = \frac{W}{Q_h} = 1 - \frac{|Q_c|}{Q_h}
\]  

(3.2)

If the engine is a reversible Carnot engine the efficiencies must be the same, thus

\[
1 - \frac{|Q_c|}{Q_h} = 1 - \frac{T_c}{T_h}
\]

and therefore

\[
\frac{Q_h}{T_h} = \frac{|Q_c|}{T_c}
\]

(3.3)

From this it was clear that not heat \( Q \) but the quantity \( \frac{Q}{T} \) stays unchanged in a Carnot engine. Clausius called \( \frac{Q}{T} \) entropy (greek for transformation). He saw two transformations going on in a heat engine:

- transformation of heat to work (1st law of thermodynamics)
- passage of heat of high temperature to heat of low temperature

Using the definition of the Carnot efficiency it follows: \( \eta_c = 1 - \frac{T_c}{T_h} = 1 - \frac{|Q_c|}{Q_h} \). The heat \( Q_c \) returned at the low temperature can only be zero for \( T_c \) being zero, which is impractical (\( \eta = 1 \)). Therefore you always need a cooler and hence two reservoirs. Heat cannot be completely transformed into work (\( |Q_c| > 0, \eta < 1 \)). This leads to another formulation of the 2nd law of thermodynamics stated by Kelvin and Planck.

Kelvin-Planck statement of 2nd law: It is impossible for any system to operate in a thermodynamic cycle and deliver a net amount of work to its surroundings while receiving energy through heat transfer from a single reservoir.
From the results of Carnot cycles Clausius extrapolated to general cycles and introduced a definition of entropy $S(T, V)$ as a state function whose significance is not restricted to cycles. The result is the mathematical formulation of the 2\textsuperscript{nd} law and is an equality. Equating the general efficiency definition and the Carnot efficiency with $\eta \leq \eta_c$ (equal sign for reversible, $<$ sign for irreversible cycle), it follows:

\[
1 - \frac{|Q_c|}{Q_h} \leq 1 - \frac{T_c}{T_h} \quad \text{and} \quad -\frac{|Q_c|}{Q_h} + \frac{T_c}{T_h} \leq 0, \quad Q_c < 0 \quad \text{(heat flux out of the system)}
\]

\[
\frac{Q_c}{T_c} + \frac{Q_h}{T_h} \leq 0 \quad \text{(equal sign for reversible, $<$ sign for irreversible cycle)} \quad (3.4)
\]

In the Carnot cycle the sum of the heat flux divided by the temperature at which the heat flux takes place ($\frac{Q}{T}$) meaning the sum of the entropy in the cycle disappears. For general reversible cycles which can be composed by the superposition of a large number of Carnot cycles with an infinitesimal heat flux the summation of the entropy fluxes becomes:

\[
\oint \delta Q_{\text{rev}} \frac{T}{T} = 0 \quad (3.5)
\]

This integral gave rise to the definition of the state function entropy $S(T, V)$. Therefore the entropy difference between two arbitrary points $a$ and $e$ in a reversible process can be expressed by:

\[
S(T_e, V_e) - S(T_a, V_e) = \int_A^E \frac{\delta Q_{\text{rev}}}{T} \quad (3.6)
\]

For an irreversible cycle where $\eta < \eta_c$ the sum of the entropy flux transferred is negative and therefore the integral (3.5) becomes negative too. The combination of both, the reversible and the irreversible case into one formula known as the Clausius inequality becomes:

\[
\text{Clausius inequality : } \oint \frac{\delta Q}{T} \leq 0 \quad (3.7)
\]

This implies that in an irreversible cycle there is a larger entropy flux out of the system ($Q_c/T_c$) than into the system ($Q_h/T_h$) and hence there must be a production of entropy in the cycle due to irreversibilities.
In the irreversible case, the difference of entropy between two points \( a \) and \( e \) must be larger than the sum of the entropy fluxes over the boundaries because entropy is produced but the difference between two states is path independent. The integral (3.1.2) becomes:

\[
S(T_e, V_e) - S(T_a, V_e) > \int_a^e \frac{\delta Q}{T}
\]

Combined, the reversible and the irreversible case into one equation is:

\[
S(T_e, V_e) - S(T_a, V_e) \geq \int_a^e \frac{\delta Q}{T}
\]  

Since entropy \( S(T, V) \) is a state variable the entropy difference in a cycle must always disappear. As we have seen before in a irreversible cycle the net entropy flux over the cycle is negative and therefore different from zero. This discrepancy comes from the internal irreversibility and is termed entropy production. For the further discussion of the entropy production it is necessary to separate the entropy difference \( dS \) into two parts \( dS = d_eS + d_iS \). The \( d_eS \) represents the change in entropy due to exchange of heat and matter with the environment, while \( d_iS \) is the internally produced entropy due to irreversibilities, i.e. dissipative mechanisms. The \( d_iS \) is always larger than zero unless we have a reversible process for which it is equal to zero. With this in mind we can formulate the entropy balance for closed systems (no exchange of matter) as follows:

\[
S_e - S_a = \int_a^e \frac{\delta Q}{T} + S_{prod} \quad \text{or} \quad S_{prod} = (S_e - S_a) - \int_a^e \frac{\delta Q}{T}
\]

\[3.2\] 2\textsuperscript{nd} Law of Thermodynamics and the Concept of Exergy

The 2\textsuperscript{nd} law of thermodynamics is an addition to the concept of the 1\textsuperscript{st} law of thermodynamics and introduces a restriction to thermodynamic processes telling in which direction they can take place. While the 1st law does not exclude the possibility of a spontaneous heat transfer from a low temperature reservoir to one of a high temperature this process does not take place in nature. The 2nd law specifies in what direction processes actually take place and how the dispersion of energy puts a limit to all real processes in that
They are irreversible. Heat can not be fully transformed into work. The work potential of the energy contained by a heat source depends on its temperature level. Real processes only take place in the direction of increasing entropy. The increase of entropy leads to a reduced potential of the system to do useful work.

The exergy of a closed system can be defined combining the 1st and the 2nd law of thermodynamics. For this purpose we consider any closed system that currently is at an arbitrary state and is reversibly brought to an equilibrium state denoted with index 0.

The energy balance for this system has the following form:

\[ E_0 - E = Q_{rev} - W_{rev} \]

where

\[ E = U + KE + PE \]

with \( U \) being the internal energy and \( KE \) and \( PE \) being the kinetic and the potential energy respectively. For the equilibrium state \( KE_0 = PE_0 = 0 \) and hence

\[ E_0 = U_0 \]

The complete balance becomes:

\[ (U - U_0) + KE + PE = Q_{rev} - W_{rev} \]

Reversible heat transfer is only possible if no temperature gradient across the systems boundary is present and therefore only if it takes place at the temperature of the equilibrium \( T_0 \). The reversible heat flux becomes:

\[ Q_{rev} = T_0(S_0 - S) \]

The work for the expansion consists of two parts, the volume work against the environmental pressure and work that can actually be used, termed the useful work.

\[ W_{rev} = W_{use,rev} + (V_0 - V)p_0 \]

If we put back all terms in the energy balance formulation and solve for \( W_{use,rev} \) we get an expression for the maximum useable work of a reversible process which we call exergy.

\[ E_x = W_{use,rev} = U - U_0 + p_0(V - V_0) - T_0(S - S_0) + KE + PE \quad (3.9) \]

The exergy is an extensive quantity and hence a specific exergy being \( e_x = \frac{E_x}{m} \) can be defined. In eq. 3.9 it can be seen that kinetic and potential can directly
be considered as exergy. These parts of energy can directly be transformed into work. The remaining terms:

\[(H - H_0) - T_0(S - S_0)\]

is usually termed the physical exergy [3]. Other exergy terms such as the chemical, nuclear exergy exist and have to be considered depending on the process analyzed. Since all the quantities appearing in eq. 3.9 are state variables, exergy is a state variable as well.

3.2.1 Loss in Available Work: Exergy Loss

Real processes involve irreversibilities which are unavoidably accompanied by an increase of entropy in the constrained system. It can be seen as a transition from a more organized form of energy to one with a higher degree of randomness. There are two phenomena found in irreversible processes: First, the direct dissipation of work into internal energy of the system such as friction and second, ”spontaneous non-equilibrium processes” [3] in which a system tends to move from a state of a non-equilibrium to one of an equilibrium. Examples for this second type of processes are unconstrained expansion or the equalization of temperature. Due to the irreversibilities in a process and its related entropy production there is an unavoidable loss of work potential. The connection between the entropy production within a system and its exergy loss is given by the Guy-Stodola theorem. This theorem is derived by combining the energy flux and the entropy flux balance for an arbitrary system which exchanges work and heat with its environment and solving for the difference between the maximum, theoretical work output of a reversible process and the effective work output, i.e. the lost useful work or exergy.

The Guy-Stodola theorem writes out as:

\[\dot{W}_{\text{lost}} = \dot{E}_{\text{x,lost}} = T_0\dot{S}_{\text{prod}}\]

The exergy lost in an irreversible process is directly proportional to the entropy produced in this process.

3.2.2 Exergy Balance

Similar to energy or entropy a balance for exergy can be put up for either closed or open systems. While the energy balance is a law of conservation, the exergy balance is a law of degradation of energy [3]. For a closed system the
energy and the entropy balance between the states 1 and 2 have the following form:

\[ E_2 - E_1 = \int_1^2 \delta Q - W \]

\[ S_{\text{prod}} = S_2 - S_1 - \int_1^2 \frac{\delta Q}{T} \]

Multiplying the entropy balance by \( T_0 \), subtracting it from the energy balance (Guy-Stodola) and using the exergy definition from eq. 3.9 lead to the following exergy balance for closed systems:

\[ E_{\text{x}2} - E_{\text{x}1} = \int_1^2 \left(1 - \frac{T_0}{T_1}\right) \delta Q - \left[W - p_0(V_2 - V_1)\right] - T_0 S_{\text{prod}} \tag{3.10} \]

The same formulation as presented by eq. 3.10 can be used for a flux balance restating the differences as derivatives. The formulation of the exergy balance for open system includes mass fluxes over the system boundaries and as a consequence also exergy fluxes. The exergy balance for open systems stated as a flux balance becomes:

\[
\frac{dE_x}{dt} = \sum_{i=1}^{k} \dot{m}_{i,in} e_{x,\text{flow},i,in} - \sum_{i=1}^{m} \dot{m}_{i,out} e_{x,\text{flow},i,out} - \dot{W}_0,\text{use} + \sum_{i=1}^{n} Q_i \left(1 - \frac{T_0}{T_i}\right) - T_0 \dot{S}_{\text{prod}} \tag{3.11}
\]

where \( e_{x,\text{flow}} \) is the so called flow exergy defined as:

\[ e_{x,\text{flow}} = (h - h_0) - T_0(s - s_0) + \frac{1}{2} v^2 + gz \tag{3.12} \]

**Exergy Balance for Moist Air**

In most air conditioning applications the air cannot be simply assumed as a single component but rather must be considered as a mixture of dry air and water vapor. Including the moisture content of the air makes it necessary to account for the change of physical properties of the air as well as to include the chemical exergy into the exergy balance. As previously introduced the physical exergy is the maximum work obtainable when a stream of substance is brought from its initial state to the environmental state \( (P_0, T_0) \) by physical processes involving only thermal interactions [3]. The chemical exergy is the exergy of the
stream of substance when the state of the substance is the environmental state and it is brought to the dead state [3]. The environmental state can be regarded as a restricted equilibrium with a constant temperature $T_0$ and pressure $p_0$ of the system and the environment. The dead state can be considered as an unrestricted equilibrium at which the chemical potentials of a system and the environment are also equal. The chemical exergy of a mixture of substances can be defined as the work needed to reversibly compress each component of a mixture from its partial pressure to ambient pressure.

The thermodynamic properties of humid air are usually described by treating it as an ideal gas mixture of components that individually exhibit ideal gas behavior. The state of a mixture is fully defined by three properties: The temperature, pressure and the composition. The composition is given by the mole fraction of either component. The sum of the mole fractions is always equal one.

$$x_a + x_v = 1, \quad \text{where} \quad x_a = \frac{n_a}{n_a + n_v}$$

and $n_a, n_v$ are the moles of air and water vapor respectively. Based on the mole fractions the mole fraction ratio $\tilde{\omega}$ can be defined.

$$\tilde{\omega} = \frac{x_v}{x_a}$$

Similarly, a specific humidity or a humidity ratio of the masses (water vapor, dry air)

$$\omega = \frac{m_v}{m_a}$$

can be defined. The specific humidity is related to the relative humidity $\Phi$ by the expression

$$\omega = \frac{0.622}{P/(\Phi P_{sat}(T)) - 1}$$

where $P_{sat}$ is the saturation pressure of water vapor at a given temperature and $P$ is the total pressure which is the sum of the partial pressures of dry air and of the water vapor in the mixture. The relative humidity

$$\Phi = \frac{p_v}{p_{sat}(T)}$$

is defined as the ratio of the partial pressure of the water vapor to the saturation pressure of water vapor. To calculate the saturation pressure at a given temperature the Arden-Buck equation [2]

$$p_{sat}(T) = 6.1121 \times \exp \left(\frac{(18.678 - T/234.5)T}{257.14 + T}\right)$$
can be used where in this equation temperatures are expressed in degree Celsius and pressures in hPa.

The general form of the physical flow exergy as shown in eq. 3.12 can be applied to a stream of air using the law of an ideal gas. In that case the equation takes the form

\[
e_{x,\text{flow}} = c_{p,a} \left\{ (T - T_0) - T_0 \left( \ln \left( \frac{T}{T_0} \right) - \frac{R_a}{c_{p,a}} \ln \left( \frac{p}{p_0} \right) \right) \right\} + \frac{1}{2} v^2 + gz
\]

If moist air is assumed the flow exergy equation has to be further adjusted. Instead of using the specific heat and ideal gas constant for dry air only it has to be replaced by values of the mixture. The specific heats of the mixture are evaluated using

\[
c_{p,mix} = x_v c_{p,v} + x_a c_{p,a}
\]

The individual \( c_p \) for water vapor and dry air can be calculated using an empirical expression found in [4]. The specific heats are a function of the temperature and the constant A, B and D are given for the water vapor and the dry air respectively as

\[
\frac{c_p}{R} = A + BT + \frac{D}{T^2}
\]

The \( \bar{R} \) is the universal gas constant and the constants for the water vapor and the dry air are listed in Table 3.1. For the ideal gas constants a value

<table>
<thead>
<tr>
<th>component</th>
<th>A</th>
<th>B ([K^{-1}])</th>
<th>D ([K^2])</th>
</tr>
</thead>
<tbody>
<tr>
<td>water vapor</td>
<td>3.470</td>
<td>1.450×10^{-3}</td>
<td>0.121×10^{-3}</td>
</tr>
<tr>
<td>dry air</td>
<td>3.355</td>
<td>0.575×10^{-3}</td>
<td>-0.016×10^{-3}</td>
</tr>
</tbody>
</table>

\( R_v = 461 J/(kgK) \) for water vapor and \( R_a = 287 J/(kgK) \) for dry air can be found in literature [1]. Including further a new term expressing the chemical exergy the total flow exergy of moist air per unit mass of dry air is given by the equation

\[
e_{x,\text{flow}} = \left( c_{p,a} + \omega c_{p,v} \right) \left\{ (T - T_0) - T_0 \left( \ln \left( \frac{T}{T_0} \right) \right) \right\} -(1 + \bar{\omega}) \frac{R_a}{c_{p,a}} \ln \left( \frac{p}{p_0} \right)
\]

\[
+ R_a T_0 \left\{ (1 + \bar{\omega}) \ln \left( \frac{1 + \bar{\omega}_0}{1 + \bar{\omega}} \right) + \bar{\omega} \ln \left( \frac{\bar{\omega}}{\bar{\omega}_0} \right) \right\} + \frac{1}{2} v^2 + gz
\]

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Bibliography


Chapter 4

Over-all, Integrated System Concept (Research at the BSG)

The BSG at the ETH Zurich dedicates its research to the goal of the zero carbon building. The zero carbon building is understood as a building that does not emit any carbon dioxide during operation. This goal is pursued choosing the lowEx approach meaning that processes and machinery which minimize the input of high grade energy to the building during operation are analyzed and promoted.

The lowEx approach is a necessity at the moment to allow for an economically feasible realization of the zeroCarb building. High grade energy is substituted with low grade energy from the environment as far as possible. In the future when enough renewably generated electricity will be available the claim for exergy minimization can again be weakened and it will be possible based on a consideration of the financial investment to determine whether more low or high grade energy should be used to operate the building. Of course the necessity for CO2 free energy remains unchanged. Comparing the earth surface with the surface of a photovoltaic (PV) element it is clear that both, when exposed to the sunlight lead to a conversion of the radiative solar energy. In the ground, heat is stored at a low temperature and hence it represents a low grade energy source. The PV element instead is able to directly convert some of the radiative energy into electricity, i.e. high grade energy. Both surfaces (ground vs. photovoltaic) are similar in that they are simply
exposed to the available sun light and hence there is no reason to refrain from using either of these energy sources. Fact is, that the high grade energy from the PV is stochastic and less easy to store than the low grade energy. The latter is stored in the environment and constantly available, day and night. The technical building systems are likely to be composed of a mixture of high and low grade energy according to the characteristic properties of their sources and according to the present needs of the buildings.

Buildings can be looked at as open systems with mass-, energy- and information fluxes crossing their system boundaries. Energy efficient buildings minimize these fluxes and try to approach the open system to a closed system in terms of energy. This goal is achieved by improving the insulation of the building envelope but also by recirculating energy fluxes that are coupled to mass fluxes. These energy is recaptured using heat recovery in the exhaust air and the waste water streams leaving the building. The lowEx approach more specifically tries to minimize the flux of high grade energy over the system boundary into the building. This minimization process at first hand aims at the minimization of exergy destruction within the system. Only when exergy destruction is minimized it can be ensured that the exergy flux into the system corresponds to the minimal theoretical flux required to operate the system. In contrast to this physically meaningful approach the mere energy approach in conjunction with primary energy factors is commonly used. This latter approach tries to minimize the input of high grade energy by minimizing the input of energy instead of aiming at the reduction of the exergy destruction within the system. The first step towards minimization of the exergy destruction is the replacement of the commonly used condensing boiler for heat generation by a heat pump. This implies that no more mass fluxes containing carbon are crossing the system boundaries. This is true when the electrical current supplied to the building is renewably generated, free from the use of fossil fuels. In that case the building becomes free from carbon dioxide emissions during operation. Having carried out the first step, replacing the combustion process by the heat pump an important aspect of the lowEx approach becomes the tuning of the heat pump’s performance to further reduce the input of high grade energy. This requires to understand the building not as a loose compound of individual components but rather as one complex, interconnected and integrated system. Only when having this systemic view on the building the full potential for exergy minimization is revealed.

The heat pump cycle is most efficient and requires least input of electricity the lower the lift from the low to the high temperature is. The high temperature level is determined by the heating and the domestic hot water temperatures. The heating temperatures can be minimized using large sur-
face areas in the emission system while the domestic hot water temperatures are more or less fixed by the temperature required for dish washing and for the shower. The lower temperature is given by the heat source chosen. This heat source mostly found in the environment can either be the outside air, the ground or a well. Particularly interesting is the ground because its temperature increase with increasing depth. Beside these environmental sources the energy fluxes coupled to the mass fluxes leaving the building are predestinated, relatively high temperature sources to be used. Hence, for a maximum efficiency of the heat pump the appropriate sources have to be selected. Having several sources with different temperature levels available allows further to dynamically optimize the system performance in that it uses the source with the highest temperature when high temperatures for domestic hot water is demanded. On the other hand when only low temperature for heating are required another source can be used. In that sense, the heat pump performance can be optimized when combining all the thermal systems in the building and dynamically manage the sources and the sinks. This again underlines the necessity of understanding the building as a system of strongly interconnected components.

The systemic notion of the building allows for finding significant potential reductions in the use of high grade energy. In this work those potentials are identified and quantified for some elements of the entire building system. This building system with its subsystems and elements is shown in Figure 4.1. The superordinate system includes all facilities including lighting, heating, ventilation and domestic hot water. Two subsystems being a thermal and a ventilation systems are identified and partly treated further in this work.

Within the ventilation system a demand controlled ventilation (DCV) system with a local exhaust is analyzed. For the supply a decentralized approach together with a networked distribution system is considered. For this decentralized supply system the additional saving potential for running it in a hybrid mode is also assessed. The savings of high grade energy are quantified by looking at the coupling of the ventilation system with the heat pump as well as looking directly at the active components such as fans in the ventilation system. One scientific paper included in this work is dedicated to the exhaust while two papers are concerned about the supply.

As a part of the thermal system a novel active insulation system has been assessed. Its heat transmission properties and the potential savings of high grade energy that come along with using this active insulation system within an integrated heat pump system was analyzed and presented in multiple published papers. This thermal element of the over-all building system however is not included in this work and hence not further explained.
Figure 4.1: Over-all building system with its subsystems and components. Elements of the ventilation system enclosed in the bold type ellipse are analyzed and discussed in this work.

The paper included right after, introduces the integrated system approach pursued with the research carried out at the BSG. This includes the investigation of new technology suitable to be used within lowEx buildings and being optimized for the operation with a heat pump. Further it covers the paradigm of decentralization and the resulting possibility for distribution and integration of technology in the space and structure of the building. On a process level this integration is continued by accounting for impacts of technological decisions made during design. This is achieved by putting up a design methodology that allows for a gross evaluation of the performance of a building already in the very early design phase and hence gives the architects a tool to evaluate the consequences of their work. The integration of technology into both, the design process as well as in the physical building structure supports the system understanding of buildings and leads to more sustainable buildings as highlighted in the paper.
4.1 Paper: Clima 2007, WellBeing Indoors

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Effective Infrastructure Distribution, Implementing An Integrative Concept for Sustainable Office Spaces

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SUMMARY

Typical infrastructure in office buildings does not facilitate the creation of a sustainable building, effectively using energy, materials, and space. One must recognize that sustainability doesn’t only refer to the use of energy, but also to spatial quality, user comfort, and ease of facility management. Different from the commonplace hierarchical systems, our goal is to distribute infrastructure such that highly networked structures arise. Our approach includes the horizontal integration of building infrastructure systems such as air, power, heating and lighting, and the vertical integration of building life cycle stages from design to implementation to management. These vertical processes are continuous, forming a “digital chain.” The concept is being integrated into the renovation of the Building Services group office at the ETH Zürich. The office space has tested these concepts while utilizing this new design theory, and will continue to provide data in analysis to support the development of more sustainable office spaces.

INTRODUCTION

Among researchers in the area of building services and technology, it is common knowledge that these services and technologies are in fact the source of two-thirds of global electricity demand. This makes buildings alone responsible for a third of all anthropogenic greenhouse gas emissions [1]. Recent conglomeration of energy demand statistics has shown that when combining the portion of residential, commercial, and industrial energy demand sectors that are related to building operation and construction, buildings are directly or indirectly responsible for over half of the global energy demand [2].

Clearly, the building sector offers an immense opportunity to have an impact on future energy demand. Sustainable design, energy efficiency, ecological design, and natural systems, among many other concepts, have all been developed and used in the creation of buildings to try and address their global impact. Still, although improvements have been made, they have not been integrated into all parts of building construction and usage.

The area of building services is the direct source of energy demand and is not well integrated into the overall design perspective, especially in the important early stages of design. Interdependencies between architecture and building services are not captured. Being aware of these interdependencies is a major step toward developing more sustainable buildings. This is done through better vertical integration along the design chain and better horizontal integration of building service components. Not only does this directly reduce the energy demand of buildings, but it also eliminates constraints imposed by separately designed infrastructure. The view of the interdependencies provides a larger range of options to improve the indoor space. Europeans spend upwards of 90% of their lives in buildings [3].
Unsatisfying spatial quality leads to less productive users. Removing constraints on spatial quality indirectly improves the sustainability of a structure.

With a holistic and inclusive planning and creation of buildings the palette of available technologies also increases. Instead of a small number of options available for one sector of design, a wide variety of systems can be considered when the entire project can be visualized through a digital design model.

Digital planning technologies gained major importance in the fields of design, construction and facility management over the last decade. However, these fields often remain separated, and information is not communicated between them. In spite of the major economic and environmental impact of building services, the usage of digital technologies in this field is not as thoroughly developed as in other fields of the building industry.

The goal of the project described herein is to define, study, and implement a framework by which seamless design processes and advanced technology can be researched, tested, and implemented in one laboratory, which will guide the creation of more sustainable buildings. In order to achieve this, a methodology was laid out to define this process based on current digital design practices and available technologies. These were applied to the ongoing renovation of the Building Services Laboratory (GT Lab). This was done in part during the real-time design process of the renovation, as well as into the renovation itself to allow future testing and analysis of design techniques and technologies. It is a functioning office space allowing for analysis of the design space from its design phase up to its usage and management. Technologies have been implemented that can be utilized in modeling processes to verify design decisions. The lab provides a system to test various permutations of how a digital chain can be linked together from design phase to operation, thus maximizing the flexibility of all phases and pieces of a building. The methods used in this design concept were applied in a collaborative network of engineers and architects.

METHODS

In the sense of horizontal integration our group picked appropriate, innovative technologies and combined these into a well performing ensemble. Every technology, before being integrated into the overall system, has been carefully chosen based on an educated selection method. The product development and implementation is done in cooperation between the chair and industry. During the selection procedure the aim was always to select technologies that contribute to a comfortable indoor environment. Again, sustainability was considered not just in a narrow, energy related sense, but in a universal sense relating to spatial quality, ease of use, etc.

**Heating Ventilation and Air Conditioning (HVAC)**

For the method of selection of HVAC components major quality criteria were the exergy efficiency, architectural and spatial aspects, flexibility, and the effectiveness of the ventilation system. Thermodynamic analysis based on the exergy principle has been applied to buildings in order to identify potential improvements of heating systems. The exergy concept classifies energy sources according to their “quality” or potential to do work. It is therefore appropriate to use energy sources of low “quality” to fulfill a low “quality” task such as maintaining a building at a low temperature [4]. Heating systems based on heat pumps are more exergy efficient than systems based on fuel driven burners. By choosing low supply temperatures the operation of heat pumps becomes even more efficient. For that reason the exergy design
principles led to the selection of heating and ventilation technologies that are designed to operate with low supply temperatures. In order to minimize the exergy demand of the complete HVAC system, the static heating and the air supply/ventilation must be decoupled from each other. This decoupling exploits the potential energy savings of low ventilation rates without affecting the indoor temperature.

Besides energy efficiency other criterium, such as indoor air quality (IAQ), were not overlooked. Therefore, the use of low ventilation rates required ventilation systems to have high ventilation effectiveness in order to maintain good IAQ. It was decided that the best way to maximize ventilation effectiveness was through a system that comprises of a large number of distributed sensor-controlled exhaust openings, which is based on the displacement principle. Past models and measurements were observed showing that this system achieves high ventilation effectiveness compared to standard systems. The local control characteristics of this kind of system allow the airflow rate to be dynamically adjusted according to the number of occupants being present in a room or a building [5].

The supply air system was matched to this distributed exhaust. The selection of a decentralized air supply was influenced by the potential for enormous space savings and for the flexibility this approach offers compared to hierarchical, centralized systems [6]. Very compact, decentralized supply air units were chosen and optimized in collaboration with industry for integration into the complete ventilation system.

Lighting

Lighting has both a large aesthetic effect as well as a significant impact on energy demand. In this selection method, the spatial and managerial aspects of building sustainability were directly balanced with aspects relating to energy demand. An important criterion was the integration of all lighting elements in the multifunctional ceiling panels which allowed a built-in height of 10 cm only. Also, the artificial lighting was supposed to efficiently complement existing daylighting. In order to define the necessary lighting to meet the required constant amount of 500 lux, model-based lighting simulations were run, comparing different lighting variations. It was found that LED lighting in combination with fluorescent lighting could provide highly controllable lighting with very high efficacy. In order to control the lighting color within the lab and to experiment with different ambient color settings, these LEDs had to be capable to display the full RGB color spectrum.

Power Supply and Distribution

Various supply topologies have been assessed to find the most suitable solution offering the largest flexibility and ease of use. A suitable topology for power distribution was found to be a ring along the boundaries of the space with branching lines that serve the inner part of the ring. This supply structure allows maximum flexibility for the layout of desks. The design also presented an option for new construction. Electrical routing has the potential to be done through ventilation ducting directly integrated into the neutral zone of structural floor elements between steel reinforcements [7]. This integration eliminates the need for either a raised floor or a ceiling plenum. This is not possible for a renovation where an underfloor system is required, but analyzing this system for future implementation was an important consideration in its selection.
**Parametric design model**

One goal is to expand the digital chain into the fields of building services, starting at the early design stages of a building. The huge impact of technical systems of buildings makes it necessary to consider consequences of building service design already in these early design stages. Performance evaluation has to be integrated into the architectural design process already at an early stage, up to 80% of the decisions most important for the building performance are already made within the first 20% of the design process [8]. Common CAD-Software however does not support an integrated planning approach.

For the design and ongoing research at the GT Lab an approach was chosen based on a parametric building information model. The use of a parametric model provided the capability to design, calculate, and simulate already at very early stages of design and still be able to judge spatial quality. A three-dimensional parametric geometry model was set up. Supply requirements were directly defined in the model database. Every change in geometry or equipment resulted in an instant adaptation in performance data for the selected room cell, floor, or the whole building. The evaluated technical systems are integrated into the model in a schematic. Their dependencies are defined in the database as well as in the resulting geometric model and displayed three dimensionally. Simulations like daylight and artificial light simulations can be directly accessed, comparing light distribution and visual impression with the calculated use of energy of different lighting systems.

**System Management Model**

It was found that through the use of digital control every system could be addressed and controlled throughout the power network. This included the ability to control everything from ventilation to standby power delivered to a coffee machine. Sensors could be used directly in the same network integrated with the lighting and ventilation systems, which they help steer. Defining a digital power management system realized avenues to reduce control infrastructure and facilitate the optimization of the entire system. The selection method also required a highly flexible system that could be easily modified for research applications, which is provided in the digital control system. This allows for a broader analysis of the digital chain from planning to operation.

The planning model has been furthermore adapted for steering and monitoring purposes within the GT Lab. The distributed systems such as those used in the GT Lab impose a paradigm shift towards the extensive use of management software. Traditional, hierarchical systems of building services like central ventilation for example, rely on centralized control and steering. Distributed components were chosen and implemented at the GT Lab, which lack this central steering component. Instead they are highly networked and act as complex systems, reacting on local and overall conditions. This reaction is based on a large amount of information obtained from sensors and systems including user input to fulfill individual comfort needs. This information has to be synchronized, resulting in control commands created by software-based rules. The actual physical components like the small decentralized ventilation units are inexpensive and energy-efficient, but are not capable of steering the whole topology of units. Therefore, the resulting network of distributed units has to be coordinated by software.

By incorporating sensors monitoring indoor data as well as environmental conditions this network could be steered and controlled to react more sensitively and efficiently. The
software would need a digital model of the built environment including geometrical and topological information of the building as well as the applied building service technologies.

RESULTS

Renovation Overview

Within the Building of the Department of Architecture (HIL) one hundred square meters has been selected by the Chair of Building Services to be transformed into a new office space and laboratory. Before the renovation, the existing infrastructure (lighting, sound absorber, air supply/exhaust) was included in a 60 cm high plenum. This means that in this particular case about 1/7 of the actual floor height was occupied by the infrastructure. During the retrofit of the office space the suspended ceiling was completely removed and a raised floor with a height of 17 cm was installed instead. The existing supply air system was substituted with compact decentralized supply air units and a ducting system incorporated in a raised floor. Electrical and IT cables were integrated into the underfloor ducting. The lighting that was previously installed in the suspended ceiling has now been integrated into multifunctional ceiling panels. These panels have a thickness of only ten centimeters and cover about 50% of the ceiling area. They combine the features of static heating with activation of thermal mass, lighting, sound absorption, and also integrated sensor-controlled exhaust openings. The control of all electrical devices is managed with a unique control system. This management system cannot only be used to control the building services but offers a simple and powerful interface for facilities management. A cutting edge microchip attached to the power supply of each device allows a unique way of digital power management. The detailed components of the renovation are described below and how each component is integrated is illustrated in Figure 1.

Figure 1: The components integrated into the renovation of the GT Lab.

Underfloor Installations

The fresh air is supplied to the office at a single point below the ceiling. From there two branches of the duct supply air to a U-shaped channel, integrated in the raised floor. This channel is separated into two parts and the airflow of the two supply branches can be regulated, such that different pressure conditions in the two chambers of the channel can be obtained. Since the façade could not be modified and the fresh air comes from the centralized system, the channel was designed to simulate the different pressure conditions that would arise on a buildings façade due to the influence of wind. The decentralized supply air units being installed in the raised floor are connected to the U-shaped channel on their suction side and to another ducting network on their pressure side. The ducting network connected to the
supply interconnects the different sides of the channel representing the facades, and allows a regular distribution of fresh air over the entire floor space. For this network a meshed topology is chosen because of its balancing pressure characteristics. The system simulates how the ventilation network can use the wind pressure differences between the facades for air transport and hence to reduce the energy costs.

The supply air ducting network also carries cables (power, data) such that central workstations can be easily connected over a branching line to the main installation built as a ring line along the building façade. Tubing was used with an oval cross section, and newly developed compact decentralized air handling devices make the integration into the floor possible.

**Multifunctional Ceiling Panels**

The ventilation system with low momentum air supply through the floor and extraction at the ceiling follows the well-known displacement principle [9]. The system designed for the GT Lab utilizes this principle in combination with localized exhaust vents that use sensors to monitor local air quality. This principle of local waste air extraction is often encountered in industries and has been analyzed for buildings [7]. Sensor-controlled exhaust openings with a built-in flap, as proposed by [7] have been integrated into the panels distributed along the ceiling. Air with a higher contaminant concentration can be extracted and the over-all ventilation rate is reduced because of the demand-controlled nature of this system. This system with its autonomous local control characteristics is much simpler and more robust than a conventional system with a master-slave control.

The ceiling panels can be used to cover static temperature control. Hot or cool water in a two-wire system activates the thermal mass of the construction and convectively heats or cools the room air with a power of about 40 W/m² within the comfort zone. In modern, well-insulated office spaces it is enough to cover 50% of the ceiling area to meet the heating power requirements. The ceiling panel is covered with a thin, perforated sheet-metal plate that has good sound absorption characteristics. The same area as needed for heating is sufficient to reduce the sound reverberation time to a value below 0.6-0.7 seconds.

The standardized ceiling panels that incorporate the features of heating and sound absorption have been altered and the lighting equipment has been included. Two different kinds of illumination with two different technologies were implemented. Energy efficient fluorescent tubes are used to meet the higher lighting requirement while for the ambient lighting LED technology has been used. The LEDs are mounted 10 cm apart along a 1 m strip, eight of which are integrated into each ceiling panel. The power of a one of the LEDs used is 1.4 W and its luminous efficacy is approximately 40 Lumen per Watt. Compared to a common light bulb with an efficacy of about 18 Lumen per Watt the LED is a clearly superior, and also a rapidly developing technology. The LED lighting has been designed, and each strip can be controlled to compensate for light gradients occurring in the room due to daylight influences. On the other hand it serves as an ambient light providing a convenient atmosphere.

**Digital Lab Management**

A newly developed chip has been integrated into the lab’s infrastructure. The chip is attached to the power line just before every electrical device connected. The power line is used to carry the control commands to the chip. This transmission method makes it unnecessary to install extra cables to communicate with terminal devices. Because of the unique addressability of
each terminal device this technology offers the possibility to implement very effective power management that reduces standby losses almost to zero.

For the management of the ventilation, exhaust air flaps, heat supply, power control chips, lighting, as well as sensor data a “building operating system” is implemented in cooperation with a software company. Therefore the digital chain is extended to actually control physical components seamlessly. All the systems exist in the virtual model as well as in physical reality, and simulation can be directly compared to actual physical operation data. A small server will allow access to all systems via the Internet. A web-based interface enables the user to control and monitor the GT Lab physical environment. Individual lighting schemes can be stored, and global settings for the entire environment are also available. Sensor data is recorded, which includes energy consumption for each device, local CO₂ concentrations, and activity recorded from the motion detector. With the building operating system it will be possible to group systems such as ventilation units individually, simulating the flexible management of space, reacting on changing user needs. No physical replacement of systems is necessary when spatial user requirements change, the system units can just be regrouped within the software model.

DISCUSSION

It has been found that it is possible to create a research space that employs the flexibility to study both horizontal and vertical aspects of design. Analysis can be done on the performance of new technologies for buildings systems while also understanding how these components can be integrated in the design process. By controlling and studying the preliminary design and modeling procedures that can then be implemented into a flexible, yet normally occupied office space, improvements in both physical operation of components can be made along with improvements in the ease of implementation, integration, and management. These last improvements have benefits easily overlooked in a strictly experimental laboratory. The laboratory office is both a normal working environment as well as a laboratory and has realized some unique concepts for a building research environment. In the physical renovation of this space some preliminary analysis of this design process has also taken place.

While many technical results from the laboratory are forthcoming in future research, the current research into the laboratory design provides a valuable description of how more sustainable indoor environments can be created and studied. It does not analyze one area of improvement, but instead allows for an integrated view in the research itself, as all systems are available for study through one digital interface. The use of this digital control system in particular, will contribute to the ability to study the entire system. It also provides a direct link back to the design phase and thus can be used to test preliminary design ideas and models. This is one way in which a digital chain can be maintained for analysis in the lab. As more integrated practices are being pursued for the design, construction, and operation of buildings, this concept presents a method for how these concepts of integration can be incorporated and studied in academic research.

Continued work on the design process is also planned in an expanded renovation of more space within the HIL building where the 100 m² lab is located. There is potential for expanded renovation of the adjoining office space, which currently houses the Chair of Computer Aided Architectural Design (CAAD) with whom collaboration studying the digital chain already exists. The entire space would then be designed to house all the chairs under the Institute for Building Technology (HBT). It has also been proposed to work toward a renovation of the entire building using concepts that would be refined in the GT Lab.
The data gathered in the GT Lab will refine the design model to be used in further designs. The direct link between the virtual and physical building model will help to validate virtual simulation results as well as optimize steering operations for the implemented services. Future research will also look at the optimization of the underfloor air distribution and power supply integration. Integrating this ducting into the infrastructure of a building will be studied. It is expected that economics savings will come from space savings, and energy savings will come from more exposed thermal mass.

One of the clearest observations made from the resulting creation of the GT Lab is the tremendous benefit of improving the spatial quality. As the office is situated directly adjacent a space which maintains the former construction, a direct comparison is possible. The higher ceiling and highly controllable lighting has substantially improved the indoor quality.

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Chapter 5

Demand Controlled Ventilation (DCV) Based on Local Detection and Local Contaminant Extraction

5.1 Concept And System Description

This chapter describes a new design of a demand controlled ventilation system that introduces the approach of local exhaust ventilation (LEV) to office type buildings. This is achieved by introducing a large amount of extraction points along the ceiling that are connected to the main exhaust branch and controlled by a local carbon dioxide sensor. The high number of extraction point allows to capture room air with high contaminant concentrations assuming an uneven distribution of contaminants in the room due to discrete sources such as people. The sensor control at the extraction point determines the ventilation activity and the allocation of exhaust power. The connection of the extraction points to a single exhaust branch leads to a variable exhaust power per point depending on the amount of simultaneously activated points. If few exhaust openings are simultaneously activated a large exhaust power per opening is available. For
the situation that all openings in the space are active, the total fan power is distributed over the entire number of openings and a smaller exhaust power per opening results. Starting with a simple mathematical model of this LEV type of ventilation, CFD simulations and measurements were performed to demonstrate the effectiveness of this new approach.

As shown in Figure 5.1 a person sitting in a room creates a convective plume rising towards the ceiling. According to the principle of displacement ventilation, fresh air from the floor supply is entrained by the boundary layer around the person leading to low contaminant concentration in the breathing zone. At the same time pollutants produced by this person are transported within the plume to the ceiling. As a consequence a higher contaminant concentration is found in the plume. The local detection system causes the exhaust system to locally increase the air flow rate in order to maximize the amount of pollutants that are removed from the room. The other extraction points do not react unless the contaminant spreads out and reaches a neighboring extraction point at a contaminant concentration above the setpoint.
5.2 State of the Art Review

Sick building syndrome is known in buildings where ventilation is not sufficient such that bad indoor air quality (IAQ) is obtained. Discomfort but also health threats are possible consequences of bad IAQ. Sufficiently high ventilation rates but also an appropriate design allowing for an effective replacement of used with fresh air is necessary to avoid such a situation. Unfortunately, the higher the ventilation rates are chosen, the lower becomes the energy performance of the building. The exchanged air has to be conditioned (heated or cooled and eventually de-/humidified) before it enters the building. Using heat recovery from ventilation in winter has weakened this statement a bit but in summer for cooling the problem remains unchanged. Consequently, a good ventilation system has to work effectively and exchange the required amount of air with the outdoor but to run only when it is required. Using this demand controlled ventilation (DCV) approach good IAQ with minimum average air change rates can be achieved. Depending on the climate, the design of the ventilation system, different usage and occupancy, energy savings in office building range between 5 and 50 % [11], [17]. In the design of DCV it is important to identify what are the driving pollutants and what are the key indicators. The driving pollutants are those that determine most the IAQ, meaning that they are either generated at the highest rate or are most harmful and hence require the largest ventilation rate [11]. The key indicator is a suitable, measurable surrogate for the pollutant to be detected. As pollutants are for example considered bioeffluents producing odors, volatile organic compounds (VOC), radon but also small particles such as dust and humidity [2].

The ventilation system has to keep the concentration of these pollutants at an acceptable level. For the bioeffluents mostly being a driving pollutant in office spaces, carbon dioxide (CO2) is a good indicator [11]. At the same time CO2 itself is an important contaminant that should be kept below a concentration level of 1000 to 1500 ppm in office spaces [19]. CO2 is predominantly used for DCV in offices and is a better indicator for human odors than the alternatively used humidity or mixed gas [15]. CO2 can often be used to determine the ventilation rate needed but as also pointed out by [6] and [14] it is not a general measure for IAQ but only a surrogate for occupant generated contaminants. It does not capture contaminants released from other sources such as building materials. The pollutant load can differ significantly from building to building and hence different requirements for the base ventilation exist [15]. In poorly occupied spaces it may be the case that the base ventilation needed is already as high or higher than requested due to the occupant load. In such a case DCV does not add any value to the system performance. The largest
energy savings using DCV are of course to be expected for spaces with large and intermittent occupancy such as class rooms, theaters and auditoria. DCV is cost effective only if occupancy related ventilation rate is larger than the base ventilation rate. Based on a literature review, [8] conclude that sensor-based demand-controlled ventilation is especially cost effective in applications with the following characteristics: (a) A single or a small number of pollutants dominate, (b) large rooms with unpredictable and temporally variable occupancy or pollutant emission or (c) climates with high heating or cooling loads. The increasing demand for gas sensors in HVAC and the technological progress caused market prices for sensing devices to decrease significantly. First inexpensive CO2 sensors for HVAC application are available since 1990 and prices further dropped by 40% since then [16], [13]. CO2 sensors with a range from 0 to 2000 ppm and an accuracy of +/- 50 ppm are usually sufficient for HVAC applications [10]. Critical issues in the design of DCV using CO2 sensors is the sensor location and the choice of the setpoint. Concerning the sensor location it has to be paid attention that the sensor is not placed in poorly ventilated zones. According to [11], in case of displacement ventilation sensors are to be placed in the occupied zone where a representative concentration can be measured. When placed in the exhaust it has to be made sure that no short circuiting takes place since it would significantly alter the concentration measured and lead to poor ventilation. If the setpoint cannot properly determined it is preferably chosen towards low values in order to assure sufficient ventilation [10].

DCV tries to reduce ventilation rates to the amount required to minimize energy losses through ventilation. As a consequence an appropriate ventilation strategy in terms of air distribution has to be picked to assure good IAQ with the least ventilation rates and hence energy losses. Two measures are commonly used to express the appropriateness or effectiveness of ventilation; the air change efficiency or the contaminant removal effectiveness [12]. The definition of the air change efficiency is based on the concept of the age of air. It is a measure for the freshness of the air in the room and relates the time needed for the ventilation to replace the entire volume of air in the room to the shortest possible replacement time achieved by a perfect piston flow [7], [12], [9]. The contaminant removal effectiveness is a measure to express the ability of a ventilation system to limit contaminant exposure from indoor pollutant sources. It is defined as the ratio between the contaminant concentration in the exhaust air and the average concentration in the room air, each relative to the supply air concentration [7], [12], [9]. There is no single concept that describes the performance of the ventilation completely. The appropriate criterion of efficiency must be chosen based on the most desirable quality of
the ventilation system. In terms of contaminant removal the most effective way is local exhaust ventilation (LEV). LEV aims at removing contaminant directly at the source where they are generated and is predominantly used in industries where strong pollutant sources and health damaging substances are present. Typically, extractor hoods are used to have a high capture efficiency and to reduce the dilution of contaminants with the room air [9], [5]. Extracting pollutants at the source with large concentration allows minimum ventilation rates to be used, still guaranteeing low contaminant concentrations in the room. If in office buildings, humans are considered as strong sources of CO2 and bioeffluents the concept of LEV could be interesting too. The problem is that no extractor hoods can be used in this particular case to capture contaminants with a high concentration. LEV in the classical sense only works when the extraction point is close to the source because of the decay of induced air velocity with the square of the distance from the point of extraction. Beside being contaminant sources humans are also thermal loads in the building such that due to buoyancy, a convective air movement around the body of a person takes place. A boundary layer forms around the human body and entrains air such that the plume created increases with increasing height. In this plume, fresh air from the floor region but also pollutants originating from the person are transported towards the ceiling [4]. The boundary layer flow around a heat source of course depends on many factors such as the room air temperature, its gradients along the room height, the surface temperature of the heat source. But as shown by [18] the plume is relatively stable and can persist in a displacement ventilation as well as in a mixing ventilation situation. Of course it depends on the disturbances occurring in the room. If a person is moving the boundary layer decays and significant mixing is induced by the movement [3]. The measurements performed by [18] show that contaminants released close to the thermal manikins where transported to the ceiling creating local peaks. Having these peaks even though only for a fairly undisturbed situation, makes it interesting to exhaust the air in the vicinity of the person, close to where the plume hits the ceiling. Using a large amount of extraction points at the ceiling and making use of the convective flows around heat sources for the transportation of contaminants allows to have a ventilation system with a similar characteristics as LEV but for regular office buildings.
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INCREASING THE EFFECTIVENESS OF BUILDING VENTILATION SYSTEMS THROUGH USE OF LOCAL WASTE AIR EXTRACTION

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ABSTRACT

This paper proposes a new building ventilation system based on local, demand-controlled waste air extraction. The exhaust air system consists of intelligent, sensor-controlled openings which are distributed over the ceilings in a relatively fine grid. The sensors detect pollutant concentration and regulate their exhaust power according to it. By this means the extraction of waste air is concentrated to regions with direct pollutant sources and hence proves to be more effective than conventional exhaust air systems with even a smaller airflow rate. This leads to significant energy savings since less air has to be displaced, heated and moistened. The layout of the supply air system is adapted to meet the requirements of the exhaust air system. A model of a room with a person working at a desk has been examined using CFD. The results have shown that the thermally induced flow around the person lifts contaminated air up to the ceiling and allows a local extraction of waste air with high contaminant concentration compared to the average concentration of the room. The effectiveness of the exhaust air system applied to a single-room office has also been confirmed by theoretical considerations and measurements.

KEYWORDS: local exhaust, CO₂ demand controlled ventilation, ventilation effectiveness

INTRODUCTION

Ventilation systems may be classified according to their way to supply and withdraw air from ventilated areas. Mixing – or dilution systems introduce the supply air into the room with a jet of high momentum. The mixing technique has the disadvantage, that contaminants produced by humans, furniture, etc. are diluted by the room air through the mixing process and are extracted at a low concentration. The effectiveness of ventilation may be increased if the contaminant concentration of the returned air is increased with respect to the room concentration and this is exactly what happens if displacement ventilation is applied. The term effectiveness used in this report is understood as the ratio between the contaminant concentration in the exhaust stream and the concentration in the occupied zone, both relative to the concentration of the supply air. The displacement principle uses the convective forces from heat sources such as humans, computers etc. to lift contaminants up to the ceiling where they can be extracted. Meanwhile air with a temperature slightly below room temperature is introduced near the floor and displaces the warmer room air creating a zone of cool, fresh air at the occupied level. Despite of its potentially higher effectiveness, a displacement ventilation system is not always first
choice; rather it has to be considered carefully in every single case what system to be used. Displacement ventilation only works properly when the supply air matches the demand of the convective current above heat sources. In modern office premises the internal heat load due to electronic devices such as computers, copy machines, etc., due to lighting and the presence of people is around 20 W/m². These heat loads lead to a significant convective current which can hardly be matched by the supply air. The convective current at a height of 2.5 meters above floor induced by a single person ranges from 100 to 200 m³/h. Meanwhile SIA-standards (SIA 382/1) propose a ventilation rate of about 30 m³/h per person to keep the CO₂-concentration around 1000 ppm. This discrepancy of airflows shows that a common ventilation system with a low air change rate, even though designed as displacement ventilation, will rather work like a mixing ventilation since the recirculation of the convective current and downdraughts from cold windows lead to a substantial mixing of the room air. Thereby the ventilation effectiveness is reduced to the one of a common mixing ventilation. In such a situation, ventilation effectiveness can be increased by use of local waste air extraction near the contaminant source, i.e. at the ceiling above the contaminant source. Using a local exhaust air system guarantees a higher contaminant concentration relative to the room concentration to be extracted and hence minimizes the negative effect of the contaminants mixing back into the room air. Although it is obvious that a local extraction of airborne contaminants is most effective compared to any other approaches no such system applied to buildings (residential or office premises) is known to the author. In welding or chemical industry local extraction is the principal commonly applied as well as in almost every kitchen where an extractor hood is used just above the cooking stove to remove odours and water vapour effectively.

When local waste air extraction shall be applied to common office spaces of course no extractor hoods can be used but other means are needed to guarantee a high contaminant concentration relative to average indoor concentration at the exhaust location. This requirement can be satisfied if exhaust openings are arranged within a relatively fine grid on the ceiling. A spacing of about two meters between the openings should be appropriate to ensure a local extraction at a higher concentration than average room concentration. In such a local exhaust system, each exhaust opening consists of a flap (that opens and closes), a carbon dioxide sensor and a controller. All openings installed on one floor are connected to a ducting network which is operated by a single, centralized exhaust fan. The sensor of an opening is located in the air stream and permanently measures the carbon dioxide concentration in the returned air. When the flap closes because of a detected concentration below the setpoint, a minimum amount of air will still be extracted to remove non-occupant generated contaminants and to ensure a continuous measuring of carbon dioxide. The exhaust power per exhaust opening and hence per square meter installed in a local exhaust system has to be larger than in conventional ventilation systems to be most effective. It allows the concentration of the exhaust power to a small area and hence an effective cleaning of this area. Due to the relatively poor simultaneous occupation of office spaces (simultaneity factor around 0.6) and the local sensing of carbon dioxide, the exhaust power will be concentrated to only few locations with high contaminant sources at a time such that the overall ventilation rate will not grow and hence there is no need for redimensioning the central exhaust fan. Rather it is expected that the local exhaust system allows the exhaust fan to be operated at part load for most of the time.

Local extraction of course is most effective when the opening is directly located above a person’s working place but especially in open plan offices this can hardly be realized since the allocation of the workstations it usually not known during projection of the building services. For this reason a relatively fine grid of openings has to be chosen to preserve
flexibility and to ensure that the ventilation system indeed works as a local exhaust ventilation system.

The requirements of a demand controlled exhaust system have to be met by the supply air system. The air change rate of the ventilated space in demand controlled ventilation (DCV) depends on occupancy and the local exhaust activity depends on the spatial distribution of occupants. The demand for fresh air is therefore fully dictated by the exhaust system. Fresh air must be supplied to the room through many different openings distributed over the floor area in order to satisfy the local needs. This report proposes a ducting network integrated directly in the statically supporting concrete floor that accommodates the ventilated area with fresh air. The ducting system must be designed in a way that flow conditions within this network exhibit low overall velocities and therefore low pressure drops such that a homogeneous static pressure distribution is attained. Rather than introducing fresh air selectively to specific region the air supply adjusts itself to the needs of the exhaust system. A homogeneous static pressure distribution in the underfloor ducting system allows uniform air supply with a very low pressure drop over the entire ventilated area. The demand for a homogeneous pressure distribution is satisfied best when supply air is introduced to the network in many different locations. Decentralised ventilation devices arranged along the building's façade are an appropriate mean to feed the network with fresh air from the periphery. Employing gauge pressure sensors some ventilation devices may be urged to temporarily stop their air supply depending on the local wind pressure on the facade. In the case of large negative pressures on the lee-side of the building the air supply may be taken over by the devices on the luff-side. An appropriate topology of the network must be chosen such that even under conditions of inhomogeneous air supply from outside there is still a regular, homogeneous air supply to the inside.

It is widely accepted that indoor carbon dioxide concentrations can be used to indicate specific aspects of indoor air quality, but do not provide an overall measure of the actual air quality. CO₂ is however a possible substitute for other occupant-generated pollutants and for the ventilation rate per occupant. CO₂ demand controlled ventilation (DCV) is most effective in situations where high occupancy and unpredictable variations in occupancy are expected, the building and climate require heating or cooling for most of the year and low pollutant emissions from non-occupant sources are expected. Lowest energy savings are seen in office spaces, which generally have low occupancy levels (Davidge, 1991). As Emmerich and Persily (2001) report in their literature review, a key issue in DCV systems is the sensor location, i.e. whether to use a single sensor centrally located in the system return or multiple sensors in particularly critical spaces such as conference rooms. By many authors, using a lower setpoint with a central sensor is often suggested as a means to deal with variability of concentrations between different locations.

In a local demand controlled ventilation (LDCV) system like the one described above every exhaust opening has its own sensor built-in. Therefore using a local sensing of the carbon dioxide concentration eliminates the need for a careful consideration of an adequate positioning of the sensor. Unlike for a common DCV the effectiveness of a ventilation system using a local, demand controlled exhaust, does hardly depend on overall occupancy level. This is true as long as every workstation within an office has its own dedicated exhaust opening or generally spoken as long as the grid of exhaust openings on the ceiling is dense enough. Because of the potential energy-costs savings with a local exhaust system, slightly increased capital costs that arise from the adoption of sensor technology may be accepted.
The objective of this study was to understand the working principles of local exhaust ventilation and to verify the hypothesis of the increased effectiveness of such a system compared to traditional ventilation systems.

**METHODS**

Simple theoretical considerations shall help to understand the nature of local exhaust ventilation and show the increased effectiveness of it compared to common mixing ventilation. For this purpose the ventilation principles are represented by simple scalar models based on mass balances. The situation of a local exhaust system is represented by a two compartment model whereas the mixing situation is modelled with a single volume (Figure1). The mass balance for both models leads to simple, linear, 1<sup>st</sup> order ordinary differential equations in the carbon dioxide concentrations.

![Figure 1: Scalar models based on 1<sup>st</sup> order ODE’s for mass balances;](image)

**a) Mass balance for one compartment model:**

\[
V_{room} \frac{dc_{mix}}{dt} = \dot{q}_{vent}(c_{amb} - c_{mix}) + \dot{Q}_{CO2} \quad \ldots (1)
\]

**b) Mass balances for two compartment model:**

\[
V_{local} \frac{dc_{local}}{dt} = \dot{q}_{conv}(c_{occu} - c_{local}) + \dot{Q}_{CO2} \quad \ldots (2)
\]

\[
V_{occu} \frac{dc_{occu}}{dt} = \dot{q}_{vent}c_{amb} + (\dot{q}_{conv} - \dot{q}_{vent})c_{local} - \dot{q}_{conv}c_{occu} \quad \ldots (3)
\]

In the one compartment model (Figure 1a) complete mixing is assumed such that the contaminant concentration of the exhaust air is equal to the concentration in the room. The carbon dioxide source is situated within the room. In the two compartment model (Figure 1b) it is distinguished between two zones: One large zone which constitutes the room the person stands in and a small zone representing a volume of highly contaminated air just above the person. The small zone contains the carbon dioxide source and there is an air exchange between this and the adjacent zone. The airflow aspirated from the small zone originates from the convective stream above the person and exhibits the same contaminant concentration as the room average. In turn some highly contaminated air is spilled back from the small zone to the room however with a smaller flow rate i.e. the flow rate of the convective stream minus the flow rate exhausted. Within each zone complete mixing is assumed.

The general input parameters required to solve the model-equations are composed in Table 1.
An analysis of a typical single-person office has also been carried out using CFD. For this purpose a transient simulation for the time period of half an hour was performed. A commercial CFD-code using a coupled solver with a fully implicit finite volume formulation is applied. The numerical scheme is second order accurate in time and space. The CFD model incorporates equations describing conservation of mass, energy and momentum. The time-averaged Navier-Stokes equations are solved for compressible, non-isothermal flows in three-dimensions accounting for buoyancy effects, neglecting radiation. As a turbulence model a k-omega-based, shear stress transport model has been selected. The indoor air is a composition of two gases (CO$_2$, air) with different physical properties. For the carbon dioxide a transport equation is solved while the air-component of the mixture is handled as a constraint.

In a room of 6 meters length, 2.4 meters width and 2.65 meters height a person, tall 1.8 meters, stands in front of a desk. A single exhaust opening directly above the person is activated. The fresh air is supplied through openings in the floor. As exhaust opening an outlet boundary condition is applied with a fixed flow rate of 40 m$^3$/h. The supply boundaries are modelled as inlets matching the mass flow of the exhaust. Supply air enters the room with a static temperature of 20°C and a carbon dioxide mass fraction of 380 ppm. The model person has a convective heat capacity of 40 W distributed over the upper part of the body. With an area of about 1.8 square meters this leads to a specific value of about 22 W/m$^2$. Through the mouth the model person exhales air with a mass flow rate of 1.4e-4 kg/s (ca 0.45 m$^3$/h) at a constant temperature of 36 °C and a carbon dioxide mass fraction of 4 percents. This results in an additional heat flux of about 5 W and a flow rate of carbon dioxide of about 0.0118 m$^3$/h. Other heat loads are installed; such as a computers and a floor lamp with a convective heat emission of 40 W each. The computer is placed in the middle of the table in front of the model person and the lamp is placed in the rear right corner of the room.

Assuming the convective heat load to be 50% of the overall heat load the latter sums up to 245W. Relating it to the floor space, a specific heat load of about 17W/m$^2$ results. In order to remove parts of the heat produced by the sources, one sidewall of the room is modelled as a heat sink with a source strength of about -50W.

A steady state solution of the flow field without carbon dioxide production has been used as an initial guess for the transient simulation. At the first time step the production of carbon dioxide has been activated. The simulation time has been limited to 30 minutes due to extensive computational costs.

Within a facility of EMPA Switzerland, measurements of carbon dioxide concentrations for two different ventilation system setups, ventilation with/without local exhaust, have been performed.
The room where measurements have been carried out has the dimensions of 6.2 m x 4.6 m x 3 m (length, width, height). This results in a floor space of 28.5 m² and a room volume of 85 m³. During the measuring period a real person is sitting at a table in the middle of the room. The equipment of the room and its arrangement is shown in Figure 2.

![Figure 2: Measuring setup of the test room](image)

The supply air is brought into the room through a low momentum exhaust nozzle located on the west side of the room near the floor according to the principle of displacement ventilation. For the reference case waste air is extracted on the opposite side near the ceiling. In case of local exhaust ventilation an opening just above the person sitting at the table is activated while the other opening is closed.

In order to have a situation similar to the one encountered in common office premises additional heat loads are included in the test room. These heat loads consist of two thermal manikins of 80W each, a floor lamp of 150W and two computers of 80W each. Together with the real person contributing another 100W approximately a total heat load of 570W is reached. Divided by the base area of the room it results in 20W/m² which is a typical value for modern office premises. The room temperature is held constant through the use of thermally active walls and floor and the cooled supply air.

The concentration of carbon dioxide and temperatures are measured at four different locations in the system: Within the room between the north-wall and the working person at a height of 1.4 meters above floor, just above the person working at a height of 1.9 meters above floor, in the exhaust- and in the supply air stream (Figure 2).

**RESULTS**

Through solving the ODE's of the mass balances in the simple scalar-model, the developing of carbon dioxide concentrations with time is obtained. The results for both, the one- and the two compartment model are displayed in Figure 3. The curve represented by a solid line corresponds to the concentration encountered in the room when complete mixing over the whole zone is assumed; the dotted curve with circles and the dash-dot curve with five-pointed stars are the exhaust and the room concentration respectively, when a two compartment model is applied.
From this figure it can be seen that the two compartment model leads to a significant reduction of contaminant concentration in the room (occupied area). After 4 hours of accumulated time the result reaches almost a steady state. Significant differences in evolution of the concentrations are however seen within the first two hours. At steady state the contaminant concentration in the two compartment model exhibits 880 ppm in the exhaust air and 747 ppm in the room air. The contaminant concentration determined by the one compartment model reaches the same value as the exhaust stream in the two compartment model. According to this, a reduction of 133 ppm in the room air has been attained applying the two compartment model. This reduction is equivalent to the ratio between the carbon dioxide source and the convective stream and hence independent of the exhaust flow rates chosen (Figure 4).
While the difference in room concentration between the two models investigated remains constant for all flow rates (Figure 4), the ventilation effectiveness of a local exhaust system increases with an increasing exhaust airflow (Figure 5). The ventilation effectiveness is proportional to: 
\[
\frac{1}{1-x}
\]
for \(0 \leq x < 1\), where \(x\) increases according to the ventilation rate. As a consequence the effectiveness starts at the value of one for \(x\) being zero and grows toward infinity for \(x\) approaching unity.

Similar to the ventilation effectiveness the potential reduction of air change rate, still guaranteeing the same room concentration as in the mixing case, also grows with growing flow rates (Figure 5). The reduction is proportional to: 
\[
\frac{q}{c + q}
\]
where \(q\) represents the ventilation rate and \(c\) is some constant. This function starts at zero and grows for growing ventilation rates asymptotically towards one, representing a hundred percent reduction.

![Figure 5: Ventilation effectiveness and potential ventilation rate reduction](image)

Figure 6 depicts the distribution of contaminants in the room resulting from the CFD simulation. The contaminant concentration is displayed in a yz-plane intersecting the model person and in a xz-plane intersecting the floor lamp. The contaminant level displayed in Figure 6 ranges from 380 to 700 ppm. This range has been chosen for the sake of a better clarity of the figure, although the effective maximum level is around 1000 ppm.

The carbon dioxide exhaled, rises in the thermal plume above the person up to the ceiling. Therefore, air with a high contaminant concentration relative to the room concentration is exhausted through the opening directly above the person. Beside the person, the presence of the computer and the floor lamp also provokes vertical air movement. Around these objects, air with a low contaminant concentration supplied at the floor level is lifted upwards. The air movement along the vertical axis of the lamp is supported by the cold downdraught forming along the cooled wall on the right side in Figure 6. This downdraught pulls some highly contaminated air along the ceiling to the right, down to the ground and to the left along the floor. As a consequence a weak air roll forms within the room. This behaviour of the flow is readily verified by the distribution of the contaminant concentration. The borders of the contours are slightly inclined to the horizontal axis in a clockwise direction. The result of this air movement is a slight mixing between the carbon dioxide and the room air.
Unlike the relatively high contaminant concentrations encountered in the thermal plume above the person, contaminant levels within the room are quite low. This is true at least for the 30 minutes that have been simulated. Highest contaminant concentrations in the room (around 500ppm) are encountered near the ceiling on the right side of the person. The contaminant distribution at the ceiling is displayed in Figure 7, using a range between 380 and 550 ppm. Once again, the transport of contaminants along the ceiling due to the air roll can be observed.

Within the range of the simulated time (30 min) the contaminant concentration in the room (monitored at three points, left, right and behind the model person, at a distance of 1 meter...
and a height of 1.6 meters above floor), remains at a very low level around 420 ppm. The contaminant concentration monitored at the location of the exhaust opening instead grows rapidly in time and reaches a value of about 850 ppm after 15 minutes. From this point on, the contaminant concentrations almost remain stationary at all locations being monitored (Figure 8).

![Figure 8: Developing of carbon dioxide mass fraction in time, monitored at exhaust opening (dark), in vicinity of the person, 1.6 m above floor (light)](image)

Results of carbon dioxide concentration in the room- and exhaust air have also been gained by measurements. The measured values were collected by a data logger and averaged over a period of a minute. The presence of the person in the room was limited to three hours per measurement and is identified in Figure 9 and Figure 10, by the two black bars. For three different ventilation rates (30, 90, 130 m\(^3/h\)), measurements have been carried out. In this paper only two sample measurements for a ventilation rate of 90 m\(^3/h\) are presented and discussed; one of the reference ventilation and another of the ventilation with local exhaust. The results from the measurements that have been excluded in this report show the same tendencies as the results presented here. The main difference between the various results is the evolution of the concentrations in time due to the different ventilation rates employed. The larger a ventilation rate is chosen, the faster of course a steady state is reached. From Figures 9 and 10 it can be seen that after about two and a half hours a nearly steady state is reached. The ordinate on the right side of the figures displays the concentration of the supply air. The levels vary strongly between one and another measurement since the air supplied to the test room originates from inside the building around the test room. In order to eliminate the unwished distortion of measured concentrations, the CO\(_2\) concentration of the supply air is subtracted from all of the other concentrations measured. These relative concentrations are shown in the ordinate on the left side of Figures 9 and 10. The different curves displayed in Figures 9 and 10 relate to the different sensor locations in the measuring facility. Highest levels are measured directly above the person at a height of 1.9 meters above floor while lowest carbon dioxide concentrations correspond to the sensor placed in the supply air. In the discussion of the results, the only concentration of interest relate to the sensors measuring concentrations within the room at a height of 1.4 meters above floor and within the exhaust air stream. The latter two concentrations are labelled in Figures 9 and 10 accordingly.
For the reference ventilation (Figure 9), there is hardly any difference in CO₂ concentration present between the room air and the exhaust air. The only difference observed already exists before the room is occupied and can therefore be explained by an offset originating from the measuring instruments.

In the case of ventilation with a local exhaust opening above the sitting person (Figure 10), there is a significant difference in carbon dioxide concentration of about 50 ppm between the room air and the exhaust air. Compared to the reference case, using a local exhaust leads to a slower rising of the room concentration on one hand and a faster increase of exhaust concentration in the beginning, on the other hand.

**DISCUSSION**

All of the methods presented in this paper (scalar model, CFD, measurements) confirm the increased effectiveness of a local exhaust system compared to traditional ventilation systems. Higher contaminant concentrations in the exhaust air lead to significantly lower concentrations in the room air. The CFD simulation and the measurements have shown that waste air can indeed be extracted with a high contaminant concentration relative to the average room concentration if the exhaust opening is placed properly.

The working principle of such a local exhaust system is readily understood considering the simple scalar model based on mass balances. For both, the one- and the two compartment model, the carbon dioxide concentration in the returned air becomes the same at steady state...
This must be the case since at steady state, the mass extracted from a volume cannot exceed the sum of the mass supplied to the volume and the mass produced by the source within that volume. The differences in carbon dioxide concentration between the exhaust air and the room air must therefore be based on the dynamic characteristics of the ventilation system. In the model example it is due to the fact that the sluggishness of the small volume directly above the person in the two compartment model is much smaller than the one of the single volume in the one compartment model. As a consequence the concentration in the small volume rises comparatively much faster, especially in the beginning and hence more contaminant is exhausted compared to the reference case (Figure 3). Although the exhaust concentration of the two compartment model and the reference become at steady state the same, integrated over time in the multi zone model, more mass of contaminant is removed and hence a low contaminant concentration in the room is the result. Reducing the sluggishness of the ventilation system is particularly interesting in a real-world application since more contaminant can be extracted in a short time period. This is important because people in office spaces often change their spatial position within time intervals much shorter than the time it takes the ventilation system to reach a steady state. For this reason a ventilation system with reduced sluggishness, i.e. a local exhaust system is favourable compared to a traditional ventilation system.

The scalar model further shows that in the case of local exhaust (2 compartment model) the ventilation effectiveness is not only above the one of a mixing ventilation but also rises with an increasing ventilation rate (Figure 5). The reason for this behaviour can be explained with a reduction of contaminated air to be spilled back into the room with increasing the exhaust airflow. For the limiting case that the exhaust flow rate matches the one of the convective stream, all the contaminants are removed from the domain and no mixing between contaminants and room air will take place. The implication of an increased ventilation effectiveness is the gain of a possible reduction of ventilation rate without degrading the indoor air quality in terms of carbon dioxide concentration. A reduction of the ventilation rate in turn leads to important energy savings. The fact that the ventilation effectiveness rises for increasing ventilation rates also has a beneficial effect on the ventilation strategy chosen for the local exhaust system, i.e. high exhaust capacity per square meter, which results in a proportionally high capacity per exhaust opening. As already mentioned in the introduction the low occupancy levels in office spaces and the local sensing of carbon dioxide allow the installation of an overcapacity of ventilation per square meter without redimensioning the central exhaust fan because only part of the exhaust openings will be active at a specific time.

A qualitative comparison of the results from CFD and measurements (Figures 8, 10) clearly shows the difference in the developing of contaminant concentrations with time. The build-up of the contaminant concentration in the returned air of the CFD results (Figure 8) is much larger than the one encountered in the measurements (Figure 10). As a consequence the opposite is true for the contaminant concentration in the room air. This discrepancy between the CFD results and the measurements must be explained with differences in the flow pattern encountered in the model room and the measuring facility respectively. Considering the measurements carried out for the reference ventilation (Figure 9) it is evident that the reference ventilation although set up as a displacement ventilation, rather works as a mixing ventilation because of the large convective streams and the cold downdraughts induced by the thermally active walls. There may still be some stratification of pollutants within the room but since the convective stream in the test room is much larger than the ventilation airflow the interface between the clean zone and the polluted zone will be close to the floor level. From the reference measurements one can conclude that the interface (if there is one) is even below 1.4 meters since the sensor measuring room concentration of carbon dioxide is located 1.4
meters above floor and displays a concentration similar to the exhaust concentration. In the simulated room a situation with a pattern similar to the one observed for displacement ventilation rather than for mixing ventilation is present. Unlike the situation in the measuring facility the thermally active wall in the simulated room exhibits only small undertemperature such that the resulting cold downdraught along the wall is very weak. As a consequence mixing occurs over short length scales rather than over the entire room height and hence a significant temperature stratification as well as a distinct layering of the contaminants in the room can be found. The layers of different contaminant concentrations are indeed slightly inclined against the horizontal axis due to the convective streams and the cold downdraught but in this case the driving forces needed to intermingle these layers are not strong enough.

In a situation with a flow pattern similar to the one encountered for displacement ventilation, the use of a local exhaust system leads to an increase of ventilation effectiveness when compared to a situation with a single, arbitrarily placed exhaust opening at the ceiling. The advantageous effect is of course largest if the contaminant source is very close to the local exhaust opening but it should always be guaranteed by arranging the exhaust openings in a fine grid at the ceiling. Is however a mixing situation predominant within the room, the potential increase of the ventilation effectiveness is comparatively larger since the effectiveness of a mixing ventilation per definition is always unity. Yet a mixing situation shows larger disturbances in the room, such that it is more difficult to capture contaminants at a high concentration at the exhaust location. However measurements and CFD-Simulation showed that irrespective of the flow pattern (mixed vs. stratified) encountered within the room the principle of local exhaust raises the effectiveness of the ventilation, given that the contaminant source is close enough to the exhaust opening.

As already mentioned before, does an increase of ventilation effectiveness lead to an increased potential of reducing the ventilation rate without reaching a higher carbon dioxide concentration in the occupied zone. As a consequence energy can be saved by reducing the ventilation rate. However, carbon dioxide is not the only aspect that has to be taken care of during the layout of a ventilation system. Other factors such as non-occupant generated contaminants and humidity in a building have to be controlled by ventilation as well. Especially in office premises where ventilation rates proposed by regulations and standards are relatively low, there is little scope to vary, i.e. to reduce the ventilation rate any further. Therefore energy saving might be small in office premises compared to other applications, such as classrooms, theatres or gates in an airport where high occupancy levels and a strong variation of those levels occur and therefore high ventilation rates are the default. While energy savings in office premises may not be as large as in other applications, a better indoor air quality is the major benefit from a local exhaust ventilation with an increased ventilation effectiveness.

Since this report was mainly concerned about the working-principle of local exhaust ventilation and the confirmation of its effectiveness under ideal conditions, it remains to show that a local exhaust system also works in practice where much larger disturbances are expected due to people walking around, opened windows, doors, etc. In particular it must be verified through field-tests in office spaces, whether high relative contaminant concentration in the exhaust air streams can still be provided, using a fine grid of openings rather than a single opening directly located above a source. More research is also required to be able to assess the true potential of this ventilation principle and to find the proper layout of the system. This includes technical (grid spacing, controller algorithm, etc.) as well as economical considerations (life cycle costs).
ACKNOWLEDGEMENTS

Thanks go to Amstein+Walthert AG for the provision of the simulation software and to EMPA Switzerland, where measurements have been carried out.

REFERENCES


5.4 Extended Research

5.4.1 Energy Saving Potential

The use of demand controlled ventilation offers a huge potential for energy savings. Thereby energy efficiency of the entire ventilation system can be significantly improved using this kind of a more advanced control. The amount of energy saved and the improvement in energy efficiency largely depends on the scenario chosen but savings up to 50% are possible as confirmed by [17]. Some of the most important parameters that determine the behavior of the system are: The type of building, the usage of the space, the occupancy schedule, the area specific ventilation rate, the ventilation effectiveness, the design of the plant and the quality of the equipment. According to [1] the average occupancy in large office buildings is around 76%. Relying on DCV therefore allows a theoretical reduction of the total ventilation rate by 24%. Further reductions in ventilation rate can be considered when making use of LEV. According to the results presented in the paper of section 5.3, potential flow rate reductions of 21\% considering the scalar model, 44\% considering the CFD simulation or 9\% considering the measurement is possible without a reduction of IAQ compared to complete mixing ventilation. Taking the results from the measurements this 9\% of further ventilation rate reduction superimposes with the 24\% reductions from DCV resulting in a 30.8 \% total reduction. These flow rate reductions leads to commensurate savings of energy due to lower ventilation losses including those for distribution and lower costs for conditioning of the air. Transmission losses over the building envelope and thermal ventilation losses determine the heat capacity required on the generation side. In older buildings without significant insulation of the envelope and ventilation rates as typically used for office spaces the ratio of ventilation to transmission losses takes a value of about 2. The better the thermal insulation of the envelope the larger this ratio gets. For an estimate of potential energy savings on the generation side a value of two is picked. Further, heat recovery with an efficiency of 80\% together with a heat pump is assumed. The energy savings are proportional to the change of flow rate weighted with the ratio of the ventilation to the total losses including heat recovery. The larger the share of ventilation losses on the total losses the larger of course is the impact of changes in the ventilation rate on the energy demand. With the numbers chosen, a reduction of the total energy demand of 8.8\% results for the 30.8\% flow rate reduction. In the case of using a heat pump this is directly the amount the electricity demand is reduced. Even more significant energy savings result for the distribution of the air when the ventilation is reduced and the cross sections of the
ducting is kept constant. Because the fan power scales with the third power of the air velocity in the ducts a 30.8% reduction in ventilation rates results in a 66.9% reduction of fan power and hence electricity. Expressed as a reduction factor this results in a value of 3.02.

These energy savings being expressed as a reduction of electricity usage show the potential for savings of high grade energy, i.e. exergy. Reductions in energy losses minimize external exergy losses that are directly proportional. A true exergy analysis however also allows to locate and quantify internal losses in a system. Even more parameters come into play when additionally the exergy performance of the system is considered. The degradation of high grade energy and its exergy efficiency can be analyzed to get a holistic view of the system. This exergy analysis for a general exhaust system is presented in the following.

5.4.2 Exergy Analysis

Looking at the exhaust side of an arbitrary ventilation system with balanced flow it can be synthesized from three components: A ducting network acting as a throttle, a fan and a heat exchanger for heat recovery as shown in Figure 5.2. The air is extracted from the room \( r \), the pressure in the duct network drops to reach pre-fan conditions \( p \). The exhaust air is accelerated and compressed when it reaches after-fan conditions \( a \). In the heat recovery unit heat is extracted and the air reaches conditions \( \text{out} \) of outgoing air before it is involved in a mixing process to finally reach its equilibrium state \( 0 \) with the environment. Based on the general description of an exhaust plant an

![Figure 5.2: A general exhaust system with its components.](image)

exergy analysis can be performed to identify the source of irreversible losses appearing in the system. Dealing with humid air, a mixture of dry air and water vapor has to be considered in the analysis. This fact results in the flow exergy incorporating contributions from physical and chemical exergy as it has been derived in Chapter 3. Moisture is added to the room by humans such that the moisture content or the specific humidity \( \omega \) of the room air differs
from the environment. This difference is considered and gives a contribution to chemical exergy. For simplicity the environment is assumed to be the dead state and the chemical potentials in the environment are neglected. This simplification allows to use a shorter form of the general flow exergy equation and does not alter the losses derived over the components but only affects the absolute value of exergy in the system. Over the individual components the moisture content of the air remains unchanged such that the chemical exergy remains unchanged too and is only transiting. The exergy loss of a component at steady state is calculated from the difference between the exergy inputs and the exergy outputs. This difference represents the losses, the exergy consumed due to internal irreversibilities.

Starting with the throttle the exergy consumed is calculated based on $E_{x,\text{in,throttle}}$ and $E_{x,\text{out,throttle}}$ as described by equations 5.1 and 5.2 respectively. Because exergy unlike energy is not conserved the exergy consumed is simply the difference of the incoming and the outgoing exergy flux. The throttle represents the ducting network with the assumption of an adiabatic flow and a constant kinetic energy. A work term representing the frictional loss that constitutes the pressure drop across the throttle is introduced. As a consequence there is a temperature drop across the throttle such that the temperature $T_p$ after the throttle is slightly below the room temperature $T_r$. For the pressure drop over the throttle a fixed value depending on the scenario considered is chosen. The room pressure $p_r$ is assumed to be the same as the ambient pressure of the environment $p_0$. As it can be seen from equations 5.1 and 5.2 the third term representing the contribution from the chemical exergy disappears when taking the difference between the incoming and the outgoing exergy. This is because the chemical exergy only transits the throttle but no chemical reactions changing the potentials is involved. The effect of the moisture onto the exergy consumed is found in the mass flow, the specific heats and the ideal gas constants that have changed.

$$\dot{E}_{x,\text{in,throttle}} =$$

$$\dot{m}_{\text{dryAir}} \left\{ \left( c_{p,a} + \omega c_{p,v} \right) \left[ (T_r - T_0) - T_0 \ln \left( \frac{T_r}{T_0} \right) \right] + \right.$$  

$$\left. (1 + \tilde{\omega}) R_a T_0 \ln \left( \frac{p_r}{p_0} \right) \right\}$$

\[ (5.1) \]
\[
R_a T_0 \left[ (1 + \tilde{\omega}) \ln \left( \frac{1 + \tilde{\omega}_0}{1 + \tilde{\omega}} \right) + \tilde{\omega} \ln \left( \frac{\tilde{\omega}}{\tilde{\omega}_0} \right) \right] \right] \\

\dot{E}_{x,\text{out,throttle}} =
\dot{m}_{\text{dryAir}} \left\{ (c_{p,a} + \omega c_{p,v}) \left[ (T_p - T_0) - T_0 \ln \left( \frac{T_p}{T_0} \right) \right] + \\
(1 + \tilde{\omega}) R_a T_0 \ln \left( \frac{p_p}{p_0} \right) + \\
R_a T_0 \left[ (1 + \tilde{\omega}) \ln \left( \frac{1 + \tilde{\omega}_0}{1 + \tilde{\omega}} \right) + \tilde{\omega} \ln \left( \frac{\tilde{\omega}}{\tilde{\omega}_0} \right) \right] \right\} 
\]

Looking at the fan in the exhaust plant, its task is to compensate the pressure drop and to transport air through the system at a given rate. The pressure drop and the flow rate together determine the power required by the fan to fulfill the task of ventilation. Including a total efficiency of the fan the electrical power usage is calculated in eq. 5.3.

\[
W_{el,fan} = \frac{\dot{V} \Delta p}{\eta_{\text{tot,fan}}} 
\]

The pressure change across the fan is the same but of opposite sign as the one assumed over the throttle. As a consequence the pressure after the fan \( p_a \) is again the same as the room pressure \( p_r \). Concerning the temperatures a temperature rise across the fan from \( T_p \) to \( T_a \) occurs given an adiabatic fan. Assuming the axial component of the velocity contributing to the kinetic energy to be constant and the radial component to be fully dissipated into heat the temperature rise is simply calculated according to eq. 5.4.

\[
T_a = T_p + \frac{|\dot{W}_{el,fan}|}{(c_{p,a} \dot{m}_a)} 
\]

The equations for the exergy consumed across the fan take the same form as for the throttle with the single exception that the \( E_{x,\text{in,fan}} \) additionally includes the work term \( |\dot{W}_{el,fan}| \).

The heat exchanger of the heat recovery unit is the interface between the air leaving the building and a water cycle that connects to the heat pump on the evaporation side. The temperature at which the heat transfer from the air
to the water takes place is hence a temperature close to the evaporation temperature of the heat pump. Eventual condensation of the humid air at the heat exchanger has been ignored in the present analysis. For the heat exchanger an efficiency $\eta_{HE}$ has been defined which expresses the percentage of the total heat that the heat exchanger is capable to recover from the air stream. Based on a simple energy balance over the heat exchanger the temperature $T_{out}$ of the leaving air stream is determined. This temperature expressed as a function of the incoming air temperature $T_{in}$ and the environmental temperature $T_0$ is shown in eq. 5.5.

$$T_{out} = \eta_{HE} T_0 + (1 - \eta_{HE}) T_{in}$$  \hspace{1cm} (5.5)

The air pressure across the heat exchanger is assumed to be constant. This is because in the model all pressure losses occurring in the exhaust plant are concentrated to the throttle. The calculated temperature $T_{out}$ is used to determine the heat flux recovered. Based on this heat flux the exergy flux recovered is calculated using the Carnot factor as expressed in eq. 5.6. The Carnot factor expresses the maximum work potential of the heat flux flowing between two reservoirs of different temperatures. The reservoir temperatures are assumed to be constant. Because the temperature $T_{HE}$ at which the heat transfer takes place is related to the evaporation temperature of the heat pump and is assumed to be constant, this calculation method using the Carnot factor is appropriate.

$$\dot{E}_{x,HR} = \left(1 - \frac{T_0}{T_{HR}}\right) \dot{Q}_{HR}$$  \hspace{1cm} (5.6)

For the exergy fluxes $E_{x,in,HE}$ and $E_{x,out,HE}$ of the air through the heat exchanger the same formulas as the ones used for the throttle can be applied. The exergy flux from the heat recovery is a flux leaving the control volume around the heat exchanger. As a consequence, the exergy consumed in the heat exchanger is calculated by subtracting both, the flow exergy of the leaving air and the exergy recovered from the flow exergy of the incoming air stream. Depending on the efficiency of the heat recovery the air leaving the building has a temperature that is above the environmental temperature $T_0$ and hence still contains some exergy. This remaining exergy will be destructed through a mixing process that takes place until an equilibrium with the environment is reached. In this mixing process all the physical but also the chemical exergy with reference to the environment is set to zero. The exergy consumed in this process is simply the exergy leaving the building at the stage out.
Evaluation of Exergy Losses

The exergy losses in a generic exhaust plant as previously introduced have been evaluated for two different scenarios, one during heating and the other during cooling operation. The heating and the cooling scenarios differ mainly in the assumptions of the climatic conditions made. In the heating mode additionally a heat recovery unit is modeled. For both scenarios, heating and cooling, an exhaust plant with a ventilation rate of 8000 m³/h and a system pressure drop of 400 Pa is chosen, where the exhaust fan is assumed to have an over-all efficiency of 70%.

For the heating mode environmental temperature was set to 0 degree Celsius and room temperature is at 24 degree Celsius. The relative humidity is taken to be a 100 % in the outside air and 60% in the room. This assumes that moisture is added in the inside by humans and ev. plants. The heat recovery during heating operation has an efficiency of 80% and the heat exchange between the air and the evaporation side of the heat pump is defined to take place at 10 degree Celsius. The exergy losses in the exhaust plant during heating season are shown in Figure 5.3. For the given boundary conditions

![Figure 5.3: Exergy losses across the stages of an exhaust system for the heating case](image)

exergy destruction is most significant over the throttle with 894 W and in the heat exchanger with 761 W. Losses in the fan and those due to mixing are 344 W and 113 W respectively. The partitioning of the the exergy losses among
the different stages can be appreciated in Figure 5.4 In the throttle the entire work potential coupled to the pressure drop over the throttle is lost because of friction. In the fan work is needed to build up the pressure by the same amount it has been reduced over the throttle. What is lost in the fan is mostly due to the fan efficiency used to express the losses between the electricity input and the fan output. The exergy losses that occur in the heat exchanger during heat recovery are influenced by the heat recovery efficiency, the average air temperature between the inlet and the outlet of the heat exchanger and the evaporation temperature of the heat pump. The efficiency determines the size of the heat flux leaving the system which contributes to the so called external losses. The larger the recovered heat the smaller the exergy lost to the environment. The temperatures involved in the heat transfer determine the entropy production and contribute to the internal exergy losses. These losses are minimized for small temperature differences between the media that exchange heat with each other. The exergy flux that finally leaves the heat exchanger and the building mixes with the outside air. In the pass of mixing entropy is generated and the exergy content of the stream decays to zero when it has reached environmental conditions.

Figure 5.4: Distribution of exergy losses among different stages for the heating case
In the case of cooling, warm air from the outside is brought with 28 degree Celsius and a relative humidity of 40% into the room. Room temperature is set to 26 degree Celsius and the relative humidity in the room is assumed to be 60%. During cooling mode the heat recovery unit is simply bypassed. The exergy losses that result for the exhaust plant during cooling season are shown in Figure 5.5. Again, most significant losses for the given boundary conditions with 893 W occur in the throttle where potential energy from the pressure is dissipated. It is of the same magnitude as the exergy loss occurring during the heating period and is independent of the reference temperature. The heat exchanger does not contribute any losses. The fan losses with 378 W differ only slightly from the ones calculated for the heating mode. The differences are due to different moisture levels and temperatures which influence the density of the air handled. The mixing losses during cooling season are with 15 W almost one order of magnitude smaller than those recorded during heating. This is simply due to the smaller temperature difference between the room and the outside in summer compared to the winter. Depending on the temperatures, in summer the room air contains little cool exergy where in winter it usually contains a larger amount of warm exergy. The partitioning of the exergy losses among the different stages in the exhaust plant is shown in Figure 5.6.

![Exergy losses across the stages of an exhaust system for the cooling case](image-url)
Figure 5.6: Distribution of exergy losses among different stages for the cooling case
Bibliography


Chapter 6

Decentralized Supply

6.1 Concept And System Description

In this chapter an air distribution system (ADS) for decentralized supply is analyzed. The system consists of a large number of decentralized supply units arranged along the periphery of the building that feed a strongly interlaced ducting network to distribute the air in the space. The ducting network can be integrated in the floor construction between the steel reinforcements of in-situ concrete without any additional space requirements compared to a regular construction. The middle zone of a concrete plate is known as the neutral zone in terms of stresses. It is where the transition from positive tensile stress at the lower to the negative stress at the upper side of the plate takes place and some void can be introduced without affecting the strength of the construction. Beside the transport of air the ducting can also be used to carry cables from supply points being located at the facade to working places distant from the facade. The ductwork forms a network and connects neighboring as well as the supply units from opposite sides of the building. The networking between air outlets in the space allows to balance out spatially variable loads that might occur for a system with LEV as described in the previous chapter. Similarly to the presented LEV where the allocation of exhaust power is determined by the distributed, sensor-controlled extraction points an ADS based on a ducting network allows for a flexible allocation of supply flow rate by a simple adjustment of the pressure equilibrium in the system. The networking between the different supply units distributed around the building allows the supply system to run in a hybrid mode and to make use of pressure differences
occurring across the building because of the influence of wind. The hybrid operation offers the possibility to save electrical power needed otherwise to transport the air through the building.

The design of such an ADS is based on the analysis of the air flows in the network evaluated through simulations. Several different networks topologies were considered to identify important performance parameters. Based on the results of the simulations energy and exergy savings were estimated and a comparison to purely centralized supply systems was made.

6.1.1 Motivation

In contrast to a centralized supply approach where the air is conditioned and handled by a single unit and then distributed over the entire building in decentralized systems the air is treated locally, close to the point at which it is released to the room. Typically a large number of decentralized supply units is needed to substitute a centralized plant. Because of the small dimension of decentralized equipment an integration of the technology in the building structure is possible whereas for centralized plants large service rooms are required that reduce the rentable space. The decentralized supply units are typically arranged along the periphery of the building where the air is directly released. In an open plan situation the air can evenly distribute over the entire floor space. Assuming the ventilation to follow a displacement flow pattern the fresh, cool air supplied at the facade spreads along the floor and is transported towards the core of the building if air is extracted from there. As the cool air moves towards the inside of the building it heats up and eventually never reaches the core of the building depending on the depth of the space. In such a case the age of the air in the core is likely to be higher than in zones that are located close to the windows. This is because of an uneven distribution of fresh air in the space. Ideally, in a sufficiently small open space the supply of fresh air from the periphery leads to a good IAQ in the entire space. The cool, supplied air does not heat up too much such that it reaches the core. The equilibrium layer with the cool air is similar to a lake where water is supplied and withdrawn through sources and sinks but without inducing significant flow in the lake itself. What is removed from the equilibrium layer by the convective currents in the room is replaced by the supply units at the periphery. Because of the large number of sources the individual flows in the equilibrium layer are low and hence static pressure distribution almost homogeneous. This situation allows for a very regular distribution of fresh air even in situations with a locally increased demand as they might arise from a system with LEV. If the space is too large to allow the "lake" of fresh air to form, fresh air has to
be brought directly to the core instead of transporting it through the room. One possibility to directly supply fresh air to the core is by using a ducting network such as suggested here. The ADS to be designed should follow the ideal of the “lake” of fresh air and approximate its characteristics as closely as possible.

Another distribution problem can occur in decentralized supply systems under the impact of wind, mainly if the space is partitioned. Under such conditions it can happen that due to locally acting negative relative pressures on the facade a given decentralized supply unit is not able to maintain the design flow rate and less air is supplied to the room. If the exhaust flow rate is adjusted to the supply, the room does not get enough fresh air no more. If the exhaust is not adjusted to the supply, a mismatch in flow rates results and the same negative pressure as acting outside is now present in the room. Assuming that the surrounding space is sufficiently supplied with fresh air and is at neutral pressure, there is a pressure difference between the office and the surrounding space. Depending on the strength of the pressure difference it will be difficult to open doors or the risk for flow induced noise due to the balancing flow occurring at the doors increases. Using a network based ADS would allow for a balancing of pressure differences between spatially separated zones and eliminate the pressure induced problems mentioned. Further a network enables not only to balance pressure in the space but also to make use of the pressure difference acting across the building as a driver for air distribution.

Both problems, the one of insufficient air supply to the core and the one of pressure differences occurring between separated rooms are addressed and tried to be solved by the approach described herein.

6.2 State of the Art Review

This section presents a comparison of decentralized and centralized air handling with its advantages and disadvantages as described in literature and field reports. Further it includes a discussion of transport networks, what the characteristics of such networks are and what kinds are commonly used for air distribution in HVAC or in other fields. Common approaches for air distribution are reviewed and compared to the approach analyzed here.

Mechanical ventilation is very common for office buildings and necessary for IAQ - and sometimes temperature control. While in older dwellings built before 1970 ventilation through leaks in the facade was common there is a trend towards controlled mechanical ventilation for dwellings too. The reasons are new regulations for the use of primary energy and for green gas emissions as
well as the pressure on the energy prices. The large and uncontrolled loss of energy through leaks in the envelope is avoided in new building by using a more airtight construction. At the same time this makes it necessary to actively ventilate to control the IAQ and moisture in the building. In newly built houses moisture problems due to the combination of an airtight construction and insufficient ventilation are quite common \cite{24}, \cite{6}. Controlled, mechanical ventilation is a suitable mean to guarantee a sufficient IAQ and much more predictable in terms of energy loss compared to classical window ventilation. If no ventilation installed, an airtight building envelope makes it necessary to manually ensure a sufficient air exchange by window opening. A continuous aeration with a tilted window is possible but not recommended. The air change with a tilted window is hard to control and may be too high ranging from 1 to 4 energetic air changes per hour for opening angles from 6 to 15 degrees given a temperature difference between the room and the outside of 20 degrees \cite{26}.

### 6.2.1 Decentralized vs. Centralized Air Handling

In office buildings new ways of operation are exploited to reach a reduction in the primary energy usage. One trend is the separation of the heating/cooling from ventilation \cite{7}. This is important because it increases the degree of freedom for the design and offers the possibility to control the ventilation rates independently from the heating or cooling requirements. Similarly the dehumidification is more and more decoupled from the cooling process to allow for a better thermodynamic performance of the machinery resulting in a lower energy usage. Another important trend concerning the building operation is the use of decentralized ventilation concepts. This is true not only for office buildings but also for dwellings. Although many engineers still look at decentralized equipment with a certain skepticism it can be considered as a valid alternative to centralized HVAC systems \cite{20}, \cite{25}, \cite{18}, \cite{37}. One of the largest benefits from decentralized equipment is its flexibility which is a key factor in todays office spaces \cite{7}. Changing tenants, reorganization of the space or local demand control require a large degree of freedom in terms of the management of technical installations. Another very important advantage especially in large, multi-story buildings is the slimness of the equipment. The construction height for a given room height using decentralized systems can be reduced such that approximately every 10 stories 1 extra story could be added given the total construction height \cite{18}. In terms of rentability of a building this means an entire, additional lettable story and consequently a 10 percent higher income. Since no or little ducting is required in a decentralized ventilation system planning expenses as well as construction time and costs
go down. The easy integration of decentralized equipment into the building structure without any need for complex supply structures favors this approach also in the retrofit of buildings. The space efficiency in buildings with decentralized equipment is higher than for buildings relying on centralized plants. The study and comparison of several buildings with decentralized and centralized technology presented by [18] clearly showed a 5 to 8 percent lower space requirement of the decentralized equipment. Depending on the building and the design of the plant this efficiency gain could be further increased. In terms of investment decentralized equipment is slightly more costly than centralized one [18], [28]. From an energy point of view the decentralized air handling with its short transportation distances leads to a 3 to 4 times smaller specific fan power (SFP) [28]. The lower electricity consumption of decentralized plants is also confirmed by [18]. The electricity consumption in buildings combining decentralized AHUs together with electrical heat pumps shows to be of the same magnitude as the electricity consumption in centrally operated buildings that use fossil fuels for heat generation. Most of the decentralized AHUs come with heat recovery incorporated. The heat from the return air can effectively recovered by the small, decentralized unit [20] but typically it is less efficient than the heat recovery in large, centralized units. Critical for the performance of the heat recovery units in general is the internal leakage that can occur. In decentralized AHUs these leakages can substantially lower the performance of the heat recovery and must be considered in the evaluation of the effective energetic performance of these devices [19],[29],[12]. According to [8] and [29] the decentralized heat recovery is not necessarily economically reasonable. From a comfort perspective decentralized ventilation concepts are almost equal to centralized ones. This was also confirmed in the survey presented by [18] which shows a good acceptance of decentralized systems by the users. A user complaint that was recorded however concerned the noise level and dry air in winter time related to decentralized air handling. Decentralized humidification is economically questionable and can lead to hygienic problems such that it is hardly implemented. In future, stand-alone applications for decentralized, adiabatic humidification based on the principle of atomization will be on the market such that this drawback can be weakened. The problem of the sound emission is related to the compact construction of the air handling units and their installation close to the user. This problem is pointed out as one of the most critical aspects of decentralized equipment by [18] and [20] and an improvement in that area is required. In Table 6.2.1 the pros and cons of decentralized and centralized air handling as typically found in literature are summarized.

There are many pros and cons for either technology. Depending on the re-
Table 6.1: Typical pros and cons of decentralized and centralized air handling

<table>
<thead>
<tr>
<th>Pros of centralized air handling</th>
<th>Cons of centralized air handling</th>
</tr>
</thead>
<tbody>
<tr>
<td>- efficient and economical heat recovery</td>
<td>- large space requirements for machinery and ductwork</td>
</tr>
<tr>
<td>- efficient fans available</td>
<td>- large construction and investment costs</td>
</tr>
<tr>
<td>- complete climate control possible</td>
<td>- planning expenses for ductwork</td>
</tr>
<tr>
<td>- good de-/humidification possibilities</td>
<td>- cleaning of ductwork</td>
</tr>
<tr>
<td>- easy function check (single unit)</td>
<td></td>
</tr>
<tr>
<td>- lower maintenance costs</td>
<td></td>
</tr>
<tr>
<td>- lower floor height (lower construction costs for fixed room height)</td>
<td>- ev. high noise level of fans</td>
</tr>
<tr>
<td>- larger space efficiency because of lacking machinery room</td>
<td>- no economically reasonable de-/humidification</td>
</tr>
<tr>
<td>- allows for flexible use of space (reconfiguration)</td>
<td>- only thermal conditioning, no economical full climate control</td>
</tr>
<tr>
<td>- integration in building structure possible (floor, facade, etc.)</td>
<td>- good air quality mainly restricted to window near areas</td>
</tr>
<tr>
<td>- easy installation during retrofit</td>
<td>- ev. lower ventilation efficiency through short-circuiting</td>
</tr>
<tr>
<td>- no or little ducting</td>
<td>- not suitable for mixing ventilation</td>
</tr>
<tr>
<td>- low pressure drop for air distribution</td>
<td>- more frequent maintenance necessary</td>
</tr>
<tr>
<td>- lower electricity consumption (SFP)</td>
<td>- higher total maintenance costs</td>
</tr>
<tr>
<td>- large flexibility for local control</td>
<td>- accessibility to floor space for maintenance required (rel. for dwellings)</td>
</tr>
<tr>
<td>- easy demand control</td>
<td>- influence of wind onto performance</td>
</tr>
<tr>
<td>- easy filter change, no expert needed</td>
<td></td>
</tr>
<tr>
<td>- savings in investment and operation costs possible</td>
<td></td>
</tr>
<tr>
<td>- easy to satisfy fire protection requirements</td>
<td></td>
</tr>
<tr>
<td>- thermal activation possible (no suspended ceiling)</td>
<td></td>
</tr>
</tbody>
</table>

Requirements it might be more favorable to choose one or another approach. At the same time there are some clear indisputable advantages of the decentralized technology such as the decreased complexity and its resulting space and cost savings that make it very interesting to use. The use of a heat pump for heat generation on the other hand makes it obvious to implement a centralized heat recovery which already is more efficient than a decentralized heat recovery. As a consequence the combination of decentralized and centralized technology also allows to combine some of the individual strengths of the different approaches and lead to a very slim and efficient over-all system. Unlike the fully decentralized systems that rely on AHUs with combined supply/exhaust and heat recovery the hybrid systems make use of decentralized supply with pure
supply units and a centralized exhaust with heat pump based heat recovery. In dwellings sometimes fully passive or actively controlled supply inlets are used instead of fan units. A large number of different designs from pressure to humidity to temperature controlled supply elements are available [31]. In office spaces but also in dwellings active fan coil units are predominantly used to supply and condition outside air. Some of these units can be run in circulation mode to convectively heat up the room air. Many large projects for office buildings especially in Germany have been realized using this combination of active decentralized supply over the facade together with a centralized exhaust that is coupled with a heat pump. The most prominent and largest project of that kind is the Post tower in Bonn, Germany.

6.2.2 Air Distribution and Transport Networks

Decentralized supply is usually based on the direct release of fresh air to the room at the place where the AHU is installed. If still air has to be brought directly to the core, raised floor systems are most commonly used. These raised floor constructions can either be designed as a pressurized plenum with direct exhaust of air to the room or as a simple non-pressurized cavity, hosting ducting for air distribution and others such as electrical installations. Although possible in combination with decentralized AHUs the air distribution over the floor space is more often used in combination with centralized air handling. While the floor based air supply is no novelty for Europe it is a rather young trend in the U.S.. The so called "underfloor air distribution systems" (UFAD) [35] are a serious alternative to the predominantly used "overhead systems" that introduce the air into the room close to the ceiling with large velocities. The UFAD systems bring the improvements that are typical for displacement ventilation in contrast to mixing ventilation.

The construction height of raised floors can vary between 15 to 60 cm. Rather in the upper but similar range are construction heights for suspended ceilings used for overhead systems. This suspended or raised parts lead to a large construction height of a story for a given ceiling height in the room. The increased construction height directly influence the construction costs in that they go up. Moreover does the coverage of the floor and/or the ceiling decouple the solid construction from the room air. This has the consequence that the thermal mass of the building cannot be activated and be used for climate control. From an installation point of view the raised floor or the suspended ceiling offer a large flexibility. Whether the full flexibility of the raised floor is required in a building or not depends largely on the type and the usage of the building. Through the use of new bus technologies in the building automation
the path lengths and hence the amount of cables required could be reduced such that the flexibility offered by a raised floor is not necessarily required no more [17]. An approach with underfloor tubes integrated in the construction and cavities in the floor along the facade to carry the electrical installations can be an alternative. A large number of commercial systems for the integration of electrical installations into the floor pavement are available. In Germany where typically the thickness of the floor pavement exceeds the thicknesses used in Switzerland, ducts for the ventilation can also be integrated. Alternatively to the air distribution in the floor pavement or in a raised floor the ductwork can directly be integrated into the actual floor construction between the reinforcement steel. The integration of most types of installations into the construction is usually possible but depends on the type of building and construction [13]. Cavities for installations are sometimes available in prefab construction elements or can be created when using a construction with in-situ concrete. The air distribution system considered in this work also makes use of the possibility of a direct integration into the floor construction. Beside the air transportation it can be used to carry cables from communication and electrical installation. As explained previously the air distribution shall follow the ideal of the "lake" of fresh air and hence show a similar characteristic. Typical transport and distribution networks in buildings exhibit a hierarchical structure showing a branching network topology. Similarly for other technical applications, e.g. gas pipelines, electrical power transmission or in nature, e.g. a tree, veins of a leaf, supply of organs, hierarchical, branching networks are encountered. A very rare exception to this in nature is found in the supply of the brain of mammals. A circle of arteries, the so called cerebral arterial circle also known as the circle of Willis is a closed loop with several supply branches shown in Figure 6.1. This loop with its multiple connections introduces redundancy in the supply of the most important organ. If the path along one branch is obstructed there is still another path sufficient to preserve the cerebral perfusion. In fact, the reason for most transport systems having the same hierarchical supply structure is because they are all single-supply systems. There is one single source location that has to feed multiple sinks. In a multi-supply system closed loops are more natural to appear. Several branching networks that join each other automatically form closed loops. A technical example of a looped network is found in traffic systems. A network of streets or a railway networks with several crossings between the different paths and the bidirectional connections of multiple destinations leads to a complex mesh as shown in the example of Figure 6.2.

Looking at the available air distribution systems, it appears, that they are all designed as single-supply systems. As a consequence they use branching
topologies for the distribution of air. This is true for the design of supply systems in office spaces as well as for the complete supply solutions available on the market for mechanical ventilation of dwellings. These ventilation systems for dwellings include a set of standard components such as AHU, tubes, fittings that can be easily combined to a complete system. In office spaces the supply and distribution systems are customized and no standard system solutions are typically available. An exception to this is an available ADS system presented by [23] which offers standardized, Plug n’ Play components that are arranged in a mesh topology. Although using a mesh topology for the air distribution this system is still a single-supply system. The air is fed into the network at a single spot and hence the air distribution has still a similar characteristic as in common branching networks. On the other hand the interconnection between several branches does help to balance out pressures and allow for a better, more regular distribution of the supply air.

Figure 6.1: A closed loop formed by a circle of arteries, the so called cerebral arterial circle also known as the circle of Willis increasing redundancy in the vital supply of the brain. Source: A.D.A.M. Inc.
Figure 6.2: An excerpt from the subway network of Moscow, Russia. Similar to a multi-supply system of a building the brown circle line can be seen as the building envelope with various supplying lines coming from outside. Inside the circle the lines cross each other such that a closed-looped network is created with multiple possibilities to travel from one point to another.

6.3 Problems Addressed

The following problems concerning the supply of air to the space are addressed in this work:

- Air flow in looped ducting network: Typical piping or ducting systems are hierarchical, branching networks. Many codes are available for calculation of branch flows. Most programs originate from hydraulics, treating incompressible media. A tailored model for simulation of looped network with several components as junctions, openings and pumps was put together and implemented.

- Analysis of topologies: It is looked systematically at different topologies that are able to lead to a uniform flow rate distribution even under influence of wind. Selections of networks are presented in two papers.
• Energy savings of decentralized vs. centralized supply: Based on calculated pressure drops for a selected topology of a decentralized system a comparison with a typical centralized system was made. A pressure drop estimation for the centralized fans was based on a comparison of commercial fans. The comparison is presented in the AIVC paper.

• Theoretical energy and exergy savings for hybrid operation mode: Using the concept of fan overcapacity an estimate of theoretical energy and exergy savings were analysed for a given wind pressure distribution. These results are presented in the PLEA paper.

• Possible control strategies and scenarios: Based on the behaviour of the fans to changing inside and outside pressures strategies for the control of the decentralized supply is developed.

6.4 Theory And Methods Used

This section provides the theoretical background and the methods used for the analysis of the decentralized supply.

6.4.1 Air Flow Simulation for Looped Duct Networks

In order to understand the behavior of the decentralized supply system with its distribution system, the routing of the air flow has to be known. The airflow distribution depends upon a large number of parameters such as the topology of the network, the flow resistance of each component, pressure conditions in the room as well as outside the building and the flow characteristics of the fans in use. The properties of a given network can hence not be simply inferred from its setup and have to be calculated considering the influence of all parameters simultaneously. A simulation of the flow in the network gives all the information needed to judge the quality of the network in terms of distribution.

A calculation method that enables to evaluate the flow rate distribution in an arbitrarily interlaced ducting network was put together. This included the selection of a simplified mathematical model to describe the flow behavior in different network components such as pipes, junctions, openings and pumps/fans as well as the algorithm used to solve for the large number of unknowns introduced in a ducting network. Finally the implementation of the mathematical model into a software model was realized.
Object Oriented Modeling and Implementation

Any real world system consists of an arbitrary number of elements that make up the system. The characteristics of the system is given by the single elements and by their relation to each other. Systems can be classified in different ways, e.g. by its functions or by its components. Considering a car, one could simply list all the components and arrange them in a hierarchical way. Following along one branch, a car consists of wheels whereof four are regular wheels and one is a spare wheel. Every wheel consists of a rim, a tube and a tire. This approach to order a car by its components is very natural and intuitively understandable. Moreover it allows to represents the car in a very structured way. This structure then helps to implement the real-world object into a programming language. Unlike procedural programming languages does the object-oriented approach lead to a very close relationship between the real world and the programmed world. Any real world object has an equivalent in an object-oriented language with its real-world attributes and behaviors/functions. This close analogy allows for a very convenient implementation of a real-world system into an object-oriented programming language. The task of structuring and reducing the complexity of a system for implementation into an object-oriented language is known as object-oriented modeling (OOM) [30]. This modeling guideline allows for an effective implementation of very complex systems. Concerning the supply network modeled in this work only two aspects of OOM that have implications on the implementation where considered. The first aspect is the ”is a” relation and the second the ”has a” relation. Recalling the above example, the spare wheel ”is a” wheel, meaning that it has the general properties of a regular wheel. This concept is known as ”inheritance” [21] and is part of object-oriented programming. The ”has a” relation is also known as ”aggregation” [30] and describes the hierarchy among components. For the previous example, the wheel has a rim, a tube and a tire, meaning they are part of the wheel, i.e. its attributes. The programming implications are such that within a ”class”, e.g. wheel, the contained components, e.g. rims, etc. are ”instantiated”. This means that whenever a wheel object is created also a rim, a tube and a tyre are created. In the supply network represented by Figure 6.3 the inheritance is displayed using a triangle, while aggregation is denoted using a diamond.

Software Model of the Ducting Network Located in a Building

The actual components comprising the ducting networks are pipes and nodes. A simple node is a junction while opening and pump are specializations of the
Figure 6.3: Structure of ducting network according to OOM for implementation in a object-oriented programming language

basic node. According to Figure 6.3 every pipe is derived from a geometrically simple line element and has two terminal components of the type node. Whether these nodes are junctions, openings or pumps has to be specified. The nodes are placed somewhere in a room which itself is defined as a polygon element that has a indefinite number of walls. A wall similarly to the pipe is derived from a simple line element and also has two terminal components but of the type boundary node (Bnode). A wall can optionally have a pump. This means that a pump can be installed within a wall that has access to the outside and because of its nodal characteristics it is also connected to a room.

The components implemented into an object-oriented language have similarly to objects in real world some attributes and methods or functionality. The object can not only passively hold values but can process these values and exchange information with other objects. A pipe object for instance actively calculates its own mass flow from known pressure values at its ends. A node object calculates the flow function which is the net between incoming and outgoing mass flows from the connected pipes. Distributing tasks among software objects leads to logical structure of the code that is easy to understand and to maintain. Instead of having one huge unit of procedural code, a well structured code comprised of many subunits is obtained.
Mathematical Model

The flow in the ducting network is described using a 1D mathematical formulation. This 1D approximation is appropriate for the information required and allows for an efficient calculation of the air distribution avoiding unnecessary complexity. In reality the flow in a piping network with internals such as the different types of nodes in use is 3-dimensional, fully turbulent and time variant. To consider the full complexity of this flow is computationally extremely expensive and not required to solve the problem of simple air distribution.

This section covers the modeling of friction losses in the flow elements pipes, junctions, openings, pumps and presents the algorithm used to solve for the unknowns.

Pipe Friction:  Friction in pipes is relevant in many engineering applications and hence standardized methods for treating these problems are available and treated in many text books such as [39]. Based on empirical observations the Darcy-Weisbach equation (eq. 6.1) describes the head loss \( h_l \) in a pipe as a function of the friction at the pipe walls, the average kinetic energy of the flow and the length to diameter ratio.

\[
h_l = f \frac{\bar{u}^2 L}{2gD} \quad (6.1)
\]

The \( f \) appearing in eq. 6.1 is the dimensionless Darcy friction factor. It is defined in terms of the shear stress at the wall of the pipe, \( \tau_w \), by eq. 6.2.

\[
f = \frac{\tau_w}{\frac{1}{2} \rho \bar{u}^2} \quad (6.2)
\]

For perfectly smooth pipes the friction factor is only a function of the Reynolds number and hence it depends on whether the flow is laminar or turbulent. The expressions for laminar and turbulent flow respectively are:

\[
f_{\text{laminar}} = \frac{64}{Re_D} \quad (6.3)
\]

\[
f_{\text{turbulent}} = \frac{2.51}{Re_D \sqrt{f}}
\]

In practical applications the pipes have a certain roughness and the friction factor becomes a function of both the so called sand-grain roughness \( k_s \) of
the pipe and the Reynolds number of the flow. An expression describing this functional relation is given by eq. 6.4 known as the Colebrook formula.

$$\frac{1}{\sqrt{f}} = -2\log_{10}\left(\frac{k_s/D}{3.7} + \frac{2.51}{ReD\sqrt{f}}\right)$$ \hspace{1cm} (6.4)

A graphical solution of eq. 6.4 for a set of different roughness heights and different Reynolds number is available and known as the Moody diagram, see e.g. [39]. Various approximations that allow an explicit evaluation of the friction factor have been developed such as the Haaland or the Swamee-Jain equation [11], [34].

The losses discussed for straight pipes of constant cross-section are referred to as major losses. In ducting networks not only losses in straight sections but also losses occurring in fittings or at inlets have to be considered. Losses within these elements are termed minor losses. In regular distribution systems friction losses in the straight pipe sections are more relevant and hence the major contributor to the over-all losses. In looped networks with many fittings however minor losses can become quite significant. The total head loss in a complex pipe system is calculated according to eq. 6.5 respecting both, major and minor losses.

$$h_L = \sum_i h_{L_i}^{major} + \sum_i h_{L_i}^{minor}$$ \hspace{1cm} (6.5)

The head loss $h_{L_i}^{major}$ is calculated using eq. 6.1. The minor losses are calculated using eq. 6.6

$$h_{L_i}^{minor} = K \frac{\bar{u}^2}{2g}$$ \hspace{1cm} (6.6)

The loss factor $K$ is usually empirically determined by measuring velocities and pressures in a particular setup. Beside some special cases like the sudden contraction or expansion where the loss factor can be approximated by a simple mathematical expression, it is a scalar value that has to be picked from a database of measured values available in many text books such as [2],[39], etc. The loss factors for a sudden contraction and a sudden expansion are given by eq. 6.7 and 6.8 respectively.

$$K \approx \frac{1}{2} \left[ 1 - \left(\frac{D_2}{D_1}\right)^2 \right]$$ \hspace{1cm} (6.7)

$$K \approx \left[ 1 - \left(\frac{D_1}{D_2}\right)^2 \right]$$ \hspace{1cm} (6.8)
Recalling eq. 6.5 and using the definitions from eqs. 6.1 and 6.6 the total head loss in a pipe system with a total pipe length $L$ of constant cross section can be expressed as

$$h_L = \frac{\bar{u}^2}{2g} \left[ \frac{fL}{D_h} + \sum_i K_i \right]$$  \hspace{1cm} (6.9)

The $D_h$ in eq. 6.9 is the hydraulic diameter which is a conceptual extension of the previously used diameter $D$ for circular pipe cross-sections. It is defined so that it is equal to the physical diameter for a circular cross section. It can be used to approximate noncircular geometries if they do not differ too much from a real circle. The formal definition for $D_h$ is

$$D_h = \frac{4A}{P}$$  \hspace{1cm} (6.10)

where $A$ is the cross section and $P$ is the perimeter of all surfaces exposed to the fluid. All the expressions for the head losses derived previously can be applied as well to noncircular cross sections using now the hydraulic diameter $D_h$.

The head loss calculated in eq. 6.9 has the dimension of meters and is readily expressed as a pressure loss by multiplying it by $\rho g$, $\rho$ being the density of the fluid and $g$ the acceleration due to gravity.

**Multiple Pipe System:** In a multiple pipe system the pipes can be connected in series, in parallel or within a branching or closed loop network. In order to determine the routing of the flow in the network and the friction losses in each branch the fundamental mass and energy conservation principles have to be applied. The mass conservation has to be fulfilled in every node of the network. The mass flowing into a network node has to equal the mass flowing out of this node. The energy balance is applied across every branch of the network. It gives a relation between the nodal pressures and the kinetic energy of the fluid.

The general flow equation for steady flow of gas in a pipe is derived from *Bernoulli’s Equation*. The pressure decreases along the pipe due to friction such that the density of the gas also decreases. For steady state the mass flow stays constant throughout the pipe. The mass conservation assuming a constant cross section of the pipe reduces to

$$\rho_1 u_1 = \rho_2 u_2$$  \hspace{1cm} (6.11)
Figure 6.4: Pipe section of length \( L \) with an infinitesimal element of length \( dx \)

Since density decreases along the pipe, the velocity and the kinetic energy of the fluid go up. Because in gas flows the velocities vary along the pipe a small element has to be considered for the calculation of the friction losses. To obtain the total head loss the result then needs to be integrated over the entire pipe length. The energy balance formulated at a small element \( dx \) of the pipe as shown in Figure 6.4 gives

\[
\frac{p}{\rho g} + \frac{u^2}{2g} + z = \frac{p + dp}{(\rho + d\rho)g} + \frac{(u + du)^2}{2g} + (z + dz) + dh_f
\]

with \( p \) being the static pressure, \( \rho \) the density, \( g \) the acceleration due to gravity, \( u \) the velocity and \( dh_f \) the head loss due to friction. For the small element \( dx \) it is assumed that changes in density, in velocity and hence kinetic energy are negligible. Furthermore, for the planar network considered in this work the elevation term can be omitted. Applying these assumptions to eq. 6.12 and using eq. 6.1 to express the head loss it leads to

\[
-\frac{dp}{\rho} = f\frac{\rho u^2}{D_h} \frac{dx}{D_h}
\]

Further simplification such as the assumption of ideal gas and isothermal flow were made. The dissipation of kinetic energy into heat is assumed to be balanced by the heat flux through the wall of the pipes. Integration of the differential equation for a pipe of length \( L \) finally gives

\[
p_1^2 - p_2^2 = \frac{16fRT_sL}{\pi^2D_h^5}\dot{m}_{12}^2
\]

Details about the derivation of a more general form of eq. 6.14 can be found in [22].
Pipes: Every pipe has two terminal nodes. These nodes can be any of the kind junction, opening or pump. Based on the pressure in the two terminal nodes the direction and magnitude of the mass flow through the pipe is calculated. During this calculation friction losses in the pipe but also in the components are considered.

As shown in Figure 6.5 the pipe model covers four different resistances. One is due to friction losses in the pipe section, the major losses and the other three are minor losses. Minor losses occur at the two ends of the pipe where the nodes and and the pipe join with different diameters. Assuming a larger diameter for the nodes compared to the pipes connected a contraction loss is calculated for incoming and an expansion loss for the leaving flow. Additionally, a minor loss is introduced to model an arbitrary flow element such as a valve which can be freely adjusted by choosing an appropriate loss factor K. Major losses are taken into account using eq. 6.14 based on eq. 6.1. For the evaluation of the minor losses, eq. 6.6 has to be used. The pipe schematic shown in Figure 6.6 is separated into two parts, a first part with major losses $R_{\text{major}}$ and a second part $R_{\text{minor}}$. Using eq. 6.6 the pressure drop due to minor loss becomes

$$p_j - p_k = K \frac{8}{\pi^2 D_n^4} \frac{RT}{p} \dot{m}_{ik}^2$$ (6.15)

On the right hand side of eq. 6.15, the density that is according to the ideal gas law a function of temperature and pressure is still in the denominator. In order to be able to express the flow through a resistive element as a function of terminal pressures only the pressure appearing on the right hand side could be picked to be either of the terminal pressures. The pressure drop would then be calculated with reference to either the upstream or the downstream pressure. It turned out though to be more convenient to pick the average between the two terminal pressures $\frac{p_j + p_k}{2}$ as a reference. Inserting this choice into eq. 6.15 the relation between nodal pressures and the mass flow becomes
\[(p_j - p_k)(p_j + p_k) = \ldots\]

\[p_j^2 - p_k^2 = K \frac{16}{\pi^2 D_h^4} RT \dot{m}_{ik}^2\]  
(6.16)

Using the average pressure as a reference value in eq. 6.16 has led to the same quadratic form as previously obtained for the major losses expressed in eq. 6.14. This fact allows the different losses to be combined nicely. The calculation of the mass flow in a pipe branch then takes the general form

\[
\dot{m}_{ik} = s_{ik} \left( \frac{p_j^2 - p_k^2}{\gamma_{ik}} \right)^{\frac{1}{2}}
\]  
(6.17)

\[
s_{ik} =
\]

\[
1 \quad \text{if} \quad (p_i - p_k) \geq 0
\]

\[
-1 \quad \text{if} \quad (p_i - p_k) < 0
\]

major loss: \[\gamma_{ij} = \alpha_{ij} = f \frac{16L}{\pi^2 D_h^5} RT\]

minor loss: \[\gamma_{jk} = \beta_{jk} = K \frac{16}{\pi^2 D_h^4} RT\]

combined loss: \[\gamma_{ik} = \sum_l \alpha_{ij,l} + \sum_m \beta_{jk,m}\]

**Nodes:** At a node the flow function which is the net between incoming and outgoing mass flows from the connected pipes is calculated. For a node it has to be known which pipes are connected to it and what is the flow direction in those pipes. Every node has a specified diameter that has to be larger than the pipe diameter. Based on the discrepancy between the two diameters a
contraction or expansion loss is calculated. In Figure 6.7 a contraction loss considering a loss factor according to eq. 6.7 is calculated for the mass flow $\dot{m}_{ij}$ and $\dot{m}_{im}$. The pipe diameter $D_{\text{pipe}}$ corresponds to $D_1$ and $D_{\text{node}}$ to $D_2$. For the mass flow $\dot{m}_{ki}$ an expansion loss is evaluated using the loss factor calculated from eq. 6.8. $D_{\text{pipe}}$ is here equal to $D_2$ and $D_{\text{node}}$ to $D_1$.

**Openings:** In addition to the simple node an opening includes a flow rate into the room as shown in Figure 6.8. A loss factor and a pressure value at the interface to the room has to be assigned to each opening in order to be able to correctly calculate the leaving mass flow. The mass flow is then calculated using eq. 6.17 considering only minor losses. The flow rate to the room at each opening is calculated based on the mass flow using the free stream density of the room, $\rho_{\text{room}}$. 

Figure 6.7: Node model with connecting pipes.

Figure 6.8: Opening model with a virtual branch leaving to the room
**Pumps:** The pump object according to Figure 6.3 is also a node object and hence has the same hydraulic diameter as a node. The connection of a pump to the pipe is therefore treated equally. However an additional diameter for the pump is needed representing the size of the exhaust duct. Having these two different diameters defined it allows a flexible and realistic modeling of the pump and its connection to a single or multiple pipes. If a single pipe is connected to a pump the nodal and the exhaust duct diameter are the same; if multiple pipes are connected the nodal diameter is going to be larger than the pipe and the exhaust duct diameter such that additional losses due to a sudden expansion from the pump into the connection node are considered. Beside the losses on the pressure side a loss factor on the suction side can be specified to account for loss elements such as for instance a protective grating mounted at the facade. The model as described is shown in Figure 6.9.

![Diagram of a pump model](attachment:image.png)

**Figure 6.9:** Pump model with a double node approach

The pump incorporates a pressure-flow rate characteristics given by the type of pump selected. The pressure-flow rate characteristics is implemented as a second order polynomial of the form

\[ p_i - p_\infty = A\dot{Q}^2 + B\dot{Q} + C \]  

(6.18)

where \( \dot{Q} \) is the volumetric flow rate and \( A, B \) and \( C \) are the coefficients of the pump’s characteristic curve. The \( C \) corresponds to the stall pressure of the pump. Since the form of eq. 6.18 does not allow to be solved uniquely for the flow rate a transformation as suggested by [15] is applied. The transformed flow rate is

\[ \eta = \dot{Q} + \frac{B}{2A} \]  

(6.19)

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Eliminating $\dot{Q}$ in eq. 6.18 leads to

$$p_i - p_\infty = A\eta^2 + D \quad (6.20)$$

$$D = C - \frac{B^2}{4A} \quad (6.21)$$

Now, eq. 6.20 can directly be solved for $\eta$, giving

$$\eta = \left\{ \frac{\Delta p - D}{A} \right\}^{\frac{1}{2}} \quad (6.22)$$

$$\dot{Q} = \left( \frac{\Delta p_i \infty}{A} - E \right)^{\frac{1}{2}} - F \quad (6.23)$$

$$E = \frac{C - \frac{B^2}{4A}}{A}$$

$$F = \frac{B}{2A}$$

The mass flow of the pump is finally calculated by multiplying the flow rate with the density at the pump node expressed through the local pressure and temperature according to the ideal gas law.

$$\dot{m}_\infty = \dot{Q} \frac{p_i}{RT_i} = \frac{p_i \left\{ \left( \frac{\Delta p_i \infty}{A} - E \right)^{\frac{1}{2}} - F \right\}}{RT_i} \quad (6.24)$$

The above equation expresses the mass flow delivered by the pump, given a certain pressure rise. However, it only describes the characteristics of the pump alone and does not include any resistive elements. The problem is that the pressure drop through a resistance depends on the flow rate and the flow rate delivered by the pump is unknown as long the pressure applied is unknown. If the calculation of pressure difference across a resistance (minor loss) and a pump had the same mathematical form an explicit solution of the mass flow could be obtained. Since this is not the case an iterative solving method is required. For this purpose an auxiliary pressure $\tilde{p}$ is introduced between the pump and the terminal resistances as shown in Figure 6.10. Based on a guess for $\tilde{p}$ the mass flow over the resistance and through the pump is calculated. The interface pressure $\tilde{p}$ is then adjusted iteratively until the value is such that the two mass flows in the sub branches become the same.
Figure 6.10: Schematic pump with intermediate pressure introduced to determine the effective pump mass flow.

**Solving Algorithm**

Using eq. 6.17 and eq. 6.24 to calculate the mass flow in pipes and through the pumps of the network it leads to a large set of nonlinear equations. The mass flows are calculated based on the pressures at the nodes. The equations are coupled with each other through the physical coupling of the pipes in the network. At steady state the mass for each node that connects several pipes together must be conserved, i.e. the net mass flow must be zero. As a consequence a valid solution of the flows in the network is obtained when the mass at every single node in the network is conserved. In the electrical circuit calculus the mass conservation at a node is known as *Kirchhoff’s 1st law*. Focusing on mass conservation at the network nodes to solve for nodal pressures is known as a nodal formulation of the routing problem. This formulation leads to a problem of the form

\[ F_i(P_j) = 0 \quad (6.25) \]

with \( P_j \) being the nodal pressures and \( F_i \) the nonlinear flow function which is a vector holding the results from the mass conservation at the nodes. \( F_i \) is expressed as

\[ F_i = \sum_{k=1}^{m} a_{ik} \dot{m}_k \quad i = 1, 2, ..., n \quad (6.26) \]

where \( a_{ik} \) is the *node-edge incidence matrix* [3] defining the topology of the network and \( \dot{m}_k \) are the mass flows of the individual pipes. The \( m \) is the total number of pipes in the network and \( n \) the total number of nodes. As a consequence \( n \) equations for the \( n \) unknown nodal pressures are available and the problem can be solved. For the solution of this system of non-linear equations the well known *Newton-Raphson method* as described for instance in [32],[10] or [40] is suitable. For the routing problem to be solved it is assumed that a unique solution exists. This is important for the selection of the initial values that have to be provided to start the iterative solving. If
multiple solutions exist, it is not clear which solution is returned by the solving algorithm. The newton method leads to quadratic convergence if an initial guess close to the final solution is provided. If the initial value is far a way from the solution convergence is not guaranteed. Because of this properties of the Newton method it is sometimes referred to as a ’local’ method. In order to make the Newton algorithm more likely to converge for a larger set of initial values a damping mechanism is introduced. The damping is based on a trust-region that it either expanded or contracted based on the appropriateness of the approximation of the model function [4]. In a trial step the quality of the approximation is verified and the step size for the correction during the next iteration is determined. The nodal pressures for the time \( k + 1 \) based on the previous values at time \( k \) using the trust-region newton method are calculated according to

\[
P_{j}^{k+1} = P_{j}^{k} - \gamma z_{j}^{k}
\]

(6.27)

The corrector step \( z_{j}^{k} \) is calculated by solving the system of linear equations

\[
J_{ij}^{k} z_{j}^{k} = F_{i}^{k}
\]

(6.28)

and adjusted according to the damping factor \( \gamma \in [0, 1] \). The matrix \( J_{ij}^{k} \) is the Jacobi matrix and holds the partial derivatives of \( F_{i}(P_{j}) \) with respect to \( P_{j} \).

\[
J_{ij}^{k} = \frac{\partial F_{i}^{k}}{\partial P_{j}^{k}}
\]

(6.29)

**Simulation: Calculation Sequence**

The calculation procedure as implemented in an object oriented programming language is subsequently explained in its main steps. First, the network described using a XML structure is parsed and translated into the software. The XML file contains all the topological information as well as global physical constants, reference values and specific parameters of the network components pipes, nodes, openings and pumps. While parsing, the network components are instantiated (objects are generated) and the incidence matrix is created. The parameters of those network components are shown in Figure 6.11. Once all data is available the actual iterative solving of nodal pressures and flow rates using the newton method starts. Any type of node is initialized with a pressure value randomly picked within a realistic range. As soon as a pressure value has been assigned to a node object the pipe objects that are connected to this node object are caused to calculate their mass flow and to return it. All nodal pressures are then collected within a vector and used to calculate
the flow function vector according to eq. 6.26. With the flow function and the pressures the Jacobian according to eq. 6.29 is put together and the first trial step without damping ($\gamma = 1$) according to eq. 6.28 is calculated. The pressure values and the flow function are then updated using eq. 6.27 and eq. 6.26. After this step the convergence is checked by comparing the norm of $F_i^k$ to the one of $F_i^{k+1}$. If the solution is converging nicely the solution procedure as described is repeated. If convergence is poor or the solution is not convergent at all a damping factor is applied. This involves again the same iterative process. The damping factor is adjusted through the calculation of trial steps until convergence is reached and the corrector step is actually applied. For every iteration it is checked whether damping is necessary or not before going into the damping sub loop. When the solution converged and the norm of the flow function $F_i$ is sufficiently close to zero the computation has ended and the results are returned.
### 6.4.2 Fan/Supply System Operation Under Variable Pressure Boundary Conditions

Any air supply system is exposed to outside conditions on one and to room conditions on the other side. In a balanced ventilation system the ambient and the room pressure should ideally be the same. A pressure difference can occur due to a mismatch in flow rates between the supply and the exhaust. This is the case if for instance a variable exhaust with a demand control is used and the supply is not properly adjusted. It also can be caused by a difference of local pressures between the air intake location and the exhaust location. In order for this pressure difference to occur a certain air movement giving rise to a dynamic component of the pressure is required. If wind is blowing around a building a non-uniform pressure distribution results on its envelope. In the case of a decentralized, supply-only system that takes in outside air through the facade this means that every device encounters a different local intake pressure depending on its location in the facade. Consequently the amount of supply air can deviate from the initial setpoint such that a mismatch with the exhaust occurs. Small fans as they are used in decentralized air handling units are usually more sensitive to changes in pressure than large ones. Depending on the type of fans and the control algorithms used the resulting change of flow rate can be significant. For the understanding of the behavior of small AHU under the influence of variable pressure boundaries, different scenarios are considered and discussed.

A decentralized AHU that has its intake directly in the facade and supplies air via an air distribution system to the room can be described by a simple model, including few pressure values as shown in Figure 6.12. The $p_\infty$ represents the local outside pressure at the intake, $p_{\text{floor}}$ is the pressure at the interface to the air distribution system assumed to be in the floor and $p_{\text{Room}}$ is the room pressure. The loss factor $R_{\text{ADS}}$ describes the loss characteristics of the air distribution system and its resistance is depending on the flow rate that is handled. The pressure drop in the ADS is simply the difference between the

![Figure 6.12: Pressures seen by the supply system.](image-url)
pressure in the floor and the pressure in the room.

\[ \Delta p_{ADS} = p_{floor} - p_{room} \]  \hspace{1cm} (6.30)

The pressure difference over the fan is written as

\[ \Delta p_{fan} = p_{floor} - p_{\infty} \]  \hspace{1cm} (6.31)

Putting eq. 6.31 into eq. 6.30 we get a new expression shown in eq. 6.32 for the pressure loss of the distribution system containing all pressures of the model.

\[ \Delta p_{ADS} = p_{\infty} + \Delta p_{fan} - p_{room} \]  \hspace{1cm} (6.32)

It is the pressure drop in the supply system and therefore appropriate for plotting the total loss against the flow rate. In a plot where the fan characteristics is included the pressure difference reduces to the fan pressure. This is for the case that the outside pressure and the pressure in the room are the same. The maximum \( \Delta p \) on the ordinate then corresponds to the stall pressure of the fan. In Figure 6.13 the fan characteristics and the total plant characteristics for different loss curves are plotted. The intersection point between the fan characteristics and the loss curve for the entire plant is the operation point (OP). If the fan is switched on, the flow rate automatically sets to the value \( Q_{OP} \) where the fans pressure rise balances out the loss of the plant at that flow rate. In the OP the total plant characteristic therefore reaches the zero \( \Delta p \)-line. The plant always automatically adjusts its operation to the OP. For instance if the loss curve is changed, the OP changes to the new intersection point of the altered loss curve and the fan characteristics. The OP also changes if the fan power, i.e. the speed is altered (Figure 6.14). Lowering the fan speed forces the operation point to move along the loss curve in direction of lower pressure rise and lower flow rate.

The question is now, how changes in the exhaust power/flow rate and changes of the outside pressure affect the pressure in the inside of the building (room, floor) and the operation point OP of the system.

Consider the case where the relative pressure in the room drops and gets negative \( (p_{room} < 0 \leq p_{\infty}) \). The reason for this negative pressure in the room is an increase of the flow rate of the exhaust fan. According to eq. 6.32 it is clear that a decrease in room pressure leads to a larger over-all pressure drop if all the other pressures are assumed constant. This is of course true because a larger flow rate is demanded and hence a larger drop according to the loss curve occurs. At the higher flow rate the AHU is not able anymore to keep the pressure rise and even less to support the increased pressure rise (as a matter
Figure 6.13: General loss curves and characteristics

of fact the pressure rise drops). As a consequence the exhaust fan must pull the system to be able to keep the flow rate. In order to avoid the negative pressure in the room the AHU must overcome the newly added pressure drop. This is only possible through an additional energy input, i.e. an increase of the fan power. In Figure 6.15 it can be seen that a higher fan speed is necessary to be able to compensate the additional pressure drop along the loss curve at a higher flow rate. If the fan power is not increased the operation point for the fan moves along the fan characteristics to the right towards higher flow rates and lower pressures. The opposite behaviour is true for the case that the exhaust flow rate reduces ($p_{\text{room}} > p_{\infty} \geq 0$). In order to avoid positive relative pressure in the room the AHU has to reduce its pressure rise and hence its fan
power because of the lower flow rates demanded (from old operation point to a new OP, in opposite direction to the one shown in Figure 6.15).

In the case of negative pressure at the facade ($p_\infty < 0$) the pressure drop according to eq. 6.32 decreases if all the other pressures are assumed to be constant. This is true because the fan has now to overcome a larger pressure drop such that the flow rate it can deliver decreases. Unlike the pressure drop in the ADS that is flow rate dependent the relative outside pressure is not. As a consequence the negative relative outside pressure brings about the loss curve shown in Figure 6.16(a) to shift towards higher values by the amount of the change of outside pressure. If it is assumed that the exhaust flow rate remains unchanged the same negative pressure as found outside builds up in the room unless the supply is matched to the exhaust. A compensation of the outside pressure drop giving a constant room pressure can only be achieved with a larger energy input, i.e. a higher fan power. Therefore the fan power of the AHU has to be increased to match the new operation condition. The wished operation point is reached when moving in Figure 6.16(b) along the shifted
loss curve to the right until the initial flow rate $Q_{old}$ is matched again. The new fan curve that can run at this condition has to go through this operation point. Again, the opposite is true for a positive outside pressure ($p_\infty > 0$). The fan power has to be reduced to keep the floor pressure constant. If the outside pressure is larger than the pressure drop from the floor to the room, the throttling of the fans is not sufficient such that additionally a damper has to be used.

### 6.4.3 Network Topologies

The topology of a network describes the arrangement of network elements, i.e. their configuration in terms of connections. The basic elements of a network are nodes and links. Two network topologies are equal if their configuration of connections is the same. In that case the logical network is also the same. A logical network can be represented by a graph as known from graph theory. A graph does only reflect the topology of a network but not its physical
properties. Two real or physical networks, although having the same topology may still differ from each other for example in link length or in type of components being connected. In a physical network like the air distribution network (ADN) the links are effectively pipes or ducts which have to be considered as connections with a resistance obstructing the flow of air. The longer the link the larger the resistance. Because of the different types of nodes present in a real ADN the behavior of the network also largely depends on the distribution of the different node types. Three types are possible in an ADN: Junctions, openings and pumps. More than the mere logical topology of the network does the location of the sources (pumps) and the sinks (opening) and the distribution of the resistances (links and openings) determine the routing of the flow in the ADS. It is important for the understanding of the air flow routing to analyze the connection pattern of resistances in different topologies. There are two fundamental connection patterns: In series or in parallel connection. In a pure serial connection there is only a single path a flow can take to travel through all components. In a parallel arrangement separate paths are provided for the flow to travel through the components in the network. In analogy to electrical circuits the flow of air through a branch with components
(nodes) connected in series remains constant throughout the entire branch. This simply follows from the continuity requirement. As does the voltage in an electrical circuit the pressure in a fluidic system drops over every resistance along the path of the flow. In a parallel arrangement the pressure drop along each separate path is the same while the flow rate is partitioned between the branches depending on the resistances of each individual branch.

The most common, basic network topologies known from communication technology are the linear, bus, tree, ring, star, mesh or the fully connected type [9], [38]. More complex networks can be formed combining these different basic types shown in Figure 6.17.

Assuming the pump in the ADS to be at one end of a network with linear topology the nodes within the network are purely connected in series. For a tree topology with the pump being the root node the connection of the nodes is purely parallel but only for a tree height of two. The tree height is defined as the depth of a tree plus one and the depth is the length of the path (number of steps) from the root to the node. For larger tree heights it becomes a mixture of parallel and in series connection of the nodes. For the bus topology if the links had no resistance it would be a purely parallel connection of the nodes. Since in a real ADN the links have a resistance, it is again a mixture of parallel and in series connections of the nodes. The bus topology for an ADS only makes sense if all terminal nodes are openings but
Figure 6.18: Difference in fluidic circuit of networks for junctions and openings

for the tree topology there is still a possibility to choose some nodes to be openings and some to be simple junctions. Replacing a simple junction with a connectivity larger than one in the network with an opening, adds another branch that is in parallel with the other leaving branches. How this exchange of node types affects a network is illustrated in Figure 6.18. In the upper part of Figure 6.18 a junction with one incoming and 2 leaving branches is shown to the left. To the right the corresponding fluidic circuit with the flow resistances of the links RL and those across the openings RO is shown. In the lower part the same situation after exchanging a junction with an opening is represented. The effect of replacing the junction with an opening can be seen in the fluidic circuit. The opening introduces an additional branch that is in parallel with the other leaving branches and carries the airflow released to the outside of the network. The impact of introducing an additional opening in place of a junction is for all types of networks the same. For all type of topologies shown in Figure 6.17 except for the linear one the connection of nodes is a mixture of parallel and in series connections. Similar to the distribution between junctions and openings the position of the pump in the network also plays an important role and influences the characteristics of the physical network.
Designing Networks for Air Distribution

Having in mind the lake of fresh air as an ideal solution to the problem of air distribution in the space, the question is what is the quality of this solution that makes it preferable. Quality criteria have to be defined such that the right design incentives for the ADN are obtained. The most obvious quality criterion is the one of regular distribution. The primary goal of the ADS is to distribute the fresh air evenly over the entire space. For a regular spatial distribution of openings this is the same as requiring the air flow to be the same for all openings installed. A second important criterion is the economy. The ADS should have low pressure drops. The lower the pressure drop in the system the higher the amount of air transported in the system for a given fan power input. As a consequence low pressure drops increase the system efficiency and lower the running costs of a plant. A third very important quality criterion is the one of responsivity. A responsive system is able to react to a small change of the boundary conditions by moving its state from one stable equilibrium to another. The small change implies that only small forces are involved. As an example consider two neighboring rooms supplied through the same ADS that by default have the same ventilation rate. Assuming the boundary conditions to change such that the rooms now have a different ventilation rate dictated by the exhaust. A responsive ADS adjusts the supply flow rate to the two rooms without imposing a significant pressure difference between the two rooms. Such a system hence allows for a very flexible allocation of capacity to supply fresh air. A non-responsive system in contrast would either resist to an increase of the ventilation rate or cause the pressure difference between the two rooms to grow.

The three quality criteria introduced are not independent of each other but they are causally interlinked. Regular air distribution with an ADN is only possible if a homogeneous pressure distribution is encountered in the ducting network. A nearly homogeneous pressure distribution however is only possible if zero or very small air flow velocities in the ductwork are encountered or large terminal resistances at the openings are used. In order for the ADS to be economic and work with small pressure drops the terminal resistances need to be small. If small terminal resistances are required even smaller pressure drops in the pipes for the routing of the air flows are necessary for the ADN to be responsive. This last requirement again calls for low velocities and accordingly for a sufficiently large capacity of the ductwork. The bottom line from this causal chain is that we have to have low air flow velocities in the ductwork. This then leads to nearly homogeneous pressures and hence flow rates, to low pressure drops in the pipes such that low terminal resistances can be used to
be economic without loosing the responsivity of the system.

Knowing the quality criteria for the ADS is a necessity for the design of a well performing ADN. For a systematic analysis of ADNs however, guidelines for the generation of suitable networks are needed. Based on the connection patterns introduced, namely the parallel and in series connection and its related properties, some design guidelines can be found. The situation can be assumed that a building is supplied from the outside with pumps located at the periphery. The path of the supply air in the network is hence from the outside to the core of the building. If a purely serial arrangement, i.e. a linear topology of openings with a single connection per opening is chosen, it is impossible to achieve an even distribution of flow rates among the openings if all of them have the same resistance. Throttling would be needed to adjust the individual resistances such that a regular distribution can result. The pure parallel arrangement is the only that leads to entirely equal resistances for the flow if each branch has the same length and the same terminal resistance. As discussed previously most common topologies like bus, tree or mesh topologies are mixtures of parallel and in series connections. As a consequence the ideal parallel character cannot be maintained in such topologies. A branching network with a tree topology does only satisfy the conditions for a regular distribution if uneven terminal resistances are used or the tree is fully symmetrical and has openings only at the end of the tree. The mesh topology although not showing purely parallel characteristics, allows the ideal parallel character for two uneven branches to be partly restored by closing loops in the network. The effect of closing loops by interlinking openings is illustrated in Figure 6.19. In the section of the networks shown in Figure 6.19 the length of link 2 is larger than the one of link 3 and consequently the flow resistance $RL_2$ is larger than $RL_3$. Because link 2 and 3 are connected in parallel more supply air will be released through the opening connected to link 3. This situation is shown in the upper part of the figure with the fluidic circuit to the right representing the resistance for the two branches in parallel with the link and the opening resistances per branch connected in series. The situation changes if an additional link is introduced to connect the two openings. Looking at the fluidic network shows the added link with its resistance $RL_4$ connecting the parallel paths after the link resistances $RL_2$ and $RL_3$. Because the pressure drop across links 2 and 3 is different there will be a balancing flow induced in link 4. If the pressure drop in links 2 and 3 is small the flow and the resulting pressure drop in link 4 will be small too. This situation leads then to almost equal pressures upstream of the openings. Because the resistances through both openings is the same if no throttling is applied the flow rate being supplied by each opening will then also be almost the same. A mesh topology
with closed loops is hence necessary to achieve the goal of regular distribution of flow rates. The example shown in Figure 6.19 only demonstrates the effect of closing a single loop. If several loops are interconnected and evolve from the periphery towards the core the in series connection of some openings cannot be avoided because only the outmost nodes are fed by pumps and the air has to be transported over several stations to the core. In this situation the use of some junctions might be considered to further redistribute the flow from the outer to the inner zone of the building.

6.4.4 Influence of Wind On Buildings

A ventilation system in a building is operated under variable conditions and influenced by various disturbances whereof wind is one component. The sensitivity of a ventilation system to these external disturbances strongly depends on its design. Decentralized ventilation with AHUs taking air from the facade, are more strongly influenced by the wind and the non-homogeneous pressure distribution than centralized systems with a single intake/outlet.
Wind is highly variable, turbulent and its speed changes with the height above ground because of the presence of an atmospheric boundary layer. The boundary layer profile of the wind speed is shown in Figure 6.20. As commonly used to describe any turbulent flow, wind can also be expressed as the sum of an average velocity component $V$ and a time variant component $v(t)$ that represents the fluctuations. The fluctuations disappear when the velocity is time averaged over a sufficiently long period. Depending on the length of the time period chosen to calculate the average wind speed $\bar{V}$, its value can fluctuate. These fluctuations of the time average are known as wind gusts. The wind profile with the time average of the wind speed depending on the height above ground is commonly expressed with a formula of the form

$$\bar{V}(z) = \bar{V}_g \left( \frac{z}{z_g} \right)^\alpha$$

(6.33)

where $\bar{V}$ are average wind speeds, $z$ is the height above ground and $\alpha$ is a coefficient which describes the terrain. The index $g$ denotes known values at a certain height above ground. Typically a value of 10 m is used for $z_g$ because meteorological stations often measure wind speeds at this height above ground. The exponent $\alpha$ takes a value around 0.15 for the open country, 0.25
for suburban areas and little towns and a value around 0.4 for city centers with tall buildings [33].

A steady CFD simulation for a selected building, assuming a constant time-averaged wind speed of 10 m/s at a height of 200 m and a value of 0.3 for $\alpha$ has been performed. The velocity vectors represented in Figures 6.21 show similarly to Figure 6.20 the various recirculation zones on the upwind side, at the foot and on the opposite side, at the foot and the head of the building. Also, when looking from the top onto the building, recirculation zones that form at the edges of the wake region can be observed. As a consequence of

![Velocity vector representation of flow around a high-rise building](image)

(a) Side view with wind from the left  (b) Top view with wind from the bottom

Figure 6.21: Velocity vector representation of flow around a high-rise building

the non-homogeneous velocity distribution a non-homogeneous pressure field builds up around the building. This pressure disturbance in the otherwise homogeneous pressure field of the freestream also affects other buildings that are located in the close neighborhood. The pressure distribution around the simulated building is shown in Figure 6.22. The red zones represent areas of neutral pressure, i.e. ambient pressure. Locally, in the stagnant region of the building the pressure is above ambient pressure level because of the dynamic head being converted into static pressure. On the sides and in the wake of the building negative pressure zones are formed due to the increased flow velocities. Further downstream the negative pressure is again recovered. Although the flow around the building was treated as steady flow in the simulation the actual flow as every real flow phenomena is unsteady. Vortex shedding in the
Figure 6.22: Static pressure distribution around a high rise building. Top view with wind from the bottom of the figure.

...wake leads to fluctuations in local velocities and pressure. The significance and the strength of those instationary effects can only be estimated when additionally performing an unsteady simulation. Also, the actual pressure distribution around a building is different for every building shape, height, etc. and largely depends on the boundary conditions chosen. For this reason, when trying to treat generic problems without making complex simulations some simplifying assumptions have to be made. A rough idea of the pressure distribution around a building can be gained using area averaged pressure coefficients. Being aware of the simplifications involved when representing the pressure distribution around a building with a constant average value this approach can be helpful for an estimation of the influence wind can have on the operation of a ventilation system. This simplified approach was used to estimate the wind impact for the ventilation in a generic building. Most frequently discussed in literature are square buildings (high or low rise) and dwellings with inclined roofs. For those types, average pressure values based on measurements are available.
6.4.5 Building Test Case Setup

For the study presented in the AIVC paper a square building with 5 floors was defined. The same building setup was also used in the study presented in the PLEA paper. The side length of the floor plan was chosen to be 20 meters which results in a total base area of 400 $m^2$ per floor. The design ventilation rate for the building was determined using a specific ventilation rate per square meter of floor area. The design building is chosen to consist of office space with a specific ventilation rate of 4 $m^3/hm^2$. The floor plan is an open space with a supporting core defined to be 2.5 times 4 meters large. This gives a design ventilation rate of 1600 $m^3/h$ per floor and an overall ventilation rate of 8000 $m^3/h$ for the building. These flow rates are used to determine the amount of decentralized air supply units that need to be installed and also to size the central exhaust fan.

For an estimation of the pressure distribution around the building in case of wind impact a trigonometric relation with surface averaged pressure coefficients suggested by [36] is used. The trigonometric relation writes out as

$$c_p(\theta) = \frac{1}{2}((c_p(0) + c_p(\pi))(\cos^2(\theta)))^{\frac{1}{4}} + (c_p(0) - c_p(\pi))(\cos^2(\theta)))^{\frac{3}{4}} + (c_p(\pi/2) + c_p(3/4\pi))(\sin^2(\theta))^2 + (c_p(\pi/2) - c_p(3/4\pi))\sin(\theta))$$

(6.34)

where the $c_p(0), c_p(\pi/2), c_p(\pi), c_p(3/4\pi)$ are the surface averaged pressure coefficients at a wall when the wind direction encloses an angle of 0, $\pi/2$, $\pi$ or $3/4\pi$ respectively to the normal of that wall. For these surface averaged pressure coefficients, values for high-rise buildings as used in the software WeatherSmart by [5] where picked. Based on wind statistics by MeteoSwiss for Zurich, Switzerland the strength of the wind was determined. The most probable wind speed of 6 and a more severe situation with speeds of 10 $m/s$ was chosen. More extreme values were not considered. In Figure 6.23 the distribution of pressure differences based on eq. 6.34 for different wind directions and a speed of 10 $m/s$ is shown. Having assumed the wind to hit the building perpendicular to one side the pressure on that side relative to ambient pressure according to Figure 6.23 is $+51.05$ Pa. On the opposite side a negative average pressure of $-31.9$ Pa is found. The most expressed negative pressures for a square building with frontal wind are found on the side walls with $-44.67$ Pa. These positive and negative average pressure differences found around the building are used as boundary conditions for the pumps being installed on the respective sides of the building.
Figure 6.23: Wind angle dependence on predicted wall pressure differences for isolated buildings

6.4.6 Pressure Drop Evaluation for Centralized Fan Units

At least half but rather more of the total pressure drop in the ventilation system occurs on the supply side. This is mainly because throttling is used to achieve the same flow rate in every supply branch in the building. Typical ventilation systems used in office buildings are balanced systems operated centrally by large radial fans for the supply and the exhaust. Radial fans of different blade designs with belt drives are most commonly used. Little published information is available about pressure drops of real plants being built. Building designers select fans according to the range of flow rate required and the actual loss characteristics of the entire supply system. Choosing the appropriate fan size means also that the fan operates close to its maximum efficiency. The total flow rate needed for all five floors of the design building are 8000 m$^3$/h. To estimate the pressure drop in a supply system of the selected size of 8000 m$^3$/h, commercially available radial fans from different suppliers have been analyzed. A compilation of different fans displayed in an efficiency-pressure rise plane is shown in Figure 6.24. The fans represented in this figure vary in diameter of the wheels, their blade design and whether they are single or double inlet fans. Fans with an efficiency below 0.75 use a forward-curved blade design. The more efficient fans have a backward inclined blading. The fans
Figure 6.24: Evaluation of radial fans of different suppliers in a pressure rise - efficiency plane for a flow rate of 8000 \( m^3/h \)

can be arranged in three groups represented by the red ellipsoids in Figure 6.24. By looking at the operation range of the different fans one can conclude that the topmost group in the figure represents fans that are at their flow rate capacity limit and are not suitable. Conversely are the fans in the lowest group slightly oversized such that they could handle larger flow rates up to 20000 \( m^3/h \). In the mid field are fans with a total pressure rise around 800 Pa which is a realistic value.
6.5 Paper: AIVC 2008, Advanced Building Ventilation and Environmental Technology for Addressing Climate Change Issues

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Advanced Distribution and Decentralized Supply: A Network Approach for Minimum Pressure Losses And Maximum Comfort

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ABSTRACT

The paper presents a new concept for a low pressure drop supply system suited for the ventilation of office spaces. Lower pressure losses for the air distribution allow a downsizing of the fans cutting down the investment costs for the equipment and the energy consumption during operation. Great potential savings in high-valued energy (exergy) are possible in large buildings where this concept could reduce pressure drops by a factor up to ten. This supply system consists of decentralized air handling units installed near the façade. They feed an interlaced duct network that is integrated in the floor construction. The air is supplied to the room at very low momentum through openings in the floor. Following the low exergy principle the heat exchangers of the air-handling units strictly operate at supply temperatures close to the room temperature, and hence would allow a very efficient operation of a heat pump with a high coefficient of performance. The proposed duct network design eliminates the problem often found in decentralized systems of insufficient air supply to the core of the building. It provides a regular distribution of the air over the entire floor space even in large open plan offices and under non-homogeneous pressure conditions from the wind on the façade or from a locally demand-controlled exhaust system as described by (Baldini and Leibundgut, 2005). As a proof of concept a steady state analysis of the air distribution has been performed for different network topologies. The simulations showed that an effective distribution of the air is possible even for non-homogeneous pressure boundary conditions when the network’s topology is close to the one of a mesh. The distribution system is most robust to changing boundary conditions if the topology of the network is such that the velocities within the pipes are small enough to lead to nearly homogeneous static pressures.

1. INTRODUCTION

Climate change forces us to reduce the anthropogenic CO2 emissions significantly in order to limit the danger of more severe natural catastrophes in future. The building sector is with a 38% share of the total primary energy consumption and 34% of the total CO2 emissions world-wide a large contributor to the problem (Price et al., 2006). More than the construction does the operation of the buildings over the entire life-span of 20 to 50 years cause the largest climatic impact. For this reason HVAC systems must be improved to minimize environmental impact during operation. There is still a large potential in actual HVAC systems for improvements concerning energy consumption as well as cost effectiveness. New concepts have to be introduced to exploit these potentials. The current paper proposes a new ventilation concept from which significant energy savings are expected.

Nowadays heat recovery for ventilation in office buildings is common practice and helped reducing ventilation losses significantly. Further reductions are possible by decoupling the ventilation from the heating or cooling needs.
This is very common in Europe while in the US or in most Asian countries air-conditioning is still the predominant approach to satisfy comfort needs. The decoupling allows very low ventilation rates to be used, to just satisfy the indoor air quality (IAQ) requirements. When making these improvements it is important also to consider distribution losses of the ventilation that now become even more relevant. The issue of pressure losses during distribution is hardly addressed explicitly in literature. Most of the studies found restrict their scope to general energy evaluations of ventilation systems without identifying the origin of the losses. Pressure losses in ducting systems directly relate to fan power and therefore to losses of high grade energy, usually electricity.

2. DECENTRALIZED VENTILATION CONCEPT

If ventilation is only used to assure a good IAQ and not for temperature control, relatively low ventilation rates are necessary and alternatives to strictly centralized concepts become an opportunity.

The decentralized supply concept described herein follows the principle of displacement ventilation and benefits from larger ventilation efficiencies compared to mixing systems (Skistad, 1994). It works similarly to classical underfloor air distribution (UFAD) systems and hence also profits from advantages of these systems (Bauman and Webster, 2001). Major differences to UFAD as commonly used in the US are the supply of fresh air from the façade with a high number of decentralized supply units and the use of a ducting network integrated in the slab construction instead of using a pressurized raised floor. The networked structure connects the different supply units such that pressure differences between different sides of the façade can be balanced.

Decentralized air handling units are more and more finding their way into modern office buildings in Europe. Especially in Germany, large projects with decentralized air supply were realized. Most air handling units used rely on small radial fans using a constant flow rate control guaranteeing the design flow rate up to pressure differences around +/- 200 Pa. In contrast to the constant flow rate control this paper presents an alternative approach that exploits the distribution means to cope with different wind pressure situations.

3. METHODS

3.1 Evaluation of Pressure Drops in Ventilation Systems

Swiss standards (SIA V382/3) require the pressure drop for ventilation systems including supply and exhaust to be below 1200 Pa with flow rates around 30 m^3/h per person recommended. At least half but rather more of the total pressure drop occurs on the supply side. Radial fans of different blade designs with belt drives are most commonly used.

For the analysis of the proposed decentralized supply system and its comparison to regular, centralized supply systems a simple square building with five floors and a floor space of 400 m^2 per floor has been defined. A specific ventilation rate of 4 m^3/(h m^2) was assumed.

To overcome the situation of lacking information concerning pressure drops in typical, centralized, mechanical ventilation systems, commercially available radial fans from different suppliers have been analyzed for a given flow rate to estimate realistic pressure drops. It has been found that suitable fans for a flow rate of 8000 m^3/h according to the previously defined reference building, have a total pressure rise around 800 Pa when working with maximum efficiency. The most efficient fans with fan efficiencies around 80% are those using backward inclined blading.

The assessment of the pressure drops in the decentralized supply system has been done using static pressure values calculated in a simulation of the supply network as described in the following.

3.2 Simulation of Air Distribution in a Decentralized Supply Network

The distribution of the flow rates in the decentralized supply system were analyzed using an adapted one dimensional, steady state formulation of the mass and energy conservation principle for compressible flows. A nodal formulation together with a damped
Newton-Raphson method was used to solve for static pressures in the network nodes. Nodes can be junctions, openings or fans. The mass flows in the pipes were calculated using a simplified equation based on an equation derived by (Osia
dacz, 1987) for isothermal flow.

\[
    p_1^2 - p_2^2 = \frac{16\lambda RTL}{\pi D_h^5} \dot{m}_n^2
\]

\(p_1, p_2\): Nodal pressures  
\(\lambda\): Dimensionless friction factor  
\(R\): Specific gas constant for air  
\(T_s\): Temperature of the supply air  
\(L\): Pipe length  
\(D_h\): Hydraulic diameter  
\(\dot{m}_n\): Mass flow evaluated at  
\(T_n=293^\circ\text{K}\) and \(p_n=1\times 10^5\) Pa

Additional assumptions of an ideal gas and zero elevation were made. For the calculation of the mass flow major and minor losses were considered. Loss coefficients for a sudden contraction and expansion as found in many textbooks such as (Wilcox, 1997) have been introduced for all nodes. Openings also include a loss coefficient for the exhaust into the room. The fans use an additional loss coefficient for the intake through a protection grating in the façade. For the fan characteristics a 2nd order polynomial has been implemented.

As Global input parameters to the simulation ambient pressure of 1e5 Pa and ambient temperature of 273 °K were used. The supply air is assumed to be at a temperature of 293 °K with a specific gas constant of 287 J/(Kg K) and a dynamic viscosity of 1.8e-5 kg/(m s).

3.3 Procedure of Analysis
The supply units assumed are similar to devices that have been developed in a collaboration between the Building Systems Group at the ETH and industry and have a nominal flow rate of 100 m³/h if no external pressure difference is applied. The coefficients of the fan characteristics are listed in Table 1. 18 fans have been used in the simulation to able to reach the design flow rate of 1600 m³/h per floor. Parameters of other components selected such as pipes and openings are shown in Table.

Table 1: Parameters of components used in simulation

<table>
<thead>
<tr>
<th>Component</th>
<th>Hydraulic diameter</th>
<th>Roughness</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pipe</td>
<td>0.118 m</td>
<td>7e-3 mm</td>
</tr>
<tr>
<td>Junction</td>
<td>0.177 m</td>
<td></td>
</tr>
<tr>
<td>Opening</td>
<td>0.177 m</td>
<td>5.39</td>
</tr>
<tr>
<td>Fan</td>
<td>0.195 m</td>
<td>10</td>
</tr>
<tr>
<td></td>
<td>-3.566e4 Pa/(m³/h)²</td>
<td></td>
</tr>
<tr>
<td></td>
<td>-0.243e4 Pa/(m³/h)</td>
<td></td>
</tr>
<tr>
<td></td>
<td>97 Pa</td>
<td></td>
</tr>
</tbody>
</table>

An appropriate topology of the air distribution network has been defined for the analysis of the air distribution. In the generation of the topology, it was avoided to have openings connected in series because it leads to an uneven distribution of flow rates between the openings. The resulting topology of the distribution network with 18 fans, 24 openings and 14 junctions is shown in Figure 1a. A very simple, unconnected topology with 18 fans and 18 openings has also been defined to be used as a reference case for the discussion of the results. This topology represents a classical way of connecting decentralized supply units and is shown in Figure 1b.

Figure 1: a. network topology, b. unconnected topology
The squares in Figure 1 represent the fans, the circles the openings and the black diamonds the junctions.

Two different cases, without and then with wind loading were studied. The same dimensionless external averaged pressure coefficients as implemented in the tool called WeatherSmart (Djebar, 2001) based on a trigonometric function developed by (Walker...
and Wilson, 1994) were used to derive pressure differences from the wind loading around the building. Pressure differences were applied at the fan locations along the façade as boundary conditions for the simulation in case of wind impact. For the evaluation of the pressure differences using the pressure coefficients, an average wind speed of 6 m/s was assumed (MeteoSchweiz, 1982-2000). A more severe wind loading with 10 m/s wind speed was also considered in the analysis. For the simulations of the air distribution an open plan situation was assumed. In the discussion of the results, a virtual separation was introduced to highlight problematic areas such as separated rooms located at the corner of the downwind side of the building. The assessment of the supply system and its effectiveness in air distribution is based on the flow rate supplied at each opening.

4. RESULTS AND DISCUSSION

4.1 Pressure Drops and Energy Savings

The largest pressure drop recorded in the distribution network for the case of no wind loading is 12.7 Pa. This is the pressure that the fans have to overcome in the supply network. Additionally to this pressure a pressure drop in the decentralized supply units of 40 Pa have to be considered. The total pressure drop in the supply system when rounded up is about 55 Pa. The small axial fans used in the supply unit have an overall efficiency depending on the operation point around 20%. In centralized systems the fan efficiency of large radial fans such as described previously is around 80%. In order to derive the total efficiency, the losses of the belt drive of typically 5% (Lexis, 2000) and the efficiency of the electrical motor of typically 90% (Lexis, 2000) have to be considered. Equating these losses and efficiencies a final total efficiency around 70% results. For the comparison of the electrical fan power consumption the 800 Pa pressure drops and the total efficiency of 70% of the centralized supply system have to be confronted with the 55 Pa and 20% efficiency of the decentralized system. In the case of using this decentralized supply system instead of a centralized one, fan power can be reduced by that factor 4.16. This leads to tremendous energy savings when cumulated over the life-span of the plant and reduces running cost of the building significantly.

4.2 Evaluation of Flow Rate Distribution

The estimated probability density functions for the distribution of the flow rates are displayed in Figure 2 and Figure 3.

![Figure 2: Flow rate distribution in the network](image)

![Figure 3: Flow rate distribution in the unconnected topology](image)

Three density curves are shown for no wind loading and for wind loading with 6 and 10 m/s wind speed. The unconnected case Figure 3 has a larger mean flow rate than the network Figure 2 because of fewer openings. The net flow for both topologies is however almost the same since there is always the same power input from the fans. The decrease of the net and the mean flow with wind depends on the fan characteristics and is hence the same for both topologies. -4.3% for 6 m/s and -14% for 10 m/s wind speed respectively. The respective net
pressures acting on the buildings envelope for those wind speed are -5.23 Pa and -14.54 Pa.

For the case of no wind loading the unconnected topology shows the most homogeneous distribution of the flow rates with a standard deviation of 0.66 m$^3$/h because the pressure drops in the pipes are very low such that only small pressure differences between the different branches occur. In the network case there are two peaks in the density function. The smaller peak to the right is due to the higher flow rates in the corners of the network. This fact is also reflected in the increased standard deviation of 1.94 m$^3$/h.

For the case of wind loading with 6 m/s wind speed the unconnected topology shows two very clearly separated peaks in the density function. This reflects the wind loading with positive pressures on the windward side of the building and negative pressures on all the others sides. A similar picture is observed for a wind loading with 10 m/s only that there is a much larger spread in between the flow rates. For the network a much more uniform picture appears for the case of wind loading. At a wind speed of 6 m/s the density function has a slightly increased standard deviation but the second peak to the right that could be observed in the case of no wind loading disappeared. This indicates that at the junctions in the corners the excess flow rates where reduced due to the negative pressure boundaries. For wind speeds of 10 m/s the flow rates are clearly spread over a larger range of values but compared to the unconnected case there is still a much more uniform picture. No distinct peaks can be observed in the network and the standard deviation is with 6.06 m$^3$/h much lower than 27.34 m$^3$/h in the unconnected case.

Other measures were used to assess the quality of distribution of the two topologies and their sensitivity to wind loadings. Figure 4 shows the relative deviations of maximum and minimum flow rates from the mean flow rate for each case. This is another measure of the spread of flow rates and expresses the quality of distribution. The slopes of the curve are a measure for the sensitivity of the system to wind loading. The ideal distribution would have only one value for the flow rate, i.e. zero deviation and a slope of zero.

As already mentioned is the spread between minimum and maximum values in the case of no wind the smallest for the unconnected topology. This changes dramatically with wind impact. A strictly larger spread for the minimum as well as the maximum values is observed for the unconnected topology. Most interesting to note is the expressed difference of deviations of the maximum values between the two topologies. The significantly lower increase of the maximum flow rates relative to the decreasing mean value again shows the effective distribution of flow rates in the network from higher toward lower values. This is also confirmed by the lower decrease of the minimum values in the network despite the fact that the mean value and the total net flow rate decreases in the same way for both topologies.

For a more severe wind loading with 10 m/s wind speed significant deviations for minimum and maximum flow rates are observed for both topologies. At that point one must recall that the simulations where all performed for a 0 Pa relative pressure boundary on the inside of the building. This means, it is assumed that the exhaust flow rate is adapted to the supply flow rate in a perfect sense such that no pressure difference due to incontinuities appear. In reality the exhaust would be constant, adjusted by coupling with the supply units or pressure controlled but with some limits to resolution. If the exhaust rate is kept constant while the supply rate decreases a negative pressure builds up in the room. This situation can be simulated by applying a negative pressure boundary condition in the inside. For this purpose a virtual
separation was introduced. It shall represent an office which is separated from the open plan situation. This separation was chosen in the most critical corner to the right in Figure 1a if it is assumed that wind is hitting the building from the left side. For that situation negative pressures occur on both sides of the façade. The separated area comprises 4 fans and 5 openings of the network. A negative pressure of -1 Pa was applied for the 5 openings lying within the virtually separated room and a wind speed of 10 m/s was assumed. The cumulated flow rate of the five openings lies only 5.5% below the value for no wind loading. When 0 Pa inside pressure is assumed as in the previous simulations the decrease of the cumulated flow rate is 18.8%. This ability of the network to redistribute the supply air between different zones with minute pressure differences is the major strength of this approach. It insures that only very small pressure gradients can build up between different rooms or zones. For the unconnected topology this of course does not hold. The only way to compensate for flow rate being to low is to fully compensate for the pressure difference applied by the wind.

5. CONCLUSIONS
As could be seen in the comparison presented, the decentralized supply system offers a possibility for tremendous energy savings in terms of fan power. A theoretical reduction of fan power by the factor 4.16 was estimated for a flow rate of 8000 m$^3$/h when compared to a centralized supply system. Concerning the air distribution it could be shown that a ducting network in contrast to a classical, unconnected topology allows a adequately regular distribution at least for relatively small wind speeds up to 6 m/s. For higher wind speeds significant deviations also in the network occur if a zero pressure boundary condition at the inside of the building is applied. For a more realistic case with slight negative pressure in the room the analysis of a virtual separation has shown that even for a wind speed of 10 m/s a local availability of the flow rate in the separated room of 94.5% could be achieved. This highlights the ability of the ducting network to distribute air with very low pressure drops which is the major strength of this network approach and differentiates it from the commonly used unconnected topologies.

6. OUTLOOK
More work is needed in the assessment of other topologies and their flow rate distribution characteristics to be able to identify key parameters for a more general topology optimization. A more in depth analysis of potential energy savings when using the wind for air transportation is needed. This includes the use of more realistic wind loadings, the analysis of suitable control strategies as well as a definition for the level of fan overcapacity needed. The overcapacity of fans increases the degree of freedom for further optimizations concerning energy savings based on outside pressures or temperatures.

REFERENCES
SIA V 382/3 (1992). Bedarfsermittlung für Lüftungstechnische Anlagen, Swiss Standards
6.6 Extended Research

6.6.1 Analysis of Different Network Topologies

Following the studies presented in the AIVC paper (section 6.5), further network topologies were considered in order to determine their quality in terms of air distribution. As it was found previously, only closed looped networks are suitable to satisfy the goal for a regular distribution of the supply air. Similar to the study in the paper the new networks were defined for the building test case described in section 6.4.5 with 18 pumps being used, 5 each on the windward and the opposite side and 4 each on the other sides. The analysis was carried out under two different conditions, once without wind and once with a wind speed of $10 \, \text{m/s}$. The wind hits the building perpendicularly at a side with 5 pumps being installed.

Network Generation

The ADNs considered in the analysis presented herein are collected in Table 6.2. The procedure of generating appropriate networks was as follows. First, networks only using pumps and openings were considered. The most straightforward network topology in that case is a regular grid. The grid is obtained by simply connecting the pumps from opposite sides. In every crossing of two pipes an opening is placed. For the generation of the second network topology the condition that two pipes have to leave from each pump was imposed. The idea behind this condition is to split the flow rate supplied by the pump at a very early stage such that velocities are low already at the source. It then was tried again to define a regular grid. This was achieved by diagonally connecting pumps with each other. Again where crossings occurred openings were placed. Similar to the previous grid solution an exception occurs in the middle where a solid core is located. In this case also at the corners a different connection pattern had to be chosen. Departing from this diagonal grid solution with 2 pipes connected to each pump a network with 3 pipes leaving each pump was defined. This was achieved by connecting the pumps to a third opening that was placed between the two openings the pumps are already connected to. This opening is then further connected to an opening lying in the second row in direction of the inside of the space. In a second step, further networks including junctions beside the openings and pumps used so far, were generated. Based on the network solution with two pipes leaving each pump, two different alternative networks were deduced. The first one was created by replacing all openings lying on an inner ring around the core.
Table 6.2: Overview of network topologies studied including a completely unconnected topology, and several mesh topologies with either only opening or openings and junctions combined.

<table>
<thead>
<tr>
<th>unconnected</th>
<th>rectangular grid</th>
<th>2perpump</th>
</tr>
</thead>
<tbody>
<tr>
<td><img src="image1" alt="unconnected" /></td>
<td><img src="image2" alt="rectangular_grid" /></td>
<td><img src="image3" alt="2perpump" /></td>
</tr>
<tr>
<td>3perpump</td>
<td>2perpump_junction_1</td>
<td>2perpump_junction_2</td>
</tr>
<tr>
<td><img src="image4" alt="3perpump" /></td>
<td><img src="image5" alt="2perpump_junction_1" /></td>
<td><img src="image6" alt="2perpump_junction_2" /></td>
</tr>
<tr>
<td>brick_junction</td>
<td>grid_junction</td>
<td>network_AIVC</td>
</tr>
<tr>
<td><img src="image7" alt="brick_junction" /></td>
<td><img src="image8" alt="grid_junction" /></td>
<td><img src="image9" alt="network_AIVC" /></td>
</tr>
</tbody>
</table>
by junctions. The second alternative was generated replacing all openings along every other diagonal line across the space with junctions. The main idea behind introducing junctions is to bring air more directly to the core and to avoid indirect path via openings. Junctions are seen as redistribution points in the network. Another network was defined that looks like a brick wall. The network has several concentric rings going around the core. The rings are connected to each other by bridging pipes that run perpendicularly to the rings and the side faces of the building. The bridges between the different rings are not in line but alternately shifted sideways such that it gives the network the appearance of a brick wall. At the intersections of the rings with the bridging pipes junctions were placed and openings are distributed on the rings between two junctions. The last network that was generated also started with a single pipe going straight from the pump to a junction. From that junction two pipe are leaving sideways to be connected to openings and one is going straight to the inside of the space and is connected to a junction. From that junction the pattern is continued that way until the core is reached and the last straight pipe is closed with an opening. The network elements with its respective occurrence for the different networks of Table 6.2 are displayed in Table 6.3. The empty squares in the figures showing the networks represent pumps, the circles are openings and the black diamond shapes are junctions. The solid black lines are the pipes. One section in each network is drawn in red. The red part is showing the fundamental connection pattern the network is based on. However, none of the network is pure in the sense that the connection pattern cannot be repeated endlessly through the network but always special situations at the boundaries or in the joint region of two sides occur where the topological pattern has to be slightly altered.

**Quality Measures**

The criteria that qualify an ADS have been introduced. For the assessment of the quality of an ADN appropriate indicators are required and yet have to be defined. Any ADS has to work well under different conditions such as the uneven pressure distribution on the building due to the influence of wind. As a consequence there are quality measures that are applied independently for each condition and some others that are based on the transitional behavior of the system between the two conditions.

The independent measures used were the total flow rate, the standard deviation of the distribution of flow rates and the relative deviation of the maximum or the minimum flow rate from the mean flow rate. The total flow rate is largest for low pressure drops which is directly influenced by the number
Table 6.3: Different network topologies considered in the analysis with indication of the number of network elements used

<table>
<thead>
<tr>
<th>networks</th>
<th># pumps</th>
<th># openings</th>
<th># junctions</th>
<th># pipes</th>
</tr>
</thead>
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<tr>
<td>unconnected</td>
<td>18</td>
<td>18</td>
<td>0</td>
<td>18</td>
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<tr>
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<td>28</td>
<td>0</td>
<td>62</td>
</tr>
<tr>
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<td>18</td>
<td>34</td>
<td>0</td>
<td>80</td>
</tr>
<tr>
<td>3perpump</td>
<td>18</td>
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<td>0</td>
<td>112</td>
</tr>
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<td>2perpump_junction_1</td>
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<td>26</td>
<td>8</td>
<td>80</td>
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<td>2perpump_junction_2</td>
<td>18</td>
<td>20</td>
<td>14</td>
<td>80</td>
</tr>
<tr>
<td>brick_junction</td>
<td>18</td>
<td>40</td>
<td>38</td>
<td>110</td>
</tr>
<tr>
<td>grid_junction</td>
<td>18</td>
<td>30</td>
<td>24</td>
<td>96</td>
</tr>
<tr>
<td>network_AIVC</td>
<td>18</td>
<td>24</td>
<td>14</td>
<td>74</td>
</tr>
</tbody>
</table>

of openings introduced into the space. More complex networks with a large number of openings have a lower flow resistance. The losses in the pipes also influence the resulting flow rate but usually less significantly than the number of openings. The standard deviation is a measure of the spread of the flow rates around its mean value. Low values indicate the wished, regular distribution. This value should be low both, under influence of wind or for no wind. The standard deviation is a weighted average of deviations from the mean and therefore it is only weakly affected by rare, extreme events. The deviation of max and min relative to the mean is another measure of the evenness of flow rate distribution. Because it addresses deviations on both sides from the mean it is also a measure for the symmetry of the distribution. Unlike the standard deviation it is calculated using the extrema only and hence is a direct measure for the presence of such extreme flow rates even if they rarely appear.

Combined measures including both, the condition with no wind and the impact of wind with a wind speed of $10 \, \text{m/s}$ are the spread of the minimum or maximum flow rates between the two conditions. Ideally, the network under impact of wind would show a decrease of maximum and an increase of minimum flow rates. This happens if the network is able to smooth out the extrema by redistribution of the flow rates. Because the net pressure due to wind impact is usually negative the mean flow rate decreases. Therefore the minima can only increase if the air is effectively redistributed. If no redistribution takes place both extrema get more expressed under the influence of wind.

A third group of measures applied were topological measures. Here, two different topology measures being at the same time cost measures have been
used; the number of nodes installed in the network and the network density. The density is the ratio between the cumulated link length in the network and the area served by the network [27]. Other measures known as indices for connectivity were also introduced but they showed to have a perfect correlation with the density such they did not appear to give any additional information. For this reason the connectivity indices were omitted.

Network Evaluation

The distribution of supply air has been simulated for all the networks shown in Table 6.2 for the situations without and with wind. The results of the distribution are shown in Figure 6.25 with some numbers listed in Table 6.4. The curves in Figure 6.25 represent the probability density functions of the flow rate distribution for different networks and wind conditions. For the sake of clarity in Figure 6.26 the same density distributions are again represented but separated into groups of three different topologies per figure. For a given color and marker type the thin line represents the situation with no wind and the bold line the one with wind. The various curves are shifted along the x-axis due to different mean values given by the different number of openings used. Some networks show a very peaked distribution density function which tells that some flow rates appear very often in the network. Others have a rather wide distribution density but show little reaction to a change in wind conditions. For a systematic evaluation of the distribution properties of these different networks the quality indicators introduced have to be used. These indicators are representative for the distribution patterns shown in Figure 6.25. The evaluation of the different networks was performed normalizing the values for each quality measure such that 0 was the minimum, and 1 the maximum value. For each criterion a different weight was picked. The weighting was chosen from a range between 1 and 4. The cost function used is then simply the weighted sum. Dividing the sum of values by the sum of the weights gives an average which again ranges from 0 to 1. The winner of the evaluation is the one that minimizes the cost function.

The total flow rate was given a weight of 2. Variations there are relatively low and since it is primarily determined by the number of openings rather than the connection pattern chosen, this aspect should not be over-weighted. The weight of 2 was chosen because in comprises physical aspects such as the pressure drop in the system and the minimum value of 1 is only chosen for monetary aspects such as the total number of all types of nodes and the network density which is the total length of pipes installed. The reason for the low weights for monetary aspects chosen in this evaluation is the selected
Table 6.4: Different network topologies with simulation results for wind/no wind condition

<table>
<thead>
<tr>
<th>topology</th>
<th>Q total</th>
<th>mean</th>
<th>min</th>
<th>max</th>
<th>std</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>wind speed 0 m/s</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>unconnected</td>
<td>1.5746e03</td>
<td>87.476</td>
<td>86.006</td>
<td>88.235</td>
<td>0.662</td>
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<tr>
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<td>1.6473e03</td>
<td>58.833</td>
<td>53.328</td>
<td>62.404</td>
<td>3.555</td>
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<td>1.6907e03</td>
<td>49.725</td>
<td>42.161</td>
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<td>7.096</td>
</tr>
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<td>1.7218e03</td>
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<td>30.876</td>
<td>46.116</td>
<td>4.291</td>
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<td>1.6690e03</td>
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<td>59.594</td>
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<td>3.945</td>
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<td>3.391</td>
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<td>9.904</td>
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<td>7.1429</td>
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<td>21.091</td>
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<td>57.554</td>
<td>50.373</td>
<td>68.291</td>
<td>6.0645</td>
</tr>
</tbody>
</table>
focus on the distribution aspect of such networks. The maximum weight of 4 is given to the standard deviation because it nicely quantifies the symmetrical average evenness of a distribution network. It is a necessary condition for a valuable solution to have a low standard deviation for the distribution for both wind and no wind. Because the standard deviation only accounts for the weighted average the effective deviation of the extrema from the mean have to be considered. Because both, negative and positive deviations from the mean can lead to comfort problems a symmetrical weight of 3 is chosen for both deviation measures. Assuming the same requirements are valid all the time the weighting is kept constants for both wind conditions. Finally the spread of the minima and the maxima between the different wind conditions was weighted with a 2. This is justified by the fact that this criterion can be misleading in qualifying solutions that appear to be insensitive by just looking at the extrema and because it is slightly redundant. Looking at the sensitivity of the parameters it turn out that it is rather low. The evaluation shows that four topologies are among the bests almost independent of the weights chosen. These are the rectangular grid, the 2perpump_junction_1, the 2perpump_junction_2 and the network_AIVC as it can be seen in Figure 6.27 for a weighting as previously described.

The most obvious losers of the evaluation as readily verified in Figure 6.27 are the networks brick_junction and unconnected. Although the unconnected
Figure 6.26: Estimated flow rate densities for different groups of networks
Figure 6.27: Ranking of different network solutions

topology performs best under no wind conditions it performs poorest under
the influence of wind. No interconnections between branches are present that
could balance some of the pressure difference. The network brick junction has
the highest monetary costs but still does not perform well. Because of the large
number of junctions and openings the redistribution of flow rates along a path
from one side of the building to the other becomes very bad. Also is the feeding
of the zones near the core with fresh air very poor because of the back pressure
in the junctions being too high such that the air gets immediately released
through the openings closest to the pumps. The next solution in the ranking
although performing much better than the previous two is the 2perpump. This
network shows a relatively large standard deviation and spread between the
extrema even for the condition of no wind. The main reason for this are
the anomalies in the corners where one opening is always connected to two
pump branches and no other openings and hence has a large flow rate. Also
the neighboring openings have a over-proportionally large flow rate such that
the weight is large enough to lead to a high standard deviation. On the other
hand this network solution shows a small change in the standard deviation
and the spread of extrema switching to a wind situation. The maxima in the
corners even decrease because of positive and negative pressure joining there
and average out the excess supply. Some redistribution of the flow rates in
the network can be observed through the change of flow directions under the
influence of wind. Following the 2perpump network performing slightly better,
there are the networks grid\_junction and 3perpump. The network 3perpump is able to reduce the standard deviation compared to the 2perpump by making use of the additional links to balance out pressure difference that already exist under the condition of no wind. An example are the excess flow rates in the corner that are reduced. Again, similar to the 2perpump network the shape of the distribution density function for the 3perpump network does hardly change for a change in wind condition. The network grid\_junction again has a larger standard deviation but because of the junctions installed the air supply to the core and its distribution in the inner zone is quite homogeneous. The large deviations between different openings is due to the fact, that too much air is already released through the openings lying closest to the outside boundary. Because these openings are not directly but only indirectly via junctions connected to the inside the excess flow rate through these openings is even more expressed under the influence of wind. This fact also confirms the relatively rigid pressure distribution found in this topology that determines the routing of the air flows. For this solution no change in routing, i.e. no effective redistribution could be observed.

The remaining four networks are very close to each other in the ranking and show significantly lower values of the cost function. The lowest total costs has the network 2perpump\_junction\_2 followed by network\_AIVC and by the networks rectangular grid and 2perpump\_junction\_1 having the same costs. Looking at the two networks that are derived from the 2perpump network the standard deviation was significantly decreased. In the first network modification shown in Table 6.2 row 2, column 2, junctions are placed symmetrically around the core. These junctions allow to maintain the pressure such that the openings in the inner most ring of the network are sufficiently supplied as can be seen in Figure 6.28 showing the flow rate distribution. This leads to only a very small drop in quantity from the outside toward the core. The routing in the network remains unchanged from the one observed for the 2perpump solution. In the second network modification, network 2perpump\_junction\_2, shown in Table 6.2 row 2, column 3, the junctions are arranged along three diagonals starting bottom left going to top right. This distribution introduces an additional asymmetry to the network. This is also reflected in the routing shown in 6.29 that now has changed with respect to the 2perpump network. Similar to the network 2perpump\_junction\_1 the supply of the core was improved by introducing the junctions. In the second modification however a larger number of junctions was installed such that the number of openings is smaller and hence the flow rate per opening larger. This modification increases the back pressure in the system which can be a benefit in terms of redistribution. Because the throttle curve of the opening is steeper (higher change of
pressure loss with a change of flow rate) than the one in the pipes, for large supply pressures at the pressure side of the building, redistribution of the air to locations of lower pressure in the ducting network becomes more probable. For the situation with a wind speed of $10 \frac{m}{s}$ the pressure in the pump nodes on the pressure side for the networks 2perpump_junction_1 and 2 are 6 and 9 Pa respectively. Because very low pressure drops occur in the ducting this pressure difference is sufficient to trigger the redistribution. The very straight forward rectangular grid solution at row 1, column 2 of Table 6.2 shows a good distribution of flow rates for the case of no wind load. Only slightly higher flow rates are observed in the outer ring closest to the pumps compared to the ones near the core. In case of wind standing on the building the flow rates on the pressure side of the outer ring does not increase because of the direct and straight connection of the openings from the outside to the inside. The pressure drop in these connecting branches is very low such that the back pressure encountered by the redistributing flow is low and no excess flow rate for the openings upstream result. While the maximum flow rates in the network stay constant the minima decreases similarly to the mean. The flow rates among openings other than those on the pressure side show a very regular distribution. This is because the negative pressure side is sufficiently supported by
Figure 6.29: Air flow distribution simulated for the network 2pump_junction_2

the pressure side. This behavior of the network can also be verified by the change of the routing of airflow shown in 6.30. The last out of the best four networks is the network_AIVC shown at row 3, column 3 of Table 6.2. Its basic connection pattern is similar to the one chosen for the grid_junction network with the difference that it is not repeated over two steps. A junction being fed from a pump is directly connected to three openings and to no other junctions. The excess flow rates in the corners are buffered by direct connections between openings. This solution has a very small standard deviation under no wind conditions but it significantly increases above the value of the other three networks when wind is present. The reason for that can be found in the limited possibilities for the air to be redistributed. Similar to the network grid_junction there is a relatively rigid pressure distribution in the network that determines the routing of the air shown in Figure 6.31. This similarity in behavior between the two networks is obviously determined by the same connection pattern they share.

A table with the values of the cost function as used for the network evaluation is shown in the Appendix. Also in the Appendix are the figures representing the air flow distribution in the networks that have not been shown here.
Figure 6.30: Air flow distribution simulated for the network *rectangular grid*

Figure 6.31: Air flow distribution simulated for the network *network_AIVC*
6.6.2 Exergy Analysis For Decentralized Supply Units

Similar to the exergy analysis presented for an exhaust system in Chapter 5 the exergy analysis for a supply system is treated here. Figure 6.32 shows a generic supply system. So far it represents an arbitrary supply system independent of being centralized or decentralized. The components comprising the supply system are a heat exchanger, a fan unit and a throttle representing pressure losses occurring in the supply system. Environmental conditions are denoted with a 0 and local inlet conditions which might differ from the freestream conditions are denoted with an \(i\). The inlet air is heated or cooled by the heat exchanger before it proceeds to the fan at pre-fan conditions \(p\). In the fan the air is pressurized and brought to after-fan conditions \(a\). Over the throttle a pressure drop occurs and the air is expanded to room conditions \(r\). The same components comprising a supply system are also found in an exhaust system only that the arrangement of the components is different. While the throttle and the fan are treated the same way as in the exergy analysis presented in Chapter 5, the heat exchanger used in the supply system is treated differently from the heat exchanger used in the exhaust system for heat recovery. For the heat recovery coupled with a heat pump a constant temperature close to the evaporation temperature of the heat pump where the heat transfer was chosen. This led to the formulation of the exergy flux coupled to the heat flux using the Carnot factor. In the case of the supply system the entropy and the exergy flux on the water side is calculated based on supply and return temperatures of the water. Expressing the heat flux as a function of infinitesimal temperature steps, integrating it, leads to a logarithmic formulation of the entropy as shown in eq. 6.35.

\[
\dot{S}_a - \dot{S}_0 = \int_0^a \frac{\delta Q}{\delta t} \frac{dT}{T} = \dot{m}c_p \int_0^a \frac{dT}{T} = \dot{m}c_p \ln \left( \frac{T_a}{T_0} \right) 
\]

The exergy consumed in the heat exchanger is calculated by taking the difference between the exergy inputs and outputs. Following, the formulas for the
The air temperatures \( T_i \) and \( T_p \) across the heat exchanger are freely chosen. Based on these temperatures the heat demand is evaluated and for a given supply temperature \( T_s \) of the water the return temperature \( T_r \) is calculated. The pressure drop of the air flow through the heat exchanger is neglected such that \( p_i \) is the same as \( p_p \). All pressure drops are concentrated to the throttle defined. Because the supply air is neither de- or humidified and condensation is avoided the chemical exergy does not change in the system. The chemical exergy is only transiting the components. Making the assumption that the effective dead state and the environmental condition coincide, the outside air has zero chemical exergy and hence it must not be considered in the flow exergy. As a consequence the third term on the right hand side of equations 6.36 and 6.37 vanishes. The moisture in the air only affects the mass flow, the specific heats and the ideal gas constant.
Evaluation of Exergy Losses

The exergy losses over the components of an AHU were evaluated for both the heating and the cooling case. The ambient conditions assumed are the same as the ones used in the calculation of the exergy losses for the exhaust plant in Chapter 5. During heating season ambient temperature is at 0 degrees Celsius and relative humidity at 100 %. During cooling season ambient temperature is at 28 degrees Celsius and relative humidity at 40 %. The ambient pressure is taken to be 1e5 Pa. For both operation modes heating and cooling the supply temperature of the air is assumed to be identical with a temperature of 20 degrees Celsius. The flow rate handled by a AHU is 95 m$^3$/h at a system pressure the fans have to overcome of 55 Pa. For this scenario the fans that are operated with an over-all efficiency of 20% have an electrical power usage of 11 W. The water fed to the heat exchanger of the AHU has a constant flow rate of 0.135 m$^3$/h. For the calculations in heating mode a water supply temperature

![Figure 6.33: Exergy losses across the stages of a decentralized supply system for the heating case](image)

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of 28 degrees Celsius is assumed. The return temperature is calculated based on the heat flux required to reach a supply air temperature of 20 degrees Celsius. The exergy losses occurring over the different stages during heating are shown in Figure 6.33. The largest losses occur in the heat exchanger and are due to the heat transfer across a finite temperature difference between the water and the air. The design of the heat exchanger, i.e. the surface area, the materials used, etc. have an effect on the temperature difference during heat transfer and hence on the resulting exergy losses. Exergy losses in the heat exchanger were calculated to be 33 W. The losses in the fan 9 W and those in the throttle 1.5 W. The pressure difference that is lost over the throttle is the same as the one that has to be built up by the fan. Because the pressure difference with 55 Pa is very low, only small exergy losses are associated with this pressure drop in the throttle. The much larger exergy losses over the fan are mostly determined by the very low over-all efficiency of 20% of the small fans used in the AHU. The partitioning of the exergy losses among the components is visualized using a pie diagram shown in Figure 6.34.

![Pie diagram showing exergy losses](image)

Figure 6.34: Distribution of exergy losses among different stages for the heating case

When the AHU is running in cooling mode the hot outside air has to be brought down to room condition. The water supply temperature for this condition is chosen to be at 18 degrees Celsius. The exergy losses calculated for the AHU running in cooling mode are shown in Figure 6.35. The exergy
Figure 6.35: Exergy losses across the stages of a decentralized supply system for the cooling case

losses occurring in the heat exchanger during cooling being 4.15 W, are now small compared to the heating case. This is due to the fact that the spread between room temperature and outside temperature during summer is much smaller and so is the heat load. Also, is the temperature spread between the outside air and the supply water much smaller such that exergy losses due to irreversible heat transfer over a finite temperature difference are much smaller too. The exergy losses in the fan which become the most expressed exergy losses in the AHU during cooling mode are 9.8 W and hence slightly above the fan losses in the heating case. The pressure drop is the same in both cases but the temperature levels are different such that density and hence the mass flow given the volumetric flow rate is slightly altered. For higher outside temperatures the exergy losses in the heat exchanger would similarly to the heating case again be larger than the losses occurring over the fan. The exergy losses in the throttle are with 1.5 W the same as in the heating case and accordingly small because of the small pressure differences involved in the
system. The significance of the exergy losses for the individual components of an AHU is again visualized in the pie diagram shown in Figure 6.36.

Figure 6.36: Distribution of exergy losses among different stages for the cooling case

The exergy analysis and its application to decentralized AHUs as shown here was used in the paper of PLEA 2008 in order to estimate the exergy savings depending on different inlet conditions and control decisions. While some savings of high grade energy can be evaluated using a simple energy balance for some a more elaborate analysis using the concept of exergy is required. The potential savings that can arise from using a decentralized supply system as the one described in this work are highlighted in subsection 6.8.1.

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584: Advanced Decentralized Ventilation: How Wind Pressure Can Be Used to Improve System Performance and Energy Efficiency

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Abstract

The current paper addresses the problem of existing, highly inefficient ventilation systems with large pressure losses and proposes a new ventilation concept that is in accordance with the lowEx principle. The ventilation system consists of decentralized supply air units and a highly interlaced ducting network. Not only does the conception of this system lead to minimal pressure losses but also allows the wind forces acting on the building to be used for air transportation. Making use of the wind adds additional value to decentralized concepts in that it significantly improves the ventilation performance and reduces the energy costs.

In order to assess the air distribution in the ducting network and the ventilation performance a one dimensional steady state analysis was used. The analysis was carried out for a virtual building with a total ventilation rate of 8000 m³/h. The steady state analysis showed that in comparison to typical centralized supply system the decentralized system lowers the total fan power demand by a factor larger than 4. Using a 40% overcapacity of fans in the decentralized system leads to a further reduction of fan power demand by almost 25% for the case of wind loading with a wind speed of 10 m/s. The overcapacity of the fans can also be used to decrease cooling load in summer and heating load in winter.

Keywords: exergy, lowEx, decentralized ventilation, hybrid ventilation

1. Introduction

There is a strong need to mitigate anthropogenic emissions of greenhouse gases at present and in the near future. The building sector is a large contributor to the problem with a 38% share of the total primary energy consumption and 34% of the total CO2 emissions world-wide [1]. Buildings have a life span around 50 years such that the cumulated energy consumption for the operation of the building is much more significant than for construction. It is therefore of utmost importance to improve HVAC equipment and reduce the CO2 emission and energy consumption of buildings during operation. An analysis for 13 industrialized countries presented by [2] has shown that space heating has a 61% share on the end use of energy in non-industrial buildings. 53% of this energy is lost through air change. If the average air change was reduced to minimum necessary to insure a good indoor air quality (IAQ) these losses could be reduced to a third of the current level [2]. It is therefore crucial to improve ventilation systems such that good IAQ can be achieved with a minimal energy input. Large saving potentials are available and known as confirmed by [3]. A 70% reduction due to heat recovery, 50% due to effective distribution and 50% due to demand controlled ventilation are possible. Further savings are possible with hybrid ventilation approaches and new control strategies.

2. A Novel Decentralized Ventilation

The ventilation concept presented in this paper is based on a decentralized approach. Instead of having a single supply branch as in a centralized ventilation system, the air is supplied through many fan units arranged along the periphery of the buildings. Decentralized systems have recently gained in popularity especially in Europe. Major benefits are lower pressure drops due to the short transportation distances, space savings and lower construction costs. Balanced, centralized systems are usually not sensitive to varying pressures and temperatures around the building as long as the air intake is placed in an nearly undisturbed region. Conversely, decentralized systems are directly exposed to locally varying pressure and temperature conditions. Increased cooling loads for decentralized ventilation systems due to high temperatures in the vicinity of the façade are usually compensated by lower heating loads in winter due to the same reason [4].

Large projects with decentralized supply systems have been realized in Germany where examples are the Post Tower in Bonn or the Skyline Tower in Munich. In both projects a double skin façade design was chosen to shield the decentralized supply units from direct wind or sun impact. In the Post Tower a highly elaborated, active façade system is used to regulate temperature and pressure distribution in the cavity between the outer and the inner building shell [5, 6]. The supply units are either integrated in the façade or
in a raised floor construction and directly supply the conditioned air to the room on the spot. Unlike these common decentralized supply systems, the system analyzed herein uses a strongly interlaced ducting network to distribute the air over the entire floor space. This network is ideally fed by a large number of supply units and has a sufficiently large capacity. When these two criteria are met, small air flow velocities appear within the network and a nearly homogeneous static pressure distribution can be achieved. Openings can then be arbitrarily distributed across the entire floor space and ensure a sufficient flow of fresh air even to the core of the building. In reality the pressure distribution and the routing in the network are also dependent on the topology chosen. A mesh, in contrast to hierarchical topologies, allows an even distribution of air flows with no need for using large final resistances by throttling the outlets.

The typical control strategy used for decentralized supply is a constant flow rate control. Throttling is used to control the flow rate in case of positive pressure acting on the building. In case of negative wind pressures the flow rate decreases because of the fan’s flow rate characteristics unless the fan power is increased. Consequently there are two concurring objectives: Keeping flow rate constant and minimizing power usage of the fans. Allowing the flow rate to vary leads to pressure differences between separated rooms. Conversely, the resisting strategy increases electricity demand for the fans. To resolve this conflict this paper suggests a hybrid approach which tries, besides the mechanical power input, to use wind pressure to enhance air transportation where possible and to cope with non homogeneous flow rates by distribution means. By installing an overcapacity of fans, the redundancy of the supply system is increased and so is the degree of freedom for optimization. A similar ideology can be found in the concept of a hybrid rooftop turbine ventilator presented by [7]. By adding a rooftop ventilator to the classically used DC fan, redundancy of the system is increased and allows for an optimization of the performance with minimal energy input. Using a hybrid approach for the supply offers the possibility not only to optimize fan power based on pressure boundaries but also to minimize heating and cooling loads based on temperature selective intake of the air. This potential for energy savings will be discussed in this paper based on a 1D steady state analysis of the air distribution. In addition to the first law analysis an estimate of the exergy savings that come along with the energy savings is presented. The exergy losses, i.e. the entropy production due to internal irreversibilities is also addressed. Exergy expresses the high valued part of energy that has the ability to do mechanical work. The minimization of the exergy input for the operation of buildings is hence most important because it minimizes the demand of high grade energy such as electricity or fossil fuels and thereby the emission of greenhouse gases.

3. Method

3.1 Steady State Simulation of airflows

The distribution of the flow rates in the decentralized supply system were analyzed using an adapted one dimensional, steady state formulation of the mass and energy conservation principle for compressible flows. A nodal formulation was used to solve for static pressures in the network nodes and the routing in the network are also dependent on the topology chosen. A previous study [9] analyzed the air distribution means. By installing an overcapacity of fans, the redundancy of the supply system is increased and so is the degree of freedom for optimization. A similar ideology can be found in the concept of a hybrid rooftop turbine ventilator presented by [7]. By adding a rooftop ventilator to the classically used DC fan, redundancy of the system is increased and allows for an optimization of the performance with minimal energy input. Using a hybrid approach for the supply offers the possibility not only to optimize fan power based on pressure boundaries but also to minimize heating and cooling loads based on temperature selective intake of the air. This potential for energy savings will be discussed in this paper based on a 1D steady state analysis of the air distribution. In addition to the first law analysis an estimate of the exergy savings that come along with the energy savings is presented. The exergy losses, i.e. the entropy production due to internal irreversibilities is also addressed. Exergy expresses the high valued part of energy that has the ability to do mechanical work. The minimization of the exergy input for the operation of buildings is hence most important because it minimizes the demand of high grade energy such as electricity or fossil fuels and thereby the emission of greenhouse gases.

\[ p_1^2 - p_2^2 = \frac{16 \lambda R T_s L}{\pi D_h^5} m_n^2 \]  

(1)

Additional assumptions of an ideal gas and zero elevation were made. For the calculation of the mass flow major and minor losses were considered. Openings also include a loss coefficient for the exhaust into the room. The fans use an additional loss coefficient for the intake through a protection grating in the façade. For the fan characteristics a 2nd order polynomial has been implemented.

As Global input parameters to the simulation ambient pressure of 1e5 Pa and ambient temperature of 273 °K were used. The supply air is assumed to be at a temperature of 293 °K with a specific gas constant of 287 J/(Kg K) and a dynamic viscosity of 1.8e-5 kg/(m s).

3.2 Air Distribution Analysis

A previous study [9] analyzed the air distribution in the floor space for two different topologies: A simple unconnected Fig. 1a and a more complex network topology Fig. 1b. The unconnected topology represents a classical, decentralized supply system. The squares in Fig. 1 represent the fans, the circles the openings and the black diamonds the junctions.

![Fig. 1: a) Unconnected topology with 18 fans, b) Network topology with 18 fans](image-url)
The network topology was generated using different types of network nodes; junctions and openings. Having several openings to be connected in series with the fans was avoided. For that purpose junctions were introduced from where airflows are further distributed to the openings as shown in Fig.2.

![Graph](image_url)

Fig. 2: Connection pattern of fans, openings and junctions

Both setups were studied for a simple square building with five floors and a floor space of 400 m$^2$ per floor. A specific ventilation rate of 4 m$^3/(h \cdot m^2)$ was assumed. 18 fans were placed along the periphery in order to satisfy the design flow rate of 1600 m$^3/h$ per floor. The exhaust air was assumed to be ideally controlled to match the supply flow rate and maintain a constant room pressure. Two cases were differentiated; no wind and wind loading perpendicular to the left sides of Fig. 1a and 1b.

### 3.2 Analysis of Energy Savings

Based on the above network topology a new network with 26 instead of 18 fans was defined to study potential energy savings possible with this 40% overcapacity of fans. Some of the fans were selectively switched off to reach the same design flow rate as in the previous study. Similar to the previous analysis a wind loading was defined. The wind speed was selected based on annually wind statistics for Zurich, Switzerland [10]. According to the statistics, values above 6 m/s are probable. A wind speed of 10 m/s was selected. Wind pressures around the building largely depend on the topological circumstances, the orientation and the geometry of the building. Accordingly, the pressure distribution varies strongly along the building envelope and for different heights. For the virtual building defined in this conceptual study it was hence not possible and not of primary interest to define a precise pressure distribution. Instead a simple correlation from [11] was used with averaged pressure coefficients for high rise buildings from Part 4 of the 1995 NBC as described in [12]. Only one situation with wind hitting the building perpendicularly was considered because it is the most critical case in terms of negative surface pressures. Based on the simulation results for the network under influence of wind, fan power was estimated. The electrical fan power for a fan unit was assumed to be 20 W at full rpm according to [13]. Along a fan curve of constant rpm these fans showed to have an almost constant electrical power demand. The power was then scaled according to the rated rpm using the proportionality laws known for fans [14]. For simplicity a constant total fan efficiency of 20% was used throughout the calculations. The electrical fan power demand of the network was compared to a system with constant flow rate control. Positive pressures were ignored because they are controlled by throttling while for the negative pressures the necessary additional power input was expressed using Equation 2.

$$\Delta W_{el,fan} = \frac{\Delta p \cdot Q}{\eta}$$  \hspace{2cm} (2)

Where $\eta$ is the total fan efficiency, $\Delta p$ is the pressure difference and $Q$ dot is the volumetric flow rate. Energy savings due to temperature selective control were also looked at. The reductions in cooling load were calculated according to Equation 3, where $n$ is the number of fans that take in air at a lower temperature $T_{w,2}$ instead of the temperature $T_{w,1}$. The mass flow of one of 18 fans that are simultaneously switched on is expressed by $m$ dot and $c_p$ is the specific heat capacity of air for constant pressure.

$$\Delta Q_{cooling} = n m c_p (T_{in,1} - T_{in,2})$$  \hspace{2cm} (3)

### 3.3 Exergy Analysis

Based on the exergy concept the degradation of high grade energy in the supply air process was analyzed. For this purpose an exergy balance for moist air was applied to the heat exchanger and the fan of the decentralized supply units under consideration. Similar to the analysis of energy savings presented here the influence of variable outside pressure and temperature on the exergy demand was analyzed.

![Diagram](image_url)

Fig. 3: Grassmann Diagram with exergy flows for the heating case with homogeneous pressure and a local inlet temperature of 12°K above environmental level

The exergy fluxes for the heat exchanger and the fan of a decentralized supply unit are represented in Fig.3 by the Grassman diagram. For the heat exchanger the exergy input originates from the warm/cold water supplied and from the incoming air stream depending on its local properties relative to the ambient condition. The exergy outputs comprise the return water and the heated/cooled air. Exergy destruction appears in the course of heat transfer over a finite
temperature difference. This losses plus the actual exergy load to heat/cool the air constitute the exergy demand of the heat exchanger and must be compensated by the heat generation system. The conditioned air stream is passed to the fan where exergy as electricity is fed and again losses due to entropy generation occur. These fan exergy losses occur due to friction, mixing, etc. and increase with increasing pressure difference applied to the fan. The exergy demand of the fan is basically equal to the electrical power supplied if the exergy flow of the warm/cold air stream is not taken into account and considered as transiting exergy.

As a reference state for the exergy calculation the properties of the external environment were assumed. In the heating case the environment takes a temperature of 273°K, an absolute pressure of 1e5 Pa and a relative humidity of 100%. For the cooling case the same pressure but an environmental temperature of 301°K and a relative humidity of 60% was assumed. For simplicity the state of unrestricted equilibrium with the environment [15] was chosen to be the dead state. Chemical potentials due to the moisture in the air are hence neglected. This simplification only influences the exergy analysis in that it alters the absolute exergy value. The moisture content in the air is accounted for in the calculation of the specific heats and the densities following the rules of psychrometrics. The humid air is treated as a mixture of ideal gases made of air and water vapor. The flow exergy of humid air per kg of dry air [16] writes out as:

\[
\dot{e}_{x,\text{mix}} = (c_{p,\text{dry, air}} + \omega c_{p,\text{vapor}}) \nabla \left( T - T_0 \right) + T_0 \left( 1 + \tilde{\omega} \right) R_{\text{dry, air}} \ln \frac{p}{p_0} \tag{4}
\]

where \( \omega \) is the specific humidity expressed as a mass ratio and \( \tilde{\omega} \) tilde is the mole fraction ratio between dry air and water vapor. \( T_0 \) is the ambient reference temperature and \( R \) is the specific gas constant for dry air. The specific heats \( c_p \) are expressed for dry air and water vapor.

The flow exergy of the water feeding the heat exchanger is:

\[
\dot{e}_{x,\text{water}} = c_{p,\text{water}} \left( T - T_0 \right) \tag{5}
\]

The exergy balance for any open system is then simply:

\[
\sum m_{i,j,\text{in}} e_{x,j,\text{in}} - \dot{E}_{x,\text{consumed}} = \sum m_{j,\text{out}} e_{x,j,\text{out}} \tag{6}
\]

with \( m \) dot being the in and outgoing mass flows and \( e_x \) dot their respective flow exergies. \( E_{x,\text{consumed}} \) dot is the consumed or irreversibly lost exergy flow. The heat exchanger assumed in the energy and exergy calculations has following properties. It has an area of 1.376 m² and is layed out for a volumetric flow rate of the water of 0.135 m³/h using constant supply temperatures of 301°K in the heating and 291°K in the cooling mode. The return temperature was adapted in the calculations to reach a constant supply air temperature of 293°K.

4. Results and Discussion

4.1 Air Distribution And Pressure Losses

In a previous study presented by [9] pressure drops for typical centralized supply systems were estimated and compared to those occurring in the decentralized supply network with 18 fans. As input data the same virtual building as the one considered in this paper has been used. It could be shown that for a total volumetric flow rate of 8000 m³/h the pressure drop in a centralized supply system is around 800 Pa while for the decentralized supply system it is only 55 Pa, including the pressure drop in the supply unit. Despite of the higher efficiencies of the large, belt driven radial fans of 70% against 20% of the small, axial DC fans, the huge potential for fan power reductions by a factor of 4.16 was identified using the decentralized supply system. The analysis of the flow rate distribution for the unconnected and the network topology showed that in case of wind loading a much more uniform distribution can be achieved using a network approach. This holds at least for low wind speeds up to 6 m/s. For a larger wind speed of 10 m/s a significant spread between minimum and maximum flow rates could be observed also for the network. But with 6.06 m³/h the standard deviation in the network was still significantly lower than the 27.34 m³³/h in the unconnected case [9].

In addition to the open plan situation with the constant room pressure boundary a virtual separation was introduced to represent a single office space in a critical corner surrounded by negative pressure boundaries of -1 Pa was applied to the openings lying within this virtual office to account for the discontinuity between the biased supply and a constant exhaust. It could be shown that the cumulated flow rate of the openings within the office was only 5.5% below the value for no wind loading. This corresponds to an availability of 94.5% and points out the unique airflow routing qualities of the network distribution system with minute pressure differences [9]. For the analyzed setups shown in Fig.1 the net wind pressure evaluated at the fan locations appeared to be negative. Under these conditions and with the 18 fans available the design flow rate of 1600 m³/h could not be maintained.

4.2 Energy Demand Reduction

An overcapacity of fans is necessary to be able to maintain the design flow rate. The new network shown in Fig. 4 with 26 fans (40% overcapacity) allows for a flexible allocation of fan power such that it is possible to make positive use of the wind pressure.
Leaving all 7 fans switched on, on the windward (left side in Fig. 4) and on the opposite side, while switching off 4 fans on both of the other sides where maximum negative pressures appear, allows to reach the design flow rate of 1600 m³/h. Again only 18 fans remain switched on, such that the total power input remained unchanged compared to the previous study. The reason for this result is mainly due to fact that a different weighting of the wind pressures becomes possible if an overcapacity of fans is available. Also, the number of openings in the network following the topological pattern shown in Fig. 3 over-proportionally increase with the number of fans introduced such that a lower pressure drop per opening occurs.

The electrical power demand of a single fan unit at a prescribed rpm ratio of 82% is 11 W. The 18 fans in use have a cumulated power demand of 198 W.

If a constant flow rate control for the 18 fans is used, an additional power input is needed to compensate for the negative wind pressures. In that case no benefit results from positive pressure because throttling is used to keep flow rate constant. For the wind speed of 10 m/s and the fan allocation according to Fig. 1a the additional power input is 63.8 Watts. In case of constant flow rate control the total electrical fan power is therefore 261.8 W. When a 40% fan overcapacity is added using a switch-off strategy the power demand can be reduced by 25%.

Further energy savings are possible if no effective wind load is present and the overcapacity can be used for a temperature controlled optimization. In summer time large temperatures are recorded on parts of the building that are exposed to direct sunlight. While in summer this creates additional cooling loads it reduces heating loads in winter time. Measurements performed during summertime on three different unbound high rise buildings in Berlin, Germany revealed a maximum difference of 12 °K between ambient temperature and temperatures recorded in the thermal boundary layer, 0.5 cm away from the façade [17].

5.4 °K temperature difference was reported by [18] between opposite sides of an urban canyon in Athen, Greece. Using this temperature differences to estimate the potential reduction of cooling load per fan unit that is switched off on the hot side and switched on a cool side leads to 354 Watts for 12 °K and 159.3 W for 5.4 °K respectively. In the case of an unbound building an overcapacity of 8 fans in the network could be switched off such that a maximum reduction of 2831.5 Watts would result.

4.3 Exergy Analysis

The exergy analysis for the fans under influence of wind leads to the same result as previously calculated using a simple energy analysis. This is because the electrical fan power calculated using Equation 2 is equivalent to exergy, i.e. high grade energy. Using a 40% overcapacity of supply units instead of using constant flow rate control a 25% reduction of the exergy demand for the fans can be achieved under the assumptions made. Much more interesting is the exergy concept for the judgement of the impact of an inhomogeneous temperature distribution around the building on the system performance. For this purpose the exergy balance has to be applied over the heat exchanger. The exergy demand of the heat exchanger directly specifies the amount of high grade energy the generation system has to deliver in order to satisfy the task of heating/cooling. The heat exchanger used in the decentralized supply units is layed out for system temperatures close to room temperatures and is therefore very efficient to use together with a heat pump. If a compression heat pump is used on the generation side the minimization of the exergy demand in the heat exchanging process directly minimizes the electricity input to the heat pump.

During summer time when the ventilation runs in cooling mode the exergy demand for a single unit to cool outside air to room temperature is 7.53 W. When due to sun impact a local inlet temperature of 12°C above environmental temperature is encountered the exergy demand increases by 224% to 24.4 W. In winter when the devices run in heating mode the exergy demand to heat up from environmental temperature is 55.8 W and 42.7% below that value at 32 W for a local inlet condition of 12°C above environmental level. The heating situation with an increased local inlet temperature is exemplified in Fig. 3 with the according scaling. When looking at different configurations and comparing the unconnected topology from Fig. 1a with the network from Fig. 4 with a 40% fan overcapacity and a switch-off strategy, the following exergy savings are observed. For the unconnected topology of Fig. 1a, five devices on the left side of the figure are assumed to be directly exposed to the sun, in winter as well as in summer. For the network topology of Fig. 4, eight devices are completely switched off such that only 18 fan units are running. In the winter case seven devices are switched on on the sunny side, and for summer conditions only devices on the shaded side are running. As a result the winter exergy demand for heating the outside air per floor reduces by 5.4% from 885.5 W to 837.9 W when using the network solution. In summer where the optimization due to switch-off is much more effective exergy savings of 38.4% from 219.9 W to 135.5 W are possible when using the network.
5. Conclusions

As a previous analysis of the flow rate distribution for different topologies showed [9], the network topology leads to an even distribution of the flow rates for small wind speeds up to 6 m/s. Further it was shown that with the slightest negative inside pressure the routing of the airflows can be influenced such that the availability can be maintained also for rooms in critical locations, being surrounded by negative pressures.

In the current study it was found that significant energy and exergy savings result using a decentralized concept with extra savings due to the fan overcapacity installed and the network distribution approach used. The amount of overcapacity determines the potential savings. The larger the overcapacity chosen, the larger the potential savings. Unfortunately, investment costs rise proportionally with overcapacity too. Therefore a trade off must be found between running and investment costs. There are also other factors that limit the amount of overcapacity to be chosen. A minimum of active fans on either side is necessary to satisfy the conditions for an even flow rate distribution. On the other hand there is a limit to the number of fans that can be connected in a certain topological pattern without endlessly increasing the number of openings in the room.

6. Outlook

All the saving potential was examined for a hypothetical building case and it is therefore left to show how the savings would be for a actual building with more rigorous constraints and more realistic wind loadings. On the other hand it would be interesting if the potential savings could be quantified more generally such that these results could be used to estimate the saving potential for an arbitrary type of buildings.

As it was found the energy and exergy saving potential largely depends on the amount of fan overcapacity used. As a consequence a methodology is needed to determine the amount of overcapacity to install. This could be done using statistical weather data. Accounting for probable wind directions and strength as well as for solar irradiation and shading from surrounding buildings in the selection of overcapacity would then lead to an “optimum” performance of the building for most of the time.

8. References


6.8 Extended Research

6.8.1 Energy Saving Potential

The decentralized supply system described and analyzed in this work offers a large potential for reductions of high grade energy with reference to classical, centralized supply systems. This saving potential was introduced in the papers in this chapter and is summarized here given certain scenarios. These scenarios are the same as the ones used in the papers. For a building with a ventilation rate of 8000 $m^3/h$ as described in the building test case setup in subsection 6.4.5 the pressure drop in a centralized supply system was assumed to be 800 Pa. This pressure drop was estimated based on the evaluation of suitable exhaust fan units shown in subsection 6.4.6. Based on this evaluation, in the AIVC paper the fan power was evaluated and compared to a selected layout of a decentralized supply and distribution system. With the 55 Pa pressure loss and the 20% fan efficiency in the decentralized system a reduction of electrical fan power by a factor of 4.16 is possible despite of the larger efficiencies around 70% of the centralized fans. This factor corresponds to a 75.9% reduction of high grade energy, i.e. electricity in the system. Further savings are possible when the system is controlled to operate in a hybrid mode. The hybrid mode is when natural forces are used to support the otherwise purely mechanically driven system. Strictly speaking this includes only the influence of wind when wind pressure is used to assists in distributing the air in the space. The decentralized supply system comprising a large number of AHUs arranged along the periphery also allows for an optimization based on temperature. The uneven temperature distribution around the building can be used to control the heating/cooling load in the building. As described in the PLEA paper, a structure of a distribution system with a 40% fan over-capacity was chosen to analyze the saving potential running the system in a hybrid mode, either pressure or temperature based. For the one-sided wind loading with 10 m/s wind speed 8 fans were switched off on the side with the most expressed negative pressures while all other 18 fans were running. Minimizing the number of fans working on negative pressure sides of the building allows to maintain the design flow rate with a minimum fan power input. Using this approach a reduction of electrical fan power by a factor of 1.28 or by 22% was achieved. Superimposing the savings, a reduction factor of 5.3 or total savings of 81.2% in electrical fan power result.

The potential of optimized temperature control was analyzed for the situation that the air temperature in the boundary layer of one side of the building’s facade is 12 degrees Celsius above the air temperature along the other sides.
This temperature difference has been observed in a study and is due to direct solar irradiation on parts of the building. The reference system had 5 out of 18 AHUs permanently exposed to the sun where for the system using a 40% fan over-capacity 8 AHUs could be selectively switched off. The latter system had 7 AHUs on the sunny side that where switched off during summer when running in cooling mode and had all 7 switched on during winter when running in heating mode. In summer hence the additional cooling load from 5 devices and in winter the additional heating load from 2 AHUs could be avoided. This fact lead to reductions of the exergy demand of the temperature controlled supply system of 6% during heating and 40% during cooling. Reductions in exergy demand in the AHUs directly reduce the exergy load of the heat pump. The exergy load of the heat pump can be related to the necessary exergy input to the heat pump by using the exergetic efficiency defined as

$$\Psi_{ex} = \frac{Ex_{out}}{Ex_{in} + W_{el}}$$  \hspace{1cm} (6.40)

where $Ex_{out}$ in the enumerator is the exergy supplied by the system and $Ex_{in}$ and $W_{el}$ in the denominator is the exergy supplied to the system through heat transfer and electricity respectively. The definition of the exergy efficiency which relates the desired output to the necessary input is known as the rational exergy efficiency [16]. For the heating case the exergy supplied to the system through heat transfer can be considered as free exergy from the environment and be omitted in the efficiency definition. This leads to a new definition of the exergy efficiency for the heat pump as also used by [14]. This efficiency definition writes out as

$$\Psi_{ex} = \frac{Ex_{out}}{W_{el}}$$  \hspace{1cm} (6.41)

According to this definition, given a certain efficiency, reductions in the electricity input are directly proportional to reductions of the exergy load of the heat pump. This means for the scenarios considered that the electricity input could be reduced by 6% during heating and 40% during cooling. This corresponds with a reduction factor of 1.06 and 1.66 respectively.

For a heat pump running in the heating mode the exergetic efficiency according to the definition from eq. 6.41 is calculated as

$$\Psi_{ex} = g \left(1 - \frac{T_0}{T_H}\right) \frac{T_H}{T_H - T_C}$$  \hspace{1cm} (6.42)

Assuming a quality factor of $g = 0.5$, an outside temperature of $T_0 = 0^\circ C$, an evaporation temperature of $T_C = 10^\circ C$ and a condensation temperature $T_H = 30^\circ C$ the exergy efficiency takes a value of 75%.
In the case of cooling the exergy efficiency is calculated as

$$\Psi_{ex} = g \left(1 - \frac{T_C}{T_0}\right) \frac{T_C}{T_H - T_C}$$

(6.43)

Assuming the same input values as for the heating case except for the environmental temperature being $T_0 = 28^\circ C$ an exergy efficiency of 42% is reached.

### 6.8.2 Control Strategies

So far the control aspects of the ventilation system were excluded from the discussion. Every active, mechanical ventilation system has to have a well-defined control concept to be able to run in a certain, also well-defined mode. The system considered here combines a centralized exhaust and a decentralized supply. Using a demand-controlled approach, the ventilation rates vary depending on the occupancy in the building. Relying on CO2 measurements that directly trigger the exhaust flow rate, the supply system merely has to assure that the continuity is satisfied. The exhaust and the supply system within such a setup follow a master/slave logic.

For the supply system and its control the variable exhaust flow rate can be considered as a disturbance. Other disturbances for the supply system are variable outside conditions due to wind impact or direct solar irradiation. A decentralized supply system relying on a large number of AHU encounters a large variety of inlet conditions that affect its operation. Because of the AHU taking outside air from the facade of the building the exposure to variable outside conditions is stronger than for a centralized supply system. Under some circumstances this can be considered as a drawback but it can also be a chance for further optimizations if the control accounts properly for these disturbances. In combination with variable outside conditions the opening of windows by the user introduce a further disturbance in that it alters the pressure conditions in the room. Knowing the disturbance variables the undisturbed target or reference mode the supply system ideally runs in has to be defined. This reference operation can be defined as a balanced operation where the supply flow rate perfectly matches the exhaust flow rate. When the flow rates are matched the pressure difference between the room and the ambient vanishes. To have a zero pressure difference between the building and the outside is the target for a balanced ventilation system. Based on this reference operation the objective of the control can be defined. First, during regular operation it is to maintain this reference operation against the influence of the various disturbances acting on the system. As a second objective, during optimized operation (hybrid mode) the optimization of the operation
towards lower energy usage can be introduced. With the reference operation, the objectives and the disturbances defined, the control problem is fully determined and possible strategies have to be formulated. In the control strategy the control variable and the controlled variable have to be defined. For the supply system to be able to adjust the flow rate and/or the pressure rise the only way is to alter the fan power, i.e. the revolutions per minute (rpm) of the rotor. As a consequence at any case the controlled variable has to be the rpm or a related quantity. Across a fan the kinetic and the potential energy of the air stream is increased. The mass flow and the pressure rise induced by the fan are coupled and constitute together the power output of the fan. As a consequence the control variable can either be the flow rate or the pressure rise across the fan. Of course it could be any other quantity defined on a basis of either of those quantities or of both.

Fundamentally there are few control strategies for a ventilation system with variable exhaust and supply possible that shall be discussed in the following. It is differentiated between the control strategy used during regular operation (RO) which aims at maintaining the reference operation and the optimized operation (OO) which aims at further reducing the fan power usage. The OO is also called hybrid mode because the wind is used to support the air distribution.

Regular Operation

**RO1: Pure constant flow rate control of exhaust and supply** The most direct way of controlling the ventilation rate in a demand controlled system based on LEV is by using the information from the individual exhaust openings. The flap of each opening is controlled by $CO_2$. When a certain setpoint is exceeded in the exhaust air the flap opens and the contaminated air is extracted. This extraction is determined by $CO_2$ and executed by the flap either being open or closed. As a consequence the position of the flap can be used to determine when to ventilate. The actual flow rate handled by a single exhaust opening depends on the size and the control of the exhaust fan. Using the exhaust flap’s position to control the ventilation rate, the exhaust fan is sized based on the maximum flow rate that has to be handled (including simultaneity estimates). A minimum flow rate is always extracted from the space, even when all flaps are ”closed” and has to be set. When an exhaust flap is opened, a fixed design flow rate of e.g. $50m^3/h$ is associated with it. The system detects the amount of opened flaps which gives the reference flow rate for the exhaust fan by multiplying the number of opened flaps times the design flow rate per exhaust opening. If more flaps are opened simultaneously than
assumed during the sizing of the exhaust fan the extraction rate per exhaust opening simply deteriorates below its design value. From the total flow rate of the exhaust the reference flow rate for the AHU is also known. Of course there must be a predefined associativity between the exhaust flaps and the AHUs within certain zones. This makes sure that the different zones receive the right amount of outside air and the system is well balanced. Both, the exhaust and the supply fans are controlled to keep the constant flow rate according to the measured/evaluated reference. This task involves a feedback loop control in the exhaust fan and the AHU similar to the generic example shown in Figure 6.37. The system e.g. being an AHU receives an input from the controller

![Feedback Loop Diagram](image)

Figure 6.37: A generic feedback loop with its input and output variables.

which is the controlled variable. This value contains the corrections needed to bring the system back to its reference operation after it has been disturbed by some external forces such as variable wind pressures.

With the direct flow rate control of the AHUs the ventilation system is always well balanced as long as the supply or the exhaust system are not required to operate outside their capacity limit. Running off-design would allow a positive or negative pressure to form in the room depending on the exhaust or the supply being the limiting capacity. Within the design range the supply system controls to a constant reference flow rate it receives from the exhaust system. This control makes the system robust to disturbances caused by the wind. The fans in the AHUs balance out the fluctuations in pressure imposed by the wind. Depending on the fan type used in the AHU however the range of pressure within which the fan can operate maintaining a constant flow rate can be strongly limited. Considering axial fans, these limitations are more severe than for radial fans. A reasonable operation with small axial fans will exclude negative wind pressures below -50 Pa.

When windows are opened in a system that is fully controlled on a flow
rate basis there is no influence of this disturbance on the system. The only effect window opening can have is the increase of air exchange in the room such that the CO2-controlled exhaust system adapts the ventilation rate according to the new situation.

**RO2: Constant pressure control of exhaust, constant flow rate control of supply**  Assuming a constant pressure control of the exhaust side, the exhaust fan is set to maintain a fixed pressure difference over the rotor. Accordingly the pressure drop in the exhaust system is kept constant. The exhaust flaps alter the resistance in the exhaust system which leads to variable ventilation rates. Relying on a pure flow rate control of the AHU the reference value for the flow rate has to be gathered from the exhaust system in order for the ventilation system to be balanced. For this purpose the flow rate of the exhaust system has to be measured. In a real installation within an office building using LEV, zones can be defined that are served by a certain amount of exhaust openings. To the same zone a fixed number of AHUs can be assigned. If the exhaust flow rate is measured in a branch that collects all the exhaust openings from within the zone, the measured value divided by the number of AHUs gives the reference for the operation of the AHUs. In a demand controlled exhaust system where the flow rate varies with time a periodical measurement of the exhaust flow rate and an update of the AHU reference is necessary. The time constant for this update depends on the hysteresis used for the demand control, the number of exhaust openings installed, the occupancy, etc.. Adjusting the supply to the exhaust in the described manner involves a direct control without feedback loop. The feedback control is used within the AHU to adjust the rpm to keep a design flow rate and to counteract the disturbances. Considering the wind, it leads to both, negative and positive pressure zones on the building. If a negative pressure is encountered by an AHU at the intake the operation point of the AHU moves along the fan curve towards lower flow rates as it has been discussed in subsection 6.4.2. With the constant flow rate control the AHU increases the fan power to be able to keep the flow rate against a higher pressure difference. The opposite happens if a positive local pressure is encountered by the AHU at the intake. When an non-homogeneous pressure distribution around the building is assumed and a window is opened for instance on the pressure side, the room gets pressurized and an increased air flow through the window takes place until the local outside pressure and the room pressure is the same. When a pressure-controlled exhaust fan is used the pressure rise over the fan decreases because of the increase of room pressure assuming a constant ambient pressure.
at the exhaust location of the fan. As a consequence the fan power and hence the flow rate is increased. This is unwanted and therefore additionally to the pressure control a flow rate limiter has to be used. In case of negative room pressure due to an open window on the negative pressure side the ventilation rate is reduced. The problem there is that people far away from the opened window are supplied with less outside air through the AHU. This can lead to locally unsatisfactory IAQ if this situation persists but it can be assumed that in an office building windows are opened only during a short amount of time and hardly during windy conditions because of problems of draughts. The flow rate control in the AHU is shown in the flow chart of Figure 6.38.

Figure 6.38: Flow chart of local control strategy for an AHU based on a constant flow rate.

**RO3: Constant flow rate control of exhaust, pressure based control of supply**  
An alternative to the constant flow rate control of the supply is the use of pressure as a control variable. The exhaust flow rate is assumed to be determined according to RO1 and hence is insensitive to changes in room pressure. The insensitivity is in so far true as the characteristics of the centralized fan is usually steeper than those of the decentralized AHUs.

When the exhaust flow rate is changed, e.g. increased, the pressure in the room becomes negative and the exhaust partly pulls outside air through the AHUs. This is true for the case that the AHU does not adjust its fan power. For the adjustment of the fan power based on the pressure the question is which pressure or pressure difference can be used as a control variable. The
pressure difference that at reference operation vanishes is the one between the room and the undisturbed environment. The target for the operation must therefore be this zero pressure difference which is the case as we know for balanced flow rate. As a consequence the AHU must be controlled based on this pressure difference. The ambient pressure can be measured at one location outside of the building where wind speed is insignificant. The room pressure has to be measured per room or if it is assumed that the entire floor is supplied with air through a highly interlaced ductwork the discrepancies in room pressures is negligible and a single measurement per floor is sufficient. These absolute pressure measurements have to be communicated to the AHUs such that they can be used for the local control. If the difference between the room and the ambient pressure \( p_{\text{room}} - p_{\text{ambient}} \) becomes negative, too much air is extracted from the room and the AHU have to increase the flow rate by increasing the fan power. If the pressure difference becomes positive less air has to be supplied by the AHU. The pressure measurement has to be periodically repeated and transmitted to the AHU. Similar to the variable flow rate of the exhaust does the influence of the wind cause the room pressure to deviate from ambient pressure. Depending on the net pressure value around the building an over- or under-supply leads to a mismatch of the supply and the exhaust flow rate. When the windows are opened e.g. on the pressure side the room pressure rises above the undisturbed ambient pressure. For the AHU this means that the pressure difference to the ambient level is positive and hence the flow rate, i.e. the fan power is reduced. If the window is opened on the negative pressure side the fan power of the AHU turns up and a larger flow rate is brought into the room. Because of the larger flow rates extracted by the wind and the low pressure drop across the window opening, the room pressure cannot be brought back to ambient level. Opening the windows when wind is hitting the building does not have a significant effect on a ventilation system with a flow rate controlled exhaust. Rather it influences the operation of the pressure controlled supply but without large consequences. Of course, the ventilation system is not able to balance out the pressure imposed by the wind such that draughts from opening the windows can occur. In the case the window is opened on the negative pressure side the supply rate is increased which however does not have any impact besides for a higher fan power usage during the opening time of the window.

The local control of the AHU must be based on the difference between room and ambient pressure but this difference cannot be the control variable. As a control variable a local value is required which reflects the situation of a single AHU. The problem at this point is that it exists a global criterion (zero pressure difference) but a uniquely related local criterion and the suitable
control variable does not necessarily exist. This is because a global behavior can eventually be inferred from a local behavior but not vice versa. A possible local strategy can be defined using the pressure difference between the outlet of the AHU in the floor and the intake in the facade \( p_{\text{floor}} - p_{\text{facade}} \). This pressure difference reflects the effective pressure rise provided by the fan in the AHU but not the pressure rise it has to provide to avoid a difference between room and ambient pressure to form. As a consequence this pressure difference has to be included for finding a meaningful reference value for the operation of the AHU. Combining the pressure values, the pressure rise over the fan can be written as

\[
\Delta p_{\text{AHU}} = p_{\text{floor}} - p_{\text{facade}} - (p_{\text{room}} - p_{\text{ambient}}) \quad (6.44)
\]

The first pressure difference in eq. 6.44 comprises the two pressure differences \( \Delta p_1 \) and \( \Delta p_2 \).

\[
\Delta p_1 = p_{\text{room}} - p_{\text{facade}} \\
\Delta p_2 = p_{\text{floor}} - p_{\text{room}} \\
p_{\text{floor}} - p_{\text{facade}} = \Delta p_1 + \Delta p_2
\]

Rewriting eq. 6.44 with the pressures \( \Delta p_1 \) and \( \Delta p_2 \) gives

\[
\Delta p_{\text{AHU}} = p_{\text{room}} - p_{\text{facade}} + p_{\text{floor}} - p_{\text{room}} - (p_{\text{room}} - p_{\text{ambient}}) \\
= \Delta p_1 + \Delta p_2 - \Delta p_3
\]

where \( \Delta p_3 \) is the difference between room and ambient pressure. From eq. 6.46 it can be seen that the pressure drop the fan has to overcome is given by the losses in the distribution system by \( \Delta p_2 \) plus the deviation of the local facade pressure from the ambient pressure represented by \( \Delta p_1 - \Delta p_3 \).

According to eq. 6.46 the fan pressure must increase if a larger flow rate is demanded from the exhaust system. This increase in the exhaust flow rate is indicated by the increase of \( \Delta p_2 \). Similarly the fan pressure has to increase if the local pressure on the facade drops. The increase required is represented by the increase in \( \Delta p_1 \). The pressure differences \( \Delta p_1 \) and \( \Delta p_2 \) are local measurements that can be performed in the AHU and are therefore readily available during operation. The pressure difference \( \Delta p_3 \) is obtained from a global measurement and is not available for a real-time control. As a consequence a calibration of the system is necessary to a state where this pressure difference vanishes and the conditions of a balanced ventilation system are met. The local pressure differences \( \Delta p_1 \) and \( \Delta p_2 \) are then used during operation to react
to disturbances that pull the system away from its calibrated state. The exact procedure of the calibration and the subsequent control is shown in Figure 6.39. With the chosen strategy, there is still no true reference pressure that could be used for the control of the supply. From the local pressure measurements the deviation of the pressure differences with references to the calibrated state are known. This knowledge can be used to calculate the future pressure rise required by the fan to balance out the change in pressure. For this purpose the proportionality laws for fans have to be applied to estimate the future rpm from the actual rpm and the ratio between the actual and the calibration pressure difference. The relation between the rpm and the pressure rise over a fan is given as:

\[
\frac{rpm_{n+1}}{rpm_n} = \left[ \frac{\Delta p_{n+1}}{\Delta p_n} \right]^\frac{1}{2}
\]

(6.47)

The calibration process has to be carried out under wind free conditions. The exhaust then must be set to a selected design flow rate. The design flow rate is the flow rate around which one expects the system to run most of the time. It is important though that at design conditions the AHUs have enough margin to adjust their fan power towards lower as well as toward higher values. Having the exhaust flow rate set, the difference between the room and the ambient pressure is measured and the fan power of the AHUs is adjusted uniformly such that the pressure difference is brought to zero. At this condition there is no pressure difference between the ambient, the facade and the room and hence \( \Delta p_1 \) and \( \Delta p_3 \) are zero. As a consequence the remaining pressure difference according to eq. 6.46 reduces to \( \Delta p_2 \). This is the pressure loss in the air distribution system that the fan has to overcome given a certain flow rate in the absence of disturbances. The pressure difference \( \Delta p_2 \), locally measured during calibration, is stored in each AHU as the reference pressure for control. During the actual operation of the ventilation system the pressure difference according to eq. 6.46 is evaluated permanently by the AHUs. For a well-calibrated system, \( \Delta p_3 \) vanishes and the local measurement includes only \( \Delta p_1 \) and \( \Delta p_2 \). If the locally measured pressure difference is larger than reference pressure the fan power is turned up. Otherwise, when it drops below the reference value the fan power is turned down. The scaling of the fan power follows the formula shown in eq. 6.47. Periodically, within a defined schedule, a global measurement of \( \Delta p_3 \) is made to check whether the ventilation is well-balanced. If a non-zero value is recorded the reference pressure in the AHUs is corrected. A positive pressure difference for instance indicates that there is an over-supply. By adding this value to the old reference the new reference pressure difference gets larger such that the fan power of the AHUs is throttled.
start calibration at no wind condition
set exhaust flow rate to design value and switch on AHU all with same rpm
measure $\Delta p = (p_{\text{room}} - p_{\text{facade}}) + (p_{\text{floor}} - p_{\text{room}})$
if $\Delta p > \Delta p_{\text{reference}}$
    yes
    increase fan power acc. to $(\text{rpm}/\text{rpm}_1)^2 = (\Delta p/\Delta p_{\text{reference}})$
no
if $\Delta p < \Delta p_{\text{reference}}$
    yes
    decrease fan power acc. to $(\text{rpm}/\text{rpm}_1)^2 = (\Delta p_{\text{reference}}/\Delta p)$
no
if $\Delta p < 0$
    yes
    increase AHU fan power
no
if $\Delta p > 0$
    yes
    reduce AHU fan power
no
store $\Delta p_{\text{reference}} = \Delta p = (p_{\text{floor}} - p_{\text{room}})$ and fan power setting / rpm
if time > T
measure $\Delta p_3 = (p_{\text{room}} - p_{\text{amb}})$ and adjust $p_{\text{reference}}$ by $\Delta p_3$
no
yes
end of calibration

Figure 6.39: Flow chart of local control strategy for an AHU based on pressure.
RO4: Constant flow rate control of exhaust, constant pressure control of supply. The pressure control of the supply described under RO3 uses a reference pressure and a deviation from the reference to determine the new operation point of the fan. The projection of the new operation point is based on the proportionality laws for fans. It is though only an estimate, an extrapolation that a posteriori has to be verified and corrected again. With this control approach similar to the approaches described under RO1 and RO2 it is tried to maintain a constant and equal flow rate for all AHUs. In contrast to this approach, pressure control for the supply could also be used differently, without imposing that the flow rates of all the AHUs necessarily have to be the same. By taking away this restriction the regular operation (RO) eventually moves already towards an optimized operation (OO) which aims at a minimization of fan power. A strict pressure control of the supply in the sense of a constant pressure control does not actively control the flow rate and hence allows the AHUs encountering different boundary conditions to work with different flow rates. For this constant pressure control of the supply the exhaust has to be flow rate controlled in order to have a stable and well defined working condition. The ventilation rate in the system is determined by the exhaust in the same way as explained for RO1.

When using constant pressure control on the supply side a reference pressure for the AHUs has to be found at which the conditions of the reference operation are met. As we know these conditions are met if the flow rates between the exhaust and the supply are matched at any time. The reference pressure for the AHUs is gained through a calibration of the ventilation system. The calibration is carried out in the same way as described for the pressure control described under RO3. In contrast to the previous case, only a single pressure difference in the AHU, being the pressure difference between the outlet in the floor and the intake in the facade ($p_{\text{floor}} - p_{\text{facade}}$) is measured and stored as a reference. During regular operation, when the exhaust flow rate changes, the AHUs fans react to keep constant pressure difference. If the ventilation rate is increased by the exhaust a negative pressure forms in the room. The pressure in the floor drops by the same amount as in the room. As a consequence the pressure difference over the fan in the AHU drops assuming no change in the facade pressure. Using constant pressure control over the fan, hence, the fan power and with it the supply flow rate will be increased. The increased flow rate leads to an increase of the pressure drop in the air distribution system. The fan power of the AHU is turned up until the negative pressure in the room and the pressure loss in the distribution system have the same value and the reference pressure over the fan is restored. Having an air distribution system such as the ducting network considered in this work
with a flat loss characteristics, the positive or negative pressure forming in the room when the exhaust flow rate is reduced or increased is very moderate. The loss curves for the AHU and the ADS are shown in Figure 6.40. The pressure loss of the ADS represents an average value calculated for an exemplary topology with highly interlaced AHUs for the situation of no wind. The loss curve is constructed from three data points using a quadratic least-squares fit. When wind is considered, the loss in the ADS depends on the flow rates of the different AHUs and no unique value can be found. The difference of the ADS losses seen by the different AHUs represents the deviation of the ADS from the ideal of a infinite pressure plenum or from the idea of a ”lake” of fresh air. As can be seen in Figure 6.40 for the loss curve of the ADS, changes of flow rate between 50 and 100 $m^3/h$ lead to a change of pressure loss from 3 to 10 Pa. As a consequence when the system is calibrated for a flow rate within the given range the negative or positive pressure that can possibly build up when the flow rate is changed is less than 7 Pa. When wind is acting on the building, the

![Figure 6.40: Loss curve of AHU and ADS](image)

AHUs that encounter negative or positive facade pressure turn the fan power down or up trying to keep the pressure rise over the fan constant. The AHUs
that cannot further reduce the pressure difference by adjusting their fan power
switch off completely.

The constant pressure control of the supply simply leads to the conditions
of constant floor pressure. The floor with its ducting network is treated as a
pressure plenum that is fed with fresh air such that the pressure stays constant.
Some AHUs under certain conditions contribute more than others because
their conditions are more favorable. Favorable conditions are, when the wind
causes a positive pressure on the facade which supports the air transport.
As a consequence this control approach automatically leads to an optimized
operation in terms of power usage because more air is brought in from the
pressure side and less from the lee side. There is a problem however if too
many AHUs stand in a negative pressure. The flow rate cannot necessarily be
compensated from the positive pressure side and a negative pressure forms in
the room. Again, as already used in RO3, a periodical adjustment of the AHU
reference pressure by the difference between the room and ambient pressure
can be realized. An increase in reference pressure for instance will cause some
AHUs that are switched off because of largely negative wind pressures to switch
back on and hence to reduce the negative pressure in the room.

Optimized Control (Hybrid mode)

Beside the control strategy for a regular operation of the ventilation system
a particular strategy for an optimized operation is required. The Optimized
operation, also called hybrid mode of the system, has the goal to minimize
energy or exergy usage for ventilation. There are mainly two components
of disturbances from outside that allow for an optimization of the system
performance. It is the wind that causes non-homogeneous pressure boundary
conditions around the building and the direct irradiation of the sun that leads
to an uneven temperature distribution along the building envelope. Wherever
there is a difference in physical quantities there is a work potential. The
difference in wind pressure can be used to support the air distribution while
minimizing electrical fan power usage. The goal is to supply higher flow rates
from the positive pressure side of the building and decrease the supply from
the negative pressure side. The temperature do not have any impact on the
dynamic behavior of the AHUs but by selectively assigning different flow rates
to the AHUs depending on their inlet temperature it allows to minimize cooling
and heating loads in summer and winter respectively.

Every AHU has to find out whether it is standing in a positive or nega-
tive pressure environment. Assuming that the regular operation assures that
there is no difference between the ambient and the room pressure a pressure
measurement between the room and the facade can tell whether an AHU is a positive or negative pressure AHU. An alternative way to classify the AHUs is to measure the supply flow rate together with a power measurement of the fan. The AHU with low flow rates and large power requirement is located on a negative pressure side. Of interest for the optimized operation is only the time average of the facade pressure taken over a certain period. This is because a real-time optimization over the entire system which is able to also make use of wind gusts would be too complex and result in large traffic for communication. If a stable wind situation exists an optimization can be run for this situation.

OO1: Based on flow rate controlled operation of the supply according to RO1 / RO2 An optimization can be run by altering the individual reference flow rates in the AHUs. Collecting the data of all AHUs concerning the average pressure difference between the room/floor and the facade or the flow rate combined with the power requirement the reference flow rate can be redistributed among the different AHUs. This is done as follows:

$$P_{el} = \frac{P_{mech}}{\eta} \quad P_{mech} \sim \Delta p_{Room,Facade,i} \dot{Q}_{ref,i}$$  \hspace{1cm} (6.48)

$$\min \sum_i \Delta p_{Room,Facade,i} \dot{Q}_{ref,i} \quad \text{where} \quad \sum_i Q_{ref,i} = \dot{Q}_{total}$$

The objective is to minimize the electrical power usage of the supply system comprising a large number of AHUs. The electrical fan power is either evaluated using the pressure difference over the fan and the flow rate or alternatively through a direct power measurement. The constraint in the optimization is that the sum of the flow rates of all AHUs remains unchanged and corresponds to the flow rate imposed by the exhaust system.

Alternatively to the optimization scheme represented in eq. 6.48 another scheme shown in eq. 6.49, based on an efficiency measure can be used for the optimization. This efficiency measure can be defined as the ratio of the flow rate delivered to the electrical power input. The larger the flow rate for a given power input the higher is the efficiency of an AHU. The optimization is hence formulated as the maximization of the efficiency measure $\tilde{\eta}$. The constraint for the optimization remains the same as before.

$$\tilde{\eta}_i = \frac{\dot{Q}_i}{P_{el,i}}$$  \hspace{1cm} (6.49)

$$\max \sum_i \tilde{\eta}_i \quad \text{where} \quad \sum_i Q_{ref,i} = \dot{Q}_{total}$$
Additionally to the fan power optimization a thermal optimization can be made. For this case the inlet temperature of each AHU must be measured. Together with the value of the room temperature and the mass flow, the heating or cooling demand of the AHU can be determined. As a consequence the heating or cooling demand of the building can be minimized based on a given temperature distribution. The optimization of the building performance is hence a superposition of the pressure and the temperature optimization. The ventilation system can be controlled to consider both quantities during optimization. Based on a evaluation of the total saving potential it has to be decided whether only a pressure-, a temperature- or a combined optimization is chosen.

Running an optimization that leads to a redistribution of reference values of the flow rates among the different AHUs requires all AHUs to communicate with each other or at least with one central unit. From a communication perspective however it is more efficient instead of having cross-communication between all AHUs to establish a link between the AHUs and a single, central unit. This unit gathers all the information from the AHUs that is needed to run the optimization. As a result of the optimization new reference flow rates for the AHUs are obtained that are fed back after the optimization. The optimization procedure is repeated periodically. Since optimization is only done for relatively stable and persisting conditions the periodicity of the procedure does not need to be high and can stay within the range from five minutes to one hour or more. The optimization can also be synchronized with the general update of the reference flow rates that are determined by the exhaust system. The regular update homogeneously scales the flow rate of the AHU while the optimization process determines a weighting for the flow rates of the different AHUs.

**OO2: Automatic fan power adjustment based on constant pressure control of supply according to RO4** Using a constant pressure control for the supply leads to a non uniform distribution of flow rates among the AHUs assuming a non uniform pressure distribution around the building. This control approach automatically adjusts the fan power in such a way that the over-all electrical power usage is minimized. No special control is hence necessary for an optimization of the ventilation performance under the influence of wind. For a temperature based optimization the constant pressure control of the supply is not suitable. Because the flow rates have to be adjusted according to the temperature distribution a flow rate based control has to be used. In a system with constant pressure control the design flow rate is only once set
during the calibration process. The calibration determines the reference pressure for the AHU which from then on is the only quantity the AHUs control for.

**OO3: Based on pressure controlled supply according to RO3**  
The pressure control during regular operation RO3 is aimed at balancing out disturbances from outside by the wind and from the variable exhaust. Pressure changes from outside are counteracted. It is essentially a constant flow rate control that is realized based on pressure balancing. For an optimized operation the positive pressure on the facade should be used for air transportation. As a consequence flow rates from AHUs on the pressure side should be maximized. While at regular operation \( \Delta p_1 = p_{room} - p_{facade} \) is counteracted for the optimization this pressure difference should be used as a support for air distribution. In order to achieve this, the control variable as expressed in eq. 6.45 has to be altered. Instead of calculating the control variable as the sum of \( \Delta p_1 \) and \( \Delta p_2 \), during optimization \( \Delta p_1 \) must be subtracted from \( \Delta p_2 \) to obtain the control variable. The implication of this modification is an increase of the supply flow rate from the positive pressure side and a decrease from the negative pressure side. However, the change of the control variable by inverting the sign of \( \Delta p_1 \) is suboptimal and does not reveal the full potential of fan power optimization. For a real optimization of fan pressure usage the AHU on the positive pressure side would have to be turned up to the maximum, the change in flow rate to be detected and accordingly fans on the negative pressure side would partially have to be switched off. The general problem with a redistribution of the flow rates among the AHUs using a pressure control is that the impact of a change in fan pressure on the flow rate is not the same on the positive pressure side as it is on the negative pressure side. For a given change in pressure the decrease in flow rate on the negative pressure side will be larger than the increase on the positive pressure side because of the non-linearity of the loss curve. This can be easily verified in Figure 6.41. The loss curve determined by the resistance of the AHU and the ADS is shifted up and downwards for positive and negative pressure standing on the facade. During regular operation an increase of the control variable by \( +\Delta p_1 \) would cause the fan to turn up to balance the negative facade pressure. During optimization using a control variable with inverted \( \Delta p_1 \) a negative facade pressure leads to a turning down of the fan. From the original operation point \( OP_{neutral pressure} \) the fan pressure is decreased by \( \Delta p_1 \). Given the new fan pressure, moving to the loss curve for a negative wind pressure gives the new operation point \( OP_{neg.pressure} \) for the AHUs on the negative pressure side. In the abscissa the
Figure 6.41: Impact of wind pressure and pressure control onto the flow rate of the AHU; non-linearity of loss curve

decrease of flow rate can be verified. Assuming the same pressure difference $\Delta p_1$ on the positive pressure side, the change in flow rate when moving to the new operation point $OP_{pos.\text{pressure}}$ is comparatively smaller. If the loss curve is very flat and small wind pressures are assumed the discrepancy between the change in flow rates on the positive and negative pressure side is of course less significant but still there. In order to be able to run an optimization without violating the conditions of a balanced ventilation the flow rates have to be known such that the increase in flow rate at some point can be properly balanced with a decrease at another place. This flow rate matching requires either knowledge about the loss characteristics of the ADS or a measurement of the flow rates in the AHUs. When using a pressure control however the change
in flow rate has to be translated into a change in pressure. To implement the optimization with pressure control, based on flow rate informations involves similar steps as in OO1 but is more complicated and less suitable (imprecise) because it includes extra translation of flow rates into pressures.
Bibliography


Chapter 7

Closure

This thesis had the goal of introducing and analyzing new concepts in the field of HVAC. With the focus on ventilation, it was claimed that using these new concepts a significant potential for reducing exergy usage in building operation exists. More precisely, looking at ventilation alone, a reduction in exergy usage by a factor of around ten was predicted.

Nowadays the awareness for the problems related to climate change has increased and strongly influenced the way buildings are designed and operated. These changing boundary conditions for the buildings trigger the development of new concepts and technology. In this work an approach focussed on the decentralization and distribution of HVAC equipment in buildings together with a perspective of their exergetic performance was chosen. Based on these two paradigms of decentralization and exergy minimization, new specific concepts for ventilation were analyzed.

This included the analysis of

- a demand controlled exhaust system relying on distributed sensing and local contaminant extraction
- an air supply and distribution system relying on decentralized air handling together with a highly networked distribution infrastructure
7.1 Demand-controlled, Local Exhaust

The ventilation concept described in this work is the one of an *exhaust ventilation*, meaning that the exhaust plays a larger role than the supply. Thus, the primary focus of the ventilation lies on the removal of contaminants from the room. The design that was studied and presented in Chapter 5 relies on a large number of active, sensor controlled suction points, which determine the overall ventilation rate and the spatial allocation of the exhaust power. It promises an effective removal of the contaminants through local extraction and an efficient operation because of the demand control. The local extraction introduces the paradigm of decentralization in that it distributes the functionality of capturing exhaust air and delegates it to a large number of decentralized, autonomously-working suction points. By using this large number of autonomously-working actuators, a complex need-based behavioral pattern of the exhaust system emerges, which could not be realized by a single, centralized control unit. In practice, the principle of local exhaust has only been used in the field of industrial ventilation. With the design considered in this work this principle was adapted to office type buildings and benefits from some of the advantages this principle brings in industries.

The demand controlled nature of the exhaust not only controls the ventilation activity but further leads to the concept of a suction driven ventilation. This concept is in contrast to the typical pressurized, throttled ventilation systems. Those systems use a top-down approach with one global supply which is adjusted to the demands through throttling. The suction principle describes a bottom-up approach. The ventilation rate is compiled through summing up individual, local demands. Instead of relying on throttling, the total demand is dynamically adapted to the needs. This inversion of the classical top-down approach also results in significant efficiency improvements through minimizing the dissipation of potential energy.

Simulations and measurements carried out and presented in the paper included in this chapter confirmed the effectiveness of the exhaust system using local extraction. Contaminants can be more effectively removed from the room, and the build up of the contaminant concentration can be significantly delayed. The local extraction ensures higher contaminant concentrations in the exhaust air and hence minimizes the spill-back of contaminants from the source into the room. This maintains the same IAQ as for mixing ventilation, but with significantly lower ventilation rates.

According to literature, energy savings using DCV can be up to 50%. This is because of the reduction of the flow rates and the resulting lower costs for the conditioning of outside air and its distribution in the building. In
office buildings the typical occupancy level is around 76%, which leads to a potential flow rate reduction of 24% assuming an ideally working DCV. Measurements of a ventilation system using local extraction presented in this chapter indicated a further reduction of the flow rate by 9% without changing the IAQ. The exhaust system considered allows for reducing the flow rate by 30.8%. Accounting for the effect on the heat demand and the electricity usage of a heat pump, savings of 8.8% were shown to be possible. More dramatic are the savings of high grade energy on the distribution side. The reduced amount of air to be handled leads to 66.9% less electricity input for the fan. In total the electricity usage of the ventilation can be reduced by a factor of 4 or greater.

7.2 Decentralized Supply

The supply side of the ventilation analyzed and described in Chapter 6 follows the ideal of a lake of fresh air. Using this picture, fresh air is withdrawn by the exhaust system according to the needs without inducing any significant velocities in the lake and hence maintaining a constant static pressure distribution. This leads to regular air supply with minimal pressure losses and results in a supply system with a very strong potential for exergy savings. Also does a similar supply approach underline the fact that the supply system serves the exhaust and does not itself determine the ventilation activity in the space. The demand controlled, local exhaust is the master, the supply system acts as the slave.

Knowing the desired behavior of the supply system derived from the concept of a lake of fresh air, the question of how this behavior can be achieved with a real system had to be answered. The quality criteria summarizing the desired behavior of the supply system are the regular distribution, the economy and the responsivity. This means that a desired system ensures a regular air distribution in the space with minimum energy usage while being sensitive to changing pressure boundaries such that spatially variable demands from a local exhaust system can be ideally served. It can easily be shown that low velocities in the air distribution are the key to satisfy all quality criteria required from the supply system. This velocity determines the capacity and size of the distribution, which is however limited because of spatial restrictions in the building.
7.2.1 Topology of Integrated Building Infrastructure

Suspended ceilings or raised floor constructions as often used in office buildings offer a sufficiently large plenum for air distribution similar to a lake. For thermal performance reasons these suspended structures must be avoided because it decouples the room from the thermal mass of the building. If the the thermal mass of the building cannot be activated, the peak power required from the heating or cooling plant increases. This is because all the power required during any extreme event in terms of temperature comes from the heating/cooling plant alone. If thermal activation is possible, the building structure can be used as a thermal store that can be managed. During night time for instance, when power requirements are usually low, the thermal store is loaded. During daytime the thermal store can be unloaded leading to smaller peak power required from the heating/cooling plant. Based on this peak power rationale, thermal activation enables heating/cooling systems running with supply temperatures close to room temperature to be installed, which in turn significantly reduces exergy destruction. The thermal activation requirement then leads to the need for air distribution infrastructure to be integrated in the primary building structure.

Hierarchical single supply systems as commonly used, try to match the pressure drops across different branches leading to a regular supply through throttling. As a consequence conservative highly dissipative systems result. Because of the branching topologies used by such single supply systems the transportation infrastructure (ducting) is usually too large to be integrated. In the case that both, the supply and the exhaust are handled centrally, the problem of ducts crossing each other also make the integration of the distribution infrastructure impossible and require suspended ceilings or raised floors to be installed. When splitting up a centralized plant into a large number of small decentralized plants, they can be distributed and integrated in the building structure. Because of the integration possibilities of small equipment, significant space savings and a more efficient use of the available space results. This integration has further implications on the construction in that it reduces the required story height which eventually allows the integration of more stories in a building of a given height, or in any case reduces the construction costs.

The substitution of a centralized single-source supply structure with a decentralized multi-source system leads to different topologies of the distribution infrastructure. Most commonly no ducting is used in decentralized systems for the distribution of supply air. Fresh air is directly released to the room at the facade where the AHUs are installed. This may not allow the formation of a
lake of fresh air, especially not in large spaces. In that case ducting could be used to connect the inner core of the building with the peripheral sources in order to guarantee a decent air supply throughout the whole space. Assuming the supply branches coming from the periphery to join each other in the space, a closed loop network between the peripheral devices automatically evolves. The supply openings in the network then act like neighboring zones in a lake that influence each other by balancing out pressure differences through inducing a flow. It was found in the study of different network patterns that closed looped network topologies show a favorable behavior similar to purely parallel arrangements.

7.2.2 Looped Ducting Networks for Air Supply

For a rigorous investigation of the behavior of highly interlaced ducting networks, i.e. routing of the air, flow rate distribution and pressure drops, a simulation had to be set up. This was done based on a 1D model that was implemented. Since only branching topologies are common in the field of HVAC, no codes where found to calculate the routing of air flows in closed loop topologies. The code that was set up represents the ducting network as a combination of pipes and nodes. The nodes are either just simple junctions to connect the pipes or also act as supply openings. An active component being a pump is included to represent the AHUs in the System. Different boundary conditions representing either the outside or the inside of the building can be input to the model. This includes temperatures and the static pressures. When running a simulation, a pressure value is assigned to every node in the network. Based on the given pressure distribution the flow in the pipes is evaluated and corrected respecting the fundamental laws of mass and energy conservation. A converged solution is obtained when the system has reached a steady equilibrium state, and all flows and pressures are well determined.

Variable boundary conditions from the influence of wind on buildings were considered in the study of the distribution network. An air flow around a building induces a non homogeneous pressure distribution. Positive pressures are found upstream of the building around the stagnant flow region. Negative pressures are encountered in the wake region downstream of the building as well as on the sides where large wind speeds occur. Decentralized AHUs that take outside air through the facade are directly influenced by the wind-made pressure distribution. Although the air flow around a building is usually turbulent and hence fully 3D and unsteady, for the study of the network a steady assumption was made. An averaged pressure profile was assumed leading to the pressure boundaries used in the simulation. The influence of non-homogeneous
pressure boundaries on the air distribution in the system was of more interest than the dynamic behavior of the wind. When a constant static pressure in the inside of the building is assumed AHUs that supply air from the negative pressure side of a building have to overcome a larger pressure difference than those from the positive pressure side. This leads to an uneven air supply to the building if not corrected by the fans. How a network supply structure connecting the decentralized AHU from the periphery could improve the distribution properties of the system was analyzed in the first study presented in the AIVC paper.

**Air Flow Distribution Study (AIVC-Paper)**

A selected network solution was directly compared to a classical decentralized ventilation system with a unconnected supply structure. A test case building was defined for the comparison of the two decentralized supply approaches. Both setups use the same amount of AHUs and only differ by their physical topology used for the ducting. The distribution quality of a system was not based directly on the spatial distribution of air flows but on the flow rate distribution among the openings used. There is of course a close correlation between the spatial air flow distribution and the one among the openings, but it is not necessarily a one to one correlation, and depends again on the physical network topology.

When no wind was assumed, the unconnected structure showed the most uniform distribution. This was because of the almost equal length of all ducts connected to the AHUs. In the network solution the more complex structure of the ducting leads to uneven distances the air has to travel from the AHUs to the different openings. When wind was acting on the building a much more uniform distribution pattern could be observed for the network solution. Changing the wind speeds in the simulation also showed that the sensitivity to wind could be greatly reduced using a network. An uneven distribution of the supply air in the building can cause problems especially in the case of single-space offices. Assuming such an office space being served by a single AHU: When, for instance, a negative pressure of 50 Pa is present the AHU will not be able to supply the same amount of air to the office any more. In order to receive the same amount of air as in the neutral case, a negative pressure of the same magnitude, i.e. 50 Pa would have to be applied from the inside of the office by the exhaust system. The pressure difference created relative to the rest of the building would then cause problems of discomfort in terms of draughts and noise and eventually would make it difficult to open the doors. With a network solution, supply openings between different rooms are connected. This
means that pressure differences between rooms are automatically balanced and none of the above problems can occur. The study showed that with a minute negative pressure of only 1 Pa applied by the exhaust fan, the flow rate in an office being located at a negative pressure side can be restored to a high fraction of the undisturbed value.

Extended Network Topology Comparison

A more extensive study of several looped networks was carried out in order to get a better understanding of the characteristics of these networks. They were composed using different connection patterns and different compositions of components. For some networks, openings are the only node types used, for others, a combination of junctions and openings were chosen. Another criterion used during the network generation is the degree of connectivity of the pump nodes. It represents the number of pipes being directly connected to the AHU and has been varied between one and three. Simulations were run for different boundary conditions, i.e. homogeneous and variable pressure distributions again representing the influence of the wind.

Several quality criteria have been defined to judge the different solutions and make them comparable to each other. These include the total flow rate of a network, the standard deviation of the flow rate distribution among the openings and a relative deviation of the minimum and maximum flow rates from the mean value. Another measure used is the spread between the minimum and maximum flow rates respectively for different wind conditions. Also included were topology measures being the total amount of nodes used and the network density which relates the total amount of piping used to serve a given area. For the final evaluation normalized values for quality criteria and weighting factors were used. The result of the evaluation proved to be robust, i.e. there was only a low sensitivity to changes in the weighting. The best four solutions out of nine considered remained the same even when the weighting was removed. As it was already seen in the previous study, the worst solution under the influence of wind was the unconnected topology. For the different network solutions under consideration a redistribution of flow rates could be observed through the change of flow directions under the influence of wind. It was found that the use of junctions in addition to openings can help to reach a better supply of the core zone of the building away from the periphery where the AHUs are located. At the same time the introduction of simple junctions increases the back pressure for the connected openings, and hence can cause an excess flow at these openings. Complex networks with many levels of junction/node connections were found to have a rather rigid pressure
characteristic such that no change in routing of air flows with wind can be observed. It can be concluded that the behavior of the network is sensitive to the complexity chosen in terms of openings/junctions and the connection pattern being chosen. The distribution capability also largely depends on the size of the network, i.e. the size of the ventilated space. Although junctions seem to help to bring the supply air further into the deep space, it has to be recognized that the very simple rectangular grid solution with only openings being used, led to a more regular air distribution than other more complex networks did.

It was found that using decentralized AHUs together with a highly interlaced ducting network allows the removal of some of the problems encountered in a regular decentralized system with direct release of the air at the periphery. Through the networking, pressure differences that build up between different rooms can be balanced out and a regular distribution, even under variable pressure conditions caused by the wind, can be guaranteed. Unlike regular systems that try to compensate pressure fluctuations by regulating fan power, the system described here, using a networked supply structure, tries to balance out flow rate anomalies by distribution means. Zones with less air supply from the outside because of the negative pressure boundary are supported from neighboring zones with an excess of supply air available. This approach reveals a new potential for energy savings that was studied and presented in the PLEA paper.

Hybrid Operation and Potential Power Reduction (PLEA-Paper)

The possibility for a hybrid operation of the ventilation system with wind support was evaluated. Besides the energy savings, the potential reduction of the exergy demand when running in a hybrid mode was considered. In order to be able to use the wind forces to assist the ventilation, the amount of the air supplied from the positive pressure side must be maximized while the amount from the negative pressure side needs to be minimized. In the presented study, a 40% over-capacity of AHUs has been used for the optimization in order to be able to selectively switch off some of the fans.

Besides the optimization based on wind pressure other parameters such as the temperature can also be used to achieve further reduction of the energy usage. In summer time the amount of air supplied from zones of the building that are directly exposed to the sunlight and hence would contribute largely to the cooling load should be minimized. The opposite is true in winter when the heating load should be minimized. The networked supply system together with its over-capacity of AHUs allows for a very flexible allocation of fan
power while maintaining a regular air distribution and allows the building to
dynamically react to changing boundary conditions such that it always runs
with an "optimal" performance.

Evaluating the electrical power usage of the decentralized ventilation sys-
tem under consideration, it was found to be 4 times smaller than the one
expected for using a centralized equipment. This is because of the signifi-
cantly lower pressure losses occurring in the decentralized system due to the
short transport distances and the low air flow speeds encountered. In central-
ized plants all air is handled by a single point and is than distributed using
throttling. The pressure losses in the system consequently increase with the
size of the building. Using a decentralized supply, each floor is separate and
the AHUs are working in parallel such that the losses do not sum up and
remain very low even for very tall buildings.

Taking into account the possibility of running the decentralized ventila-
tion system in a hybrid mode revealed further saving potential. For a selected
network with a 40% fan over-capacity and a wind speed of 10 m/s, a 25% re-
duction of fan power could additionally be achieved using the hybrid approach.
The savings could further be increased by using an even larger amount of over-
capacity of AHUs but at some point there are restrictions from a topological
as well as from an economical point of view. The amount of AHUs in the
network cannot be altered independently of the amount of the other network
nodes without affecting the fundamental network pattern.

Not only energy, but also the exergy performance of the supply system was
analyzed. For this purpose the exergy destruction in every single component
was examined for both heating and cooling mode. This analysis identified the
location of the largest exergy losses in an air handling system. The largest
exergy losses for the heating situation occur in the heat exchanger due to the
heat transfer process across a finite temperature difference between the water
and the air. The AHUs analyzed in this work show quite moderate exergy
losses because they are designed in such a way that they can operate with
small temperature spreads between the water and the air. Other significant
exergy losses occur in the fan. The fan uses the electrical power to convert
it into kinetic energy of the fluid transported. Some of the exergy input is
needed to overcome the pressure losses in the system and cannot be used
to accelerate the air. The largest losses in the fan though occur due to low
overall efficiency of small fans which is only around 20%. Energy savings as
reported above that are due to a reduction of pressure losses in the system
lead to commensurate exergy savings. This is because of the direct correlation
of the pressure loss and the fan power requirement. It is of course not true for
thermal losses in the system. The exergy losses occurring in the heat exchanger
are temperature dependent such that a change of the air temperature at the entrance of the AHU directly affects the degree of entropy generation and hence exergy destruction.

Similar to the above results for the energy savings the total exergy savings in the system can be quantified. For the system setup chosen, switching from a centralized supply to a decentralized leads to a fan power reduction by a factor of 4.16 which corresponds to a 75.9% savings. Including further fan power reduction potential from using a pressure based hybrid mode leads to a total reduction factor for electricity of 5.3 corresponding to a 81.2% reduction. Looking at the temperature based hybrid operation mode assuming one facade of the test case building to take a temperature of 12 degrees above environmental air temperature along with a 40% fan-overcapacity selected, a reduction of exergy demand of 6% and 40% results during heating and cooling respectively. These reductions of the exergy demand in the AHUs proportionally reduce the exergy load of the heat pump system. Through the definition of the exergetic efficiency, the exergy load of the heat pump is also directly related to its electricity input. As a consequence the final demand of high grade energy by the heat pump system used for conditioning of the outside air, can also be lowered by the same 6% and 40% respectively.

7.2.3 Control Strategies with Variable Exhaust

The issue of control when combining a DCV approach with decentralized supply was studied. Different scenarios were considered together with different control approaches. The contaminant concentration detected in the exhaust air stream is what determines the total ventilation activity. When a local control scheme is used the amount of ventilation does not only vary in time but also in space. In some places ventilation rate may be high while in other places it may be very low.

When taking a steady, flow-balanced ventilation with constant rates as a reference, the local demand control of ventilation activity can be considered as a disturbance to the system. A disturbance introduced from the exhaust side of the system has to counteracted by the supply system for continuity reasons to avoid build up of pressure fluctuations in the room. Besides the variable exhaust flow rates, a non-homogeneous pressure distribution around the building due to the influence of wind also has to be considered as a disturbance to the ventilation system. A decentralized supply system with AHUs being directly installed in the facade is prone to pressure disturbances and reacts with significant flow rate deflections. Again, these disturbances have to be balanced out by the supply system.
Four different control strategies for the AHUs together with the centralized exhaust fan have been considered. The overall system behavior was inferred under the influence of several disturbances from the local control strategy chosen. Besides a ”regular operation” of the ventilation system where it should balance all disturbances and maintain a steady operation, an ”optimized operation” of the system was also considered. Latter operation mode was serving the idea of further reducing energy usage through using natural driving forces to run the system in a hybrid way.

The most straightforward and also most robust control is based on flow rate for both, the exhaust fan and the AHUs. In this case a fixed flow rate has to be assigned to each autonomously controlled exhaust flap such that the total exhaust flow rate is given by the number of flaps simultaneously being opened. The total flow rate in the exhaust is then distributed among the AHUs. The AHUs use constant flow rate control to withstand pressure fluctuations originating from the wind. A constant flow rate control of the AHUs can also be used together with a constant pressure control of the exhaust fan. In this case the exhaust flaps act as variable flow resistances in the ducting such that according to their position open/closed the fan power is adjusted. In such a setup an additional measurement of the flow rate in the exhaust branch of a zone is necessary in order to know the flow rate needed from the AHUs in that same zone. The pressure control of the exhaust leads to a different allocation of the exhaust power than is the case with a constant flow rate control. The combination of both, a constant pressure control for the AHUs and the exhaust is not possible because it leads to an indefinite system state and does not guarantee a stable operation. Instead, a constant pressure control of the AHUs together with a constant flow rate control of the exhaust fan was considered. This setup requires a calibration measurement in order to find the setpoint of the differential pressure for AHU control. Using a highly interlaced ducting network between the AHUs with a very flat loss characteristics, the local negative pressure that will occur in the room due to an exhaust flow rate above or below the calibration value is very low, and was found to be less than 7 Pa. The constant pressure control of the supply is particularly interesting for the operation of the system in hybrid mode using wind pressures. Using this approach, the wind automatically causes AHUs on the pressure side to maximize and those on the negative pressure side to minimize their flow rate. This increases the overall efficiency of the system, which can be expressed as the amount of air delivered, divided by the power used by the fans. The ducting network will balance out the uneven flow rate distribution among the AHUs. If a temperature based optimization is desired, the constant pressure control of AHUs is not suitable. Rather, the AHUs have
to be flow rate controlled, and the distribution of the flow rates among the AHUs has to be determined by an additional control unit. For this purpose, this unit has to be aware of the local conditions encountered by all AHUs. The optimization can then either be carried out using an efficiency measure related to the pressure or temperature condition.
# Appendix A

## Network Evaluation

### A.1 Cost Function Evaluation for Different ADNs

Figure A.1: Table showing the calculation of the ranking based on the cost function evaluated for every network.

<table>
<thead>
<tr>
<th>Criteria</th>
<th>Ranking of topologies</th>
<th>Weights</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>unconnected</td>
<td>grid open</td>
</tr>
<tr>
<td>Total flow rate 0 m/s</td>
<td>1.000</td>
<td>0.506</td>
</tr>
<tr>
<td>Total flow rate 10m/s</td>
<td>1.000</td>
<td>0.511</td>
</tr>
<tr>
<td>absolute spread max-min, 0 m/s</td>
<td>0.000</td>
<td>0.229</td>
</tr>
<tr>
<td>absolute spread max-min, 10 m/s</td>
<td>1.000</td>
<td>0.022</td>
</tr>
<tr>
<td>standard deviation, 0 m/s</td>
<td>0.000</td>
<td>0.313</td>
</tr>
<tr>
<td>standard deviation, 10 m/s</td>
<td>1.000</td>
<td>0.032</td>
</tr>
<tr>
<td>spread min-min</td>
<td>1.000</td>
<td>0.041</td>
</tr>
<tr>
<td>spread max-max</td>
<td>1.000</td>
<td>0.131</td>
</tr>
<tr>
<td>rel deviation min/mean, 0 m/s</td>
<td>0.000</td>
<td>0.216</td>
</tr>
<tr>
<td>rel deviation min/mean, 10 m/s</td>
<td>0.098</td>
<td>0.135</td>
</tr>
<tr>
<td>rel deviation max/mean, 0 m/s</td>
<td>0.000</td>
<td>0.129</td>
</tr>
<tr>
<td>rel deviation max/mean, 10 m/s</td>
<td>0.880</td>
<td>0.133</td>
</tr>
<tr>
<td>network density</td>
<td>0.000</td>
<td>0.521</td>
</tr>
<tr>
<td>number of nodes</td>
<td>0.000</td>
<td>0.167</td>
</tr>
</tbody>
</table>

| Costs    | 0.48 | 0.18 | 0.37 | 0.18 | 0.14 | 0.31 | 0.58 | 0.32 | 0.17 | 30 |
| Ranking  | 7    | 3    | 5    | 4    | 1    | 4    | 8    | 5    | 2   | 194 |
A.2 Air Flow Distribution

Figure A.2: Air flow distribution simulated for the unconnected, reference topology
Figure A.3: Air flow distribution simulated for the network 2perpump

Figure A.4: Air flow distribution simulated for the network 3perpump
Figure A.5: Air flow distribution simulated for the network \textit{brick\_junction}

Figure A.6: Air flow distribution simulated for the network \textit{grid\_junction}
CURRICULUM VITAE

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