AIR
Swegon Air Academy

Part E
Air treatment and indoor climate

DUNCAN:
THIS
CASTLE
HATH
A PLEASANT
SEAT;
THE
AIR
NIMBLY
AND
SWEETLY
RECOMMENDS
ITSELF
UNTO
OUR
GENTLE
SENSES

[William Shakespeare *Macbeth*]

CONTENTS

FOREWORD

Our need for fresh air, essential to our functioning as human beings, is not normally contested by anyone. This is because we have basic physiological needs - om brains and the cells in our bodies need oxygen so that they can develop and perform properly. However, the air we breathe contains more or less harmful substances and these cause more problems than most of us can probably imagine or understand. On their own, these substances might be troublesome - but collectively, they could be disastrous! Remember, that while we need about 0.75 kg of food and about 1.5 kg of liquid per day, we need at least 15 kg of air!

It is quite reasonable to compare an air handling system in a building to our own respiratory system with its airways and lungs, as both systems have enormous significance for our health and well-being. And, as the air is often used to supply or remove heat, this makes the importance of the air handling system even greater, as it significantly affects our comfort, well-being, productivity and efficiency. Buildings, too, need a continuous change of air to feel good.

LACK OF COMMON POINT OF VIEW

Bearing in mind all of the above, it is rather odd that those involved in lanning a building rarely see things from the same point of view. Short-term economic interests ~re often allowed to determine the choice of technical solutions and, when costs are not critical, buildings are all too often designed in such a way that they are neither pleasant to occupy nor energy-efficient. And, up to now, it has been rather difficult to acceptfeedback and learn from expensive mistakes, and thereby avoid repeating them.

PUBLIC HEALTH, ENERGY AND ENVIRONMENTAL ISSUES - AND ECONOMIC REALITIES

This book focuses on three main areas: Public health, energy and the environment. We are also convinced that economical aspects must be considered as well and this is a recurring topic throughout the book. No matter how efficient and health-promoting an investment might look, it would most probably never be carried out if it were not shown to be economically viable. Today, reliable research results show that there is a clear connection between poor indoor climate and ill-health. And ill-health costs a great deal of money. In other words, there is a real incentive for property owners to invest in good indoor climates now, as future tenants will almost certainly step up their demands.

CONTROL, REGULATION AND MONITORING - THE DECIDING FACTORS

In this electronic age, it is becoming increasingly obvious that control, regulation and monitoring will play a decisive role when it comes to maintaining good indoor climates and ensuring energy-efficient operation of buildings, with subsequent minimal impact on the outdoor environment. The importance of providing solutions that give the clientmaximum freedom of choice and flexibility is illustrated in Chapter 31/The intelligent building- a matter of choice, which discusses centralized building management systems, so-called BMS systems, for control, monitoring and management.

COMPLEX RELATIONSHIPS

- SIMPLE EXPLANATIONS

In this book, *Swegon Air Academy* has compiled ideas and points of view from a wide range of experts. The aim of the book is to put a spotlight on factors and circumstances that are important in the quest for pleasing - indoor environments and comfortable indoor climates, with due regard to energy issues and the outdoor environment. Our ambition has been to explain complex relationships in an intelligible way. It is our profound belief that it is possible to radically improve poorly functioning systems - if we can explain the whys and wherefores.

The passage of air through an air handling system is described, from the outdoor air intake, via an air-conditioned room and into our lungs, with a full account of what happens on the way. The physiological aspects, as well as the comfort, energy and environmental aspects, are examined. How different building designs affect the opportunities for creating good indoor climates is also discussed. Here, the effects that different factors have on each other are not always self-evident or discernable, nor are their specific effects on the indoor climate and total costs. We have, therefore, chosen to illustrate a number of them in greater detail and hope that this will contribute to future developments, for the benefit of all concerned.

TARGET GROUPS

This book is intended not only for clients, property owners and engin-eers, who can influence the design, layout and indoor climate of a build-ing, but also for everyone who would like to learn more about the air we breathe and how it affects us.

Increased insight will make it possible to avoid unnecessary costs, both in the investment stage and in the operational stage of a building project. Attractive premises are a must for survival on a competitive property market.

THE AUTHORS

This book blends theoretical knowledge from the academic world with practical market experience. Our ambition has been to portray the present-day situation and the opportunities in store in an objective and unprejudiced way, by engaging highly distinguished experts and writers from a representative cross-section of the industry.

Proceeds from the sales of this book will be reinvested in the activities run by *Swegon Air Academy,* i.e. in objective transfer of know-how and exchange of information via seminars, technical articles and publications. The contents of this book are available to schools and training programmes connected to the heating and ventilation industry at a subsidized rate.

Enjoy the book!

CONNY NILSSON
Director of the *Swegon Air Academy*

[AIR *Swegon Air Academy*]

AIR

Swegon Air Academy

Part E

Air treatment and indoor climate

E. AIR HANDLING SYSTEMS AND THE INDOOR CLIMATE

CONNY NILSSON *Swegon Air Academy*

This part of the book traces the path of fresh air as it is transported from an outdoor to an indoor environment via an air handling system. Something happens to it at every stage of its journey- primarily to make it as pleasant and as suitable as possible for its intended purposes. Some factors, however, can have negative effects, if they are not taken into account in a serious manner. We'll be taking a closer look at air intakes, filters, heat recovery systems, heating and cooling units, humidification and dehumidification, fans and sound absorbers, ducting systems, airborne and waterborne indoor climate systems, control systems, balancing and commissioning, building management systems and quality-assured indoor climates, and how they all can have an effect on the indoor environment in a building.

When it comes to handling air, and just as in many other disciplines, it is important to express oneself clearly and concisely when using certain words and expressions. Unequivocal definitions are a must to avoid misunderstandings. In each chapter, relevant definitions are included to help clarify meanings and facilitate understanding.

Readers might comment that some important aspects have been left out, both here and in other parts of the book. This is true and it would, of course, have been fantastic, if we could have published a book that covered every aspect of all related technologies. For practical reasons, this would have been an impossible task, not least for reasons of space. This means that we have left out everything that has to do with fire cells and how fire and combustion gases can be spread between them. Nor have we provided recommendations about how much ventilation is required in different types of buildings or for their different uses. This book is aimed at a number of international markets in which national ventilation regulations and standards vary considerably. Furthermore, there is a great diversity of climate conditions, both within and between different countries, which also makes it difficult to formulate general recommendations. On the other hand, the book does discuss the question of the size of hygienic air flows in Chapter 5 /Legislation, standards and established practice.

Each system or component discussed in this part of the book has its own special purpose. We have not taken sides for or against any particular solution or technology. Every solution is applicable in its own right, although no solution is universal enough to warrant its use in every conceivable situation. All systems and components are, however, expected to work in unison, but all too often one category of fundamental aspects is overlooked - the demands placed by the client. If we are to create a good indoor climate, one that corresponds to the client's requirements and needs, then these must be very clearly stipulated from the very beginning. The design and structure of a building, as well as its intended use, are two more crucial factors that we must come to terms with, before we start to consider which solution will best meet the client's expectations. Far too often a project is tackled from the wrong end - with a product or technical solution that we then try to apply, as skilfully as we can, to fulfil the requirements. In the best of worlds, it will only offer a reasonably efficient compromise. Choosing the wrong starting point often leads to poorly functioning systems - no matter how high-quality the products are that are installed. The first chapter, about the client and the building process, is fundamental to the following discussions in this part ofthe book.

14. THE CLIENT AND THE BUILDING PROCESS

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BACKGROUND

Although the whole building process, from the moment a new building is envisaged until it is completely finished, may be found interesting and sometimes even exciting, it is always found to be very complicated. The client starts the ball rolling and defines the conditions for the project: the purpose of the building and its location, size and cost. Then more and more parties become involved: local authorities, design engineers, contractors etc. When rebuilding, extending and renovating most of the conditions that will affect the project will be dictated by the existing building. However, if the changes are extensive, the initial conditions will be similar to those for new buildings.

In small projects, for example, when building detached houses, there is often only one client and a single contractor who oversees everything. In large projects, such as blocks of flats, offices, schools, hospitals and industrial buildings, a great number of players are involved: architects, design engineers, project managers, tenant's representatives, contractors etc.

In both small and large projects, the client is ultimately responsible and must make all the final decisions during the building process. The client is responsible for stipulating the physical and economical limits for the project as well as the properties of the building.

All those involved in the project – the architect, the design engineer and the contractors – are, in turn, responsible to the client for delivering a building that functionally, aesthetically and technically displays the properties that the client has stipulated. One of the conditions, therefore, that must be fulfilled is that these properties must be clearly defined by the client so that all those involved fully understand them. This is done by expressing them in a requirements specification. Those involved in the project must then design the building and choose, design and install the building services solutions so that the building fulfils these requirements.

The realms of responsibility between the client and others involved in the project must be clearly defined and adhered to during the whole of the building process. This is because it is important to draw a clear demarcation line between requirements and solutions. The client can, of course, have an opinion about the choice of certain solutions and might even demand special solutions, but this must only be done if they can actually fulfil all the requirements regarding properties and functions. Otherwise, it could be difficult to point out who was responsible for fulfilling a particular requirement and this could cause problems if, for example, mistakes are made or faults occur.

Before the actual planning work starts and before any binding decisions are made, the client should have a good idea of the economical and technical consequences of the different requirements. Similarly, the other players involved in the project must be fully convinced that it will be possible, within reason, to fulfil the demands made by the client within the technical and economical limits of the project. When planning a new building, for example, at the preliminary architectural design stage, an unbiased feasibility study should be carried out to determine whether the client's requirements can be fulfilled within the technical and economical limits of the project.

Requirements and requirements specifications, the difference between requirements and technical solutions, and the importance of carrying out consequence analyses at an early stage in a project are discussed in the following pages of this chapter.

REQUIREMENTS

Above all, a building must be designed so that it is safe to use. Those who live or work in the building must never risk being injured by a part of the building collapsing on them or becoming ill because of a poor indoor climate. There are always a number of laws and official regulations that must be followed when constructing a new building, or altering an existing one, and these are primarily aimed at ensuring that it is designed and dimensioned so that those using it will not risk being injured or harmed in any way. Demands regarding energy efficiency and environmental impact are also being introduced to an increasingly greater extent.

Regulations often state basic performance levels that must be fulfilled.

However, requirements for individual projects are often higher, if there are advanced demands regarding the use of the building. This is especially true in non-residential buildings, where usage often requires stricter limits than those stipulated in official regulations.

The reason why new buildings are built, or existing buildings are rebuilt or extended, is because they are needed for a specific purpose or use.

There are always a number of basic *usage requirements* that must be fulfilled if the building is to be used at all. There are specific requirements, for instance, concerning the layout of the building so that it can be lived in or used for work. There are also functional requirements, for instance, that the temperature of the room air must lie between certain limits, that the air quality is good, that the lighting level is suitable for the activities in the building, that there are no annoying noises or disturbing electrical fields, etc. All these requirements must be fulfilled, if the building is to be used as intended.

However, these requirements must not just be fulfilled now, while the building is new, but also for years to come. The building must be designed to last: it must be able to withstand moisture, it must not contain individual materials or combinations of materials that might deteriorate over time, and the building services installations must continue to function while remaining cost-efficient. Requirements like these are always included in the specifications concerned with the usage of the building.

The specifications relating to the usage of the building are, on their own, not enough to ensure that the building will be satisfactory from all points of view. The building must also be architecturally attractive, only require a reasonable amount of maintenance, use little energy, etc. These properties, which are really an expression of the quality of a building, are stipulated by the client in the form of a *building requirements* specification. Quite naturally, it is often these that are focused on when planning work starts and even later on during the detailed planning work. The usage requirements specifications, which are often more abstract, tend to be disregarded. And this is where demands regarding the indoor climate are to be found. The very idea behind dividing the requirements specifications into two categories – building and usage – is to emphasize the importance and relevance of the usage requirements.

There is no difference in value between the two different categories and this is why focus on the building requirements must never mean that the usage requirements will not be fulfilled.

There are, then, three different types of requirements and all of them must be fulfilled in the finished building:

- Official requirements
- Usage requirements
- Building requirements

Building regulations often indicate a basic level for performance requirements in residential buildings, both concerning the structure of the building as well as installations to provide a suitable indoor climate. However, it is sometimes motivated to draw up a separate requirements specification as a complement the formal regulations if, for example, the building has been designed in such a way that the indoor climate will be subject to large loads. This could be necessary, for example, when there are large window areas without extensive solar protection.

When it comes to large buildings, and especially non-residential buildings, it is very important that a well-prepared requirements specification is drawn up. Renovation work, if carried out to a reasonably large extent, also falls into this category. And, although the client is responsible for drawing up a requirements specification, consultants can, of course, be engaged but the final responsibility still lies with the client.

Official regulations cannot be questioned whereas the two other categories can be discussed further.

USAGE REQUIREMENTS AND BUILDING REQUIREMENTS

There are two types of usage requirements, both of which must be fulfilled so that the building can be used for its intended purpose:

- Those which relate to the building itself, i.e. those which specify sizes, areas and numbers of elements etc. These requirements are used to describe the building and quantify design parameters.
- Those which relate to the indoor climate, i.e. those which are specified in terms of physical units such as temperature, highest allowable concentrations of pollutants, illuminance etc.

The usage requirements can normally be expressed in figures and can, therefore, be used as a basis for planning. They can be easily followed up during the planning process and checked in the finished building. The building requirements should also be expressed, as far as possible, as specific quantities. As shown in the example below, a number of the building requirements can only be expressed qualitatively.

Examples of performance requirements

Usage requirements	**Building requirements**
Relating to the building • Number of rooms • Areas of rooms • Types of rooms • Communication routes *Relating to the indoor climate* Desired functions • Admission of daylight • Thermal climate • Air quality • Lighting quality • Acoustics • Barriers to prevent the spread of particles and gases (laboratories, hospitals) • Hygienic design *Relating to nuisances* • Draughts • Noise • Electrical fields • Dazzle/glare *Relating to durability* • Moisture protection • Operating reliability	*Requirements formulated to ensure:* • Aesthetic solutions • Efficient use of space and numerical values for assessment • Floor space coordination • Insulation and airtightness of the building envelope • Quality, durability and ease of maintenance, including assessment criteria • General applicability and flexibility, including assessment criteria • Ecological solutions and recycling • Technically and economically optimal solutions, and criteria for optimisation • Minimized life cycle costs with criteria for assessment • Energy efficiency and criteria for assessment

In the remainder of the chapter, when requirements and requirements specifications are discussed, focus will be on the indoor climate and the use of energy.

THE TERM 'REQUIREMENTS'

If a requirements specification is to be justified at all and the document is to be regarded as a fundamental ingredient, essential to the project, then all those who are engaged in the building process must be fully aware that the requirements:

- *are absolute*, i.e. they must be fulfilled completely within stipulated limits.
- *must be verifiable and measurable* when the building has been completed and put into use.

- *must be well founded*, i.e. there must be indisputable grounds for every requirement.
- *must be used as a basis for planning* and that they must be possible to fulfil, taking into account the conditions under which the project is carried out.

These points must be strictly adhered to, not least when it comes to the requirements that specify the indoor climate.

Requirements are absolute

If a requirement is absolute, it means that it must be fulfilled completely. The word requirement is itself absolute and it is therefore used together with the verb 'must'. It is quite unacceptable to stipulate requirements and then express them using the verb 'should', which is, unfortunately, often the case.

A prerequisite for achieving the desired indoor climate and expected energy efficiency in a building project is that there must be no ambiguities in the formal communication between the client and those involved in the building process. This means that it must be fully understood that every requirement must be fulfilled completely.

If, for example, it is stipulated in a requirements specification that the temperature in a room must be +22 °C ±2 °C, this means that the room temperature must never exceed +24 °C. This will require a very expensive and advanced indoor climate system, especially if the requirement is to be fulfilled in outer rooms. However, if it can be accepted that the room temperature is higher than specified for short periods, the requirement can be fulfilled using considerably simpler plant. In this case, the temperature requirement must be expressed quite differently, for example, as follows:

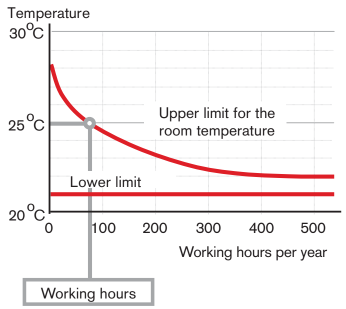
Example 1

The room temperature must be +22 °C ±2 °C, but higher temperatures can be accepted when the outdoor temperature is higher than the design outdoor temperature.

Example 2

The room temperature must be +22 °C ±2 °C, but higher temperatures can be accepted for 80 hours per year.

Example 3
During working hours, the room temperature must always be kept below an upper limit, defined by a duration diagram, and above a stipulated lower limit.



The figure shows how the requirements stipulated in Example 3 can be defined.

The climate system must be designed so that:
- The room temperature does not exceed +25 °C for more than 80 working hours per year.
- The room temperature must always be kept above +21 °C.

The requirements stipulated in Example 3 mean that the climate system must also be designed so that the temperature can be kept at around +22 °C, if desired. A common problem that occurs in modern office buildings is that it is seldom possible to lower room temperatures much below the maximum allowable level, even in winter.

The importance of being aware of the fact that requirements are absolute can be illustrated by the following: The highest concentrations of, for example, nitrogen monoxide, nitrogen dioxide, sulphur dioxide, ozone, etc, in the indoor air are sometimes stipulated in the requirements specifications. In cases like these, it is important that the stipulated limit values are not lower than those in the outdoor air. Otherwise, the requirement will mean that these gases will have to be removed from the supply air and this will be prohibitively expensive.

It is the sole responsibility of the client to formulate and compile the requirements to be fulfilled by all the parties involved in the project. As mentioned previously, the client can consult specialists when drawing up the requirements specification, but they will still remain the client's responsibility.

Requirements must be verifiable and measurable
Requirements are only meaningful if fulfilment can verified. And there must be no doubt about how this is to be done. If the measuring method to be used is not perfectly obvious, then it should be described in an appendix to the requirements specification.

It must be assumed that each and every requirement will be checked to ascertain whether it has been fulfilled. This work must be possible to complete by taking a reasonable number of measurements in the finished building. Although checks are not always necessary, it must still be possible to carry them out.

This is one of the reasons why parameters such as the PPD Index (the percentage dissatisfied with the thermal climate) and the DR Index (percentage dissatisfied with the level of draughts), [EN ISO 7726] and [EN ISO 7730], should not be included in a requirements specification. Measuring the percentage of dissatisfied users with acceptable accuracy requires a great deal of work, with questionnaires that have to be filled in and analysed. This takes so much time and is so costly that it is, in practice, impossible for the client to demand amendments if the requirements are not fulfilled.

The same situation will arise if demands are made regarding the air change efficiency and the ventilation efficiency, [Mundt et al., 2004]. Checks based on measurements in these cases are so complicated that requirements tend to be meaningless, except in special cases, such as in high rooms, large halls and residential buildings with airborne heating systems.

Requirements must be well-founded
As requirements are absolute and must be fulfilled completely it is important that they reflect actual needs. It is essential that the client is well-informed about the technical and economical implications of every individual requirement. This is discussed below in the section on consequence analysis.

Requirements must be used as a basis for planning
It must be possible to propose solutions that are based on the stipulated requirements. This means that the requirements must be defined using standard physical units that can be expressed quantitatively. The PPD Index and the DR Index are, however, not suitable for use in a requirements specification, as mentioned above. On the other hand, they are useful indicators to the client when drawing up the requirements specification and choosing limit levels. Thermal climates must be defined using units such as temperature in °C, air speeds in m/s, etc.

EXAMPLES OF REQUIREMENTS SPECIFICATIONS

The first example refers to usage requirements related to the indoor climate in offices. The second example refers to building requirements related to energy usage.

Every country has its own basic requirements for residential buildings. In Sweden, there are also well-defined requirements regarding indoor climates in office buildings, [Ekberg, 2007] and there are a number of other sources with similar recommendations, for example, [EN 15251, 2007], [CEN CR 1752, 1998], [ASHRAE Standard 55, 2004] and [ISIAQ-CIB, Report 292, 2004].

For other types of non-residential buildings there might also be detailed requirements that must be fulfilled and these are stipulated in the relevant publications for each building type.

To provide an idea of how a requirements specification can be drawn up, an example is given for an office building, see Table 1. The limit values shown here have been adopted by a large number of Swedish property companies.

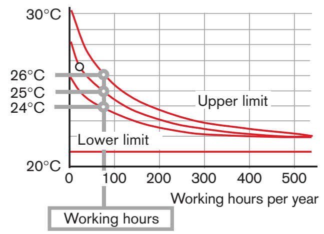
The example shows which parameters should be included when addressing different climate factors. The requirements specification should include the following:

- Climate factors
- Limit values
- References that either state why the specific limit value has been chosen or a source reference
- Follow up methods that state how to check that the requirements have been fulfilled

The limit values in Table 1 are recommendations. It might be necessary to have stricter limits regarding noise, for example, 5dB(A) and 5dB(C) lower than the values given. It is, of course, up to the client to specify other limit values than those in the recommendation.

Today, requirements regarding energy efficiency are some of the most important requirements with respect to the performance of a building. And they will, most probably, become even more important. The following list shows some of the present requirements for office buildings stipulated by Akademiska Hus, a major Swedish property company, which owns and manages most of the university buildings in the country. The requirements are given as limit values with respect to the size of the building or the indoor climate system. The size of the building in this

TABLE 1. An example of a requirements specification for conference rooms.

Climate factor	Requirement/ Limit value	Reference	Follow up
Carbon dioxide CO_2	<1 000ppm	Current standard or recommendation	Applicable control method, for example, according to ISO 8760:1990 Workplace atmospheres
Radon	<200 Bq/m^3	Current standard or recommendation	Applicable control method
Formaldehyde 0.5h mean value	<0.05 mg/m^3	Current standard or recommendation	Applicable control method
Temperature	One of the levels in the diagram below  		Kontrollmetod
Air speed in the occupied zone Wintertime Summertime	 <0.18 m/s <0.22 m/s	Current standard or recommendation	Applicable control method
Noise from building services Individual office Large room	 35 dB(A)/55 dB(C) 40 dB(A)/55 dB(C)	Current standard or recommendation	Applicable control method
Lighting Dazzle/glare Luminance Intensity of illumination Contrast reduction Colour reproduction Colour temperature Flicker-free lighting	 Shaded lighting max 1000/ 2000 cd/m^2 500 lux max 15% Ra index ≥ 80 2700 to 4000 K HF fitting	Current standard or recommendation	Applicable control method
EMF	<0,2 µT, <10 V/m	Current standard or recommendation	Applicable control method

context is defined as the GFA, the gross floor area of the building enclosed by the exterior walls.

The limit values for the following key indicators must not be exceeded when the building is in use:

Indoor climate system	
Electrical power/design air flow (SFP)	1.5 kW/(m^3/s)
Electrical power/design cooling power	0.3 kW_{el}/kW_{cool}
Heating power/design air flow	10 kW_{heat}/(m^3/s)
Cooling power/gross floor area	40 W_{cool}/m^2_{GFA}
Lighting system	
Electrical power/gross floor area	6 W_{el}/m^2_{GFA}
Whole building, annual usage	
Electrical energy in buildings with own cooling system	60 kWh_{el}/m^2_{GFA}
Electrical energy in buildings with district cooling	50 kWh_{el}/m^2_{GFA}
Cooling energy in buildings with district cooling	30 kWh_{cool}/m^2_{GFA}
Heat energy (excluding domestic hot water)	30 kWh_{heat}/m^2_{GFA}
Heat energy (including domestic hot water)	35 kWh_{heat}/m^2_{GFA}

Continual efforts are being made to lower the limit values of these key energy indicators, for example, specific fan power, SFP, ratings. In ventilation systems installed in the 1980s, or earlier, the limit SFP rating was around 4 kW/(m^3/s). It was relatively simple to reduce this to just over 2 kW/(m^3/s), now specified for new systems. For a number of years it was thought that this was the lowest realistic value that could be used, if space requirements and costs for air handling equipment were also taken into account.

There have also been some positive side effects of the demands for lower SFPs. Among other things, noise from ventilation systems has been almost totally eliminated thanks to lower fan speeds and lower pressure drops. It can also be seen that a 1.5 kW/(m^3/s) limit is also possible, if systems are carefully planned. And even lower limits are being discussed.

TECHNICAL SOLUTIONS

The step from a requirements specification to a fully functioning building is accomplished via applying technical solutions. Technical solutions comprise system solutions and technical principles as well as detail design. If the design and planning work is to be carried out efficiently, it is essential that system solutions are chosen, and design criteria and technical principles are stipulated, before detail design work is commenced. In large building projects it is often required to draw up documents for the client that describe the proposed system solutions and include a cost estimate. Detail design is only begun when the client has given his approval.

There are always a number of technical solutions that can be adopted so that the usage requirements for a building can be met. The solutions that are chosen, however, must also meet the requirements regarding the building with respect to its architectural design, efficient use of space and minimized life cycle costs. Good planning, therefore, entails close cooperation when deciding on architectural design, structural solutions and building services solutions. The architectural design can have far reaching consequences with respect to the structural design and building services. It determines which solutions are feasible and how complex they have to be to fulfil all the requirements. The design also influences the construction and operating costs. This means that the consequences of a particular building design must be analysed at an early stage in the planning process.

Structural components often have two distinct functions, one that is practical, i.e. the forming of a barrier between the indoors and outdoors, and one that is physical, i.e. the prevention of heat flow through the building envelope. If, for example, the correct technical solution is used for a facade of a building, which is then built correctly, both its practical and physical functions will be fulfilled. An experienced design engineer will be able to discern whether a particular design solution will fulfil requirements regarding thermal insulation, moisture protection, airtightness, durability, etc.

There are two main types of building services installations – those that determine the indoor climate, the HVAC systems, and those that serve the activities in the building. Heating systems, ventilation systems and cooling systems determine the indoor climate. Domestic hot water and drain systems, and power, telephone and computer networks serve the activities in the building.

No matter whether the installations form part of the HVAC system or serve other purposes, as listed above, they all have a physical function. The HVAC systems are installed to keep room air clean and room temperatures within desired limits. They can be designed in different ways to achieve these functions but, most importantly, they must provide the room temperatures and air qualities that have been specified.

This is why it is essential to consider functionality first and then choose, design and size the solutions that best fulfil both the usage requirements and the building requirements. It is not justifiable to base choices purely on different technical solutions and then, as sometimes happens, amend the functional requirements to suit a particular solution.

Examples of technical solutions

System solutions

Building
- Types of frames and floors
- Sizing and detail design
- Thermal storage capacities of the frame and facades
- Choice of U-values for walls and ceilings/roofs
- Choice of U-values for windows
- Airtightness of the building envelope
- Choice of window sizes
- Type of sun shading
- Acoustic design and sound insulation

Climate controlling installations
- System solutions for:
  Ventilation and air treatment
  Heating
  Cooling
  Control and monitoring
  Lighting
  Daylight admission
  Water and sanitary installations
- Choice of air flows in individual rooms
- Choice of heating power in individual rooms
- Choice of cooling power in individual rooms
  Operation and control
  Regulation of air flows
  Regulation of heating power
  Regulation of cooling power

Detail design

Building
- Design and detail solutions for frames and floors
  Structural strengths
  Moisture protection
  Durability
- Detail solutions for facades
  Structural strengths
  Avoidance of thermal bridges
  Moisture protection
  Durability
- Choice of cladding, flooring, etc.

Climate controlling installations
- Design and detail solutions for building services installations
- Location in the building of
  Plant and equipment rooms
  Vertical and horizontal ducting zones
- Choice of plant/equipment and components

It is often much easier to discuss technical solutions than explain functional requirements. It is also easier for a design/project engineer to vouch for the features offered by the proposed solutions than to see that they fulfil their functions in the finished building. Both the discussions with the client and the work within the planning group, comprising the architect and the design/project engineers, often become focused on technology whereas the requirements, what the services are installed to fulfil, are relegated to second place.

Quite often, more or less complete technical solutions are presented and chosen without first investigating whether they can fulfil the usage and building requirements. These requirements are sometimes not even properly specified before choices of building services systems are made. Building techniques and building construction are all about hard facts and it is easier to look at detailed technical solutions instead of engaging in more abstract reasoning about functions.

This is where cause and effect get mixed up and become difficult to understand, with confused reasoning as a result. However, what is really serious is that:
- the building might be given other properties than those that are really important to the future tenant, and that these new properties might not have been investigated thoroughly enough.
- the result might be a solution that cannot fulfil the requirements, as these were not stipulated until after the building process had begun.
- no unbiased comparisons of different technical solutions were made and the best solution is therefore not guaranteed, i.e. the building will not function as well as it should and will be less cost-effective than it would have been with a functionally and economically optimal solution.
- technical development stagnates and follows the wrong paths, which could cause problems rather than solve them.
- if there are no clear requirements regarding function, it is difficult to hold someone responsible and to demand amends if the building does not function properly.

A consequence analysis is an important part of the planning process

CONSEQUENCE ANALYSIS

When the causes of problems in finished buildings are analysed, especially in non-residential buildings, it is often found that the demands made on the building services solutions have been reduced during planning. One reason for this is that the client might not have been fully aware of the consequences of not fulfilling some of the requirements. Another reason could have been that the architect had not fully appreciated what effect the design of the building would have on the solutions and the possibilities of fulfilling all the client's requirements. The reason could also be, quite simply, that the project group was too late in realizing, given the chosen design of the building, that the requirements could not be fulfilled within the cost limits.

The result in each of the above cases could be a poorer building than one in which the functional requirements, design, structural solutions and building services solutions had been harmonized from the beginning.

By carrying out a consequence analysis at the beginning of the project, with regard to the decisions and technical choices that have been made, the risk of emergency solutions late in the planning process can be avoided.

In practice, the design engineers, and especially those engaged in planning the HVAC systems, are the ones who must initiate and carry out a consequence analysis and make the consequences clear to both the client and the architect.

Consequence analysis of requirements

Requirements are basically absolute and serious planning work must assume that the requirements stipulated by the client must be completely fulfilled in the finished building. At the same time, the client must be fully aware of the resources needed, so that the requirements can be fulfilled.

When special requirements are specified, their consequences must be thoroughly investigated. For example, when there are demands regarding room temperatures that imply that cooling will be required, the client should be made aware that this should be unnecessary in residential buildings in northern Europe. Another requirement could be the stipulation of highest allowable levels of gaseous pollutants, such as NOx and SOx, in the room air. If the levels specified are lower than those that can be normally expected in outdoor air, then gases will have to be removed from the air and this is both complicated and costly.

In non-residential buildings the list can be extensive. The project group must, at an early stage, bring it to the client's attention if some of the requirements will have significant economical consequences, if the requirements are contradictory, or if they cannot be met using reasonable solutions. An analysis and a report concerning the consequences of the stipulated requirements should be a normal ingredient of the initial phase of the planning work. The client must be given a clear and comprehensible account of these consequences and should also be made aware of any requirement that has been overlooked.

An example of what should always be brought to the client's attention is the requirement that states that the humidity of the air must not fall below a given level. Such a requirement would demand special solutions for the building envelope and these can be expensive. It could also mean a considerable increase in energy demands and running costs. If there is a legitimate reason for maintaining a high level of humidity, then this must be taken into account in the design work, but the client must, at an early stage, be informed of the consequences and accept them.

Another example is the requirement, mentioned previously, that stipulates that the room temperature must not exceed a certain level. Such a requirement can mean the installation of an HVAC system that is both expensive and requires a great deal of space, especially if the upper permissible temperature limit is relatively low and large windows are also desired. If, on the other hand, the requirements stipulate that the room temperature must be kept under a certain upper limit, but this can be exceeded for a few hours, say 80 hours per year, then the HVAC plant required will be considerably simpler and less costly.

The design engineers, especially those engaged in the building services, but also the structural engineers, should, before the planning work is begun, explain any particular consequences of the requirements to the client and show what effects any simplifications of them would have. The client can then decide whether any of the requirements can be changed. However, the client must always be made aware of the consequences at an early stage and accept them or make the necessary changes. If the consequences cannot be accepted and the requirements cannot be changed, then the building may have to be redesigned.

The requirements stipulated by the client must always be taken into account when choosing HVAC solutions. Sometimes design engineers try to introduce special solutions that cannot meet the client's requirements and suggest that some of the requirements should be changed or deleted, so that the solution can be used. This is ethically and technically inexcusable. The role of the design engineer is to design the best building possible, from all points of view and one that fulfils the client's requirements within the given economical limits. The design engineer's role is not to implement special technical solutions by convincing the client to lower his standards or delete functional requirements.

Consequence analysis at the initial stage of the planning process

An architect is normally engaged at the start of the planning process for a new building. To avoid costly design solutions, the architect, before designing the building in any detail, must have an ongoing dialogue with building engineers and building services engineers. In practice, the architect often finalizes the design without giving sufficient consideration to the consequences that it will have, for example, on energy demands, thermal comfort and the scope of the HVAC installations. The client should always demand that the architect take into consideration such consequences from the very start.

BV2arch [www.belok.se] is a simulation program that has been developed for architects so that they can quickly and easily see how different design features in a building, such as facades, windows, sun shades etc, affect its heating and cooling needs. Such an analysis, at the beginning of an architect's initial work, increases the probability that the basic design of the building will have a good chance of providing an overall cost-effective and energy-efficient solution. When the client commissions an architect, he should demand that such an analysis be carried out. This requirement should also be included in design competitions and taken into account when judging the entries.

A normal, basic requirement is that a building must have an efficient layout and spatial solution. From the initial project planning stage onwards, the architect should be able to quantify key data for the planned building. This can be done using ratios such as sales floor area/gross floor area, gross floor area/work station, gross volume/work station or similar. The ratios to be used, and the limits within which they must lie, should be included in the client's requirements specification.

A structural engineer is normally consulted if the architect is unsure about the structural design. The engineer can often indicate possible structural solutions and how they will affect costs. This also applies to the structural design of the building envelope. An experienced structural engineer normally knows how a functional building envelope should be designed and how it should not be designed. Unfortunately, and far too often, shortcomings occur during the actual construction process, causing problems in the finished building. Know-how concerning the design and construction of building envelopes, so that they will function properly during the lifetime of the building, is available and it is imperative that we make use of this competence.

When it comes to a building's climate and energy functions a consequence analysis can be more difficult to carry out, especially in non-residential buildings. The following discussion is therefore primarily applicable to the design of non-residential buildings. As previously mentioned, the design engineers, and especially the building services engineers, are responsible for carrying out the analysis. This must be made clear in the conditions for the planning assignment.

The intended design of a building and the loads due to its usage determine the function and capacity of the building services, so that the usage requirements can be met. As soon as the basic plan of the building and a description of its use, together with the architect's design sketches, general floor layouts and facade designs have been presented, a preliminary system solution for the HVAC plant can be proposed. It is then possible to examine the consequences of the building design and the stipulated requirements with respect to the HVAC system, the indoor climate and the energy demands.

On the basis of the climate demands and the proposed architectural solution, it is possible, with the help of simulation programs, to calculate preliminary design air flows, ratings for the heating and cooling plant, and power demands so that the climate requirements can be met. From these design figures it is relatively simple to estimate the cost of each building service based on experience data ($€/kW_{cooling}$, $€/(m^3/s)_{air}$, $€/kW_{el}$). At the same time, the energy demands (kWh_{heat}/year, $kWh_{cooling}$/year, kW_{el}/year), can be calculated and their corresponding energy costs.

Numerous simulation programs are available for calculations like these. Nonetheless, it is important to stress that when dealing with non-residential buildings, only programs that have been designed for this specific purpose must be used. There are numerous programs for residential buildings and especially detached houses. However, these cannot normally be used for simulation of non-residential buildings, as they mainly deal with heat losses from a building and not with the systems that are used to remove the heat surplus. In non-residential buildings, the latter often determines the cost of the plant and the energy requirements.

It is important to emphasize that the person carrying out the calculations must be reasonably familiar with how the simulation program works and how the choices of initial conditions affect the results. There must also be a genuine awareness of how a building and the proposed building services work in practice. A simulation program must not be used as a 'black box'.

Once the basic details about a building and how it is going to be used have been fed into a simulation program, it is normally quite simple to change window sizes, type of sun shading, lighting levels, lighting system settings etc. The simulation should therefore include sensitivity analysis functions, at least regarding variations of window sizes, sun shading and lighting levels. This will provide important data when discussing how to improve the building with regard to climate and energy use.

Summing up, a consequence analysis must include the following points:

- An assessment of the design of the HVAC system and its required capacity.
- An assessment of space needs and costs. These are determined primarily by the design heating and cooling loads, and the design air flows. Based on these, the space required can be calculated using experience data. Costs can be estimated by using standard rates, such as €/kW_{heat}, €/$kW_{cooling}$, €/$(m^3/s)_{air}$, etc. Current rates and space requirements should be second nature to the building services design engineer.
- A sensitivity analysis with regard to the design of the building, to assess how this affects the design loads and, consequently, the plant costs and space requirements for the HVAC system as well as the total energy requirements for the building.
- Information to the client and project engineers, i.e. the architect and the structural engineer, about needs regarding space and costs, so that the stipulated requirements for the proposed building solution can be fulfilled.
- The information must be clear and presented in such a way that all parties concerned are fully aware of all consequences.

In order to proceed, the client and all the engineers engaged in the project must agree to accept the consequences of the stipulated requirements and building design, and thereby also accept the consequences regarding space and costs for the HVAC systems. If agreement cannot be reached, the design of the building must be changed and/or the usage requirements considered again and, if necessary, changed.

Only the client can decide to change the requirements and any changes made are the client's sole responsibility. However, when the consequences of the new requirements are made clear, the client might want to reassess any excessive requirements and adjust them accordingly.

Changes in the design of a building are basically a question for the architect. If a consequence analysis shows that the intended design will entail cost, energy or space demands that will be unacceptable to the client, then the architect must be instructed to change the design.

It is very important that all consequences are brought to light at an early stage, before the labour-intensive detail design work is commenced.

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15. AIR QUALITY

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Air quality is a term used to describe the degree of cleanliness of a particular volume of air and how pollutants in the air can affect both people and matter. Clean outdoor air is comprised of nitrogen (78%), oxygen (21%), argon (0.9%), carbon dioxide (0.04%) and low concentrations of a number of other gases, as well as water vapour. Although carbon dioxide is a natural ingredient in clean outdoor air, human combustion of fossil fuels is clearly causing a steady rise in its concentration. Measurements carried out at the end of the 1950s showed that the background level then was lower than 320 ppm [Keeling and Whorf, 2005]. Today, the concentration has risen to over 380 ppm. The unit ppm denotes parts per million, i.e. millionth parts of a given volume. A carbon dioxide concentration of 400 ppm means that there are 400 litres of carbon dioxide dispersed in one million litres, or 1000 cubic metres, of air.

AIR QUALITY AND AIR PURITY

Air always contains a number of other substances than those shown above, irrespective of whether it is indoor air or outdoor air. These substances originate from exhaust gases from vehicles, industrial emissions, fires and volcanic eruptions, gases and particles from people and animals, indoor processes, etc.

It is far from fully understood how and to what extent these substances, individually or collectively, affect the quality of the air. What we do know is that pollutants in general can significantly affect health and comfort. Some substances are allergens, others irritate mucous membranes and some of them affect us because of their odours, both pleasant and unpleasant.

Many of the substances that are handled industrially have relatively well known health effects and their allowable presence is regulated by industrial hygiene limits, quite often expressed as ppm levels. In non-industrial environments, a large number of substances normally occur in concentrations that are far smaller. Here, relevant concentrations are at ppb levels, parts per billion, i.e. billionth parts of a given volume. 1000 ppb is therefore the same as 1 ppm.

The health effects of a limited number of pollutants, present in the air in houses, schools and other public buildings, are regarded as being sufficiently well known to be covered by either official restrictions or target levels that have been stipulated by other, unofficial, bodies. These substances include radon, carbon monoxide, formaldehyde, nitrogen dioxide and ozone. The health risks of most substances, however, are not so well known and there are no clear limit levels.

The concentration of carbon dioxide is often used as an indicator for pollutants emanating from people. In ventilated rooms, however, it is not the carbon dioxide itself that causes the air quality to be regarded as poor. This is due to other pollutants, often unpleasant gases and particulate substances. Nonetheless, carbon dioxide is used as an indicator for these substances, quite simply because its concentration is easy and cheap to measure: if the concentration of carbon dioxide is high, the concentrations of other pollutants created by people and their activities will also be high. A common target level for carbon dioxide is 1000 ppm. Concentrations above this will often mean that the air flow, in relation to the number of occupants, is too low. As seen in Figure 1, a concentration of 1000 ppm corresponds to a supply air flow of 10 l/s per person. Carbon dioxide can, therefore, be used as a tracer gas to determine whether supply air flows are sufficiently large.

It is relatively simple to measure the concentration of carbon dioxide and it is quite easy to interpret the results and compare them to stipulated target levels. However, most other air pollutants require elaborate and expensive measuring techniques. Even if it is possible to measure concentrations of volatile organic compounds, VOCs, emitted from building materials and furnishings, or concentrations of airborne dust, problems still arise when the measurements are to be interpreted, especially as many reference levels are poorly defined or unknown, i.e. there are seldom unequivocal target levels that define what good air quality actually is. This is due to the fact, mentioned previously, that the health effects of the majority of these substances are, to a large extent, unknown.

As mentioned initially, target levels for some substances have been

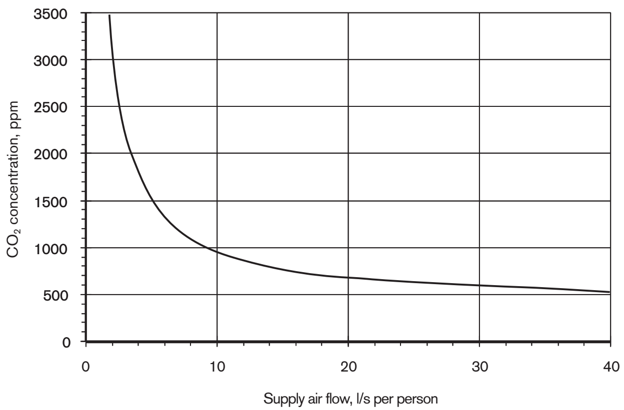

Figure 1. The relationship between the concentration of carbon dioxide (CO_2) in a room and the supply air flow. Every occupant is assumed to exhale 18 litres of CO_2 per hour and the concentration of the gas in the outdoor air is assumed to be 400 ppm.

published, for example, by the World Health Organization [WHO, 2000], and other international organizations, such as the International Society for Indoor Air Quality and Health [ISIAQ-CIB, 2004], see Table 1. In many countries, guidelines issued by HVAC industry organizations refer to more or less universally accepted recommended values [ANSI/ASHRAE, 2007; FiSIAQ, 2000; VVS-Tekniska Föreningen, 2006].

Table 1. Examples of limit values for the highest acceptable concentrations of a number of pollutants in indoor air, [VVS Tekniska Föreningen, 2006].

Pollutant	Symbol/unit	Highest concentration	Information source
Radon	Rn	100 Bq/m^3	FiSIAQ (2001)
Carbon monoxide	CO	2 mg/m^3	FiSIAQ (2001)
Ozone	O_3	50 µg/m^3	FiSIAQ (2001)
Nitrogen dioxide	NO_2	40 µg/m^3	ISIAQ-CIB (2004), WHO (2000)
Formaldehyde	HCHO	50 µg/m^3	FiSIAQ (2001)
Airborne particles <10 µm	PM10	40 µg/m^3	FiSIAQ (2001)
Airborne particles <2,5 µm	PM2.5	15 µg/m^3	FiSIAQ (2001), ISIAQ-CIB (2004)

A number of important air pollutants, their sources and their effects on health and well-being are summarized in the following sub-sections.

Radon

Rock containing high concentrations of uranium is the principal source of naturally occurring radon. Concentrations of radon in the ground vary substantially from one place to another. In some locations they might cause indoor contamination levels to become even higher. These levels are greatly dependent on the design of the building's foundations and their degree of airtightness. In some areas, the water in drilled wells can be a significant source of radon. Building materials, made using uranium-contaminated substances, and especially a type of concrete known in Sweden as blue concrete, can also contribute to high radon levels indoors.

When a radioactive element decays radioactive radiation is emitted and new elements are formed. These newly formed elements then start to decay themselves and more radioactive radiation is emitted. The decay chain for uranium finally ends with lead, a stable and non-radioactive element. During this decay chain, radon is formed. This is an inert gas which, when it starts to decay, creates so-called radon daughters. The majority of these radon daughters then attach themselves to small airborne particles. As particles tend to lodge in our respiratory organs, while gases do not, the radon daughters are regarded as more dangerous to health than radon gas. Radon can be measured using different methods, though it is more complicated to measure the concentration of radon daughters than of radon itself.

Radon and radon daughters have a health effect as they increase the risk of contracting lung cancer. It has been shown that cancer risks among people subject to radon increase dramatically if they are also subject to tobacco smoke. Limits for highest allowable levels are normally given for concentrations of radon gas, not radon daughters. In most countries, official limit levels are around 200 Bq/m^3 (1 Bq – Bequerel – is equivalent to a decay rate of one radioactive atom per second).

Carbon monoxide

Carbon monoxide is the product of incomplete combustion, for example, in petrol or diesel engines. The gas is also formed by smoking tobacco. Consequently, the largest source of carbon monoxide indoors is often created by traffic outdoors. The concentration of carbon monoxide in the outdoor air and, thereby, in the supply air is usually much lower than official health limit levels. In urban areas in Sweden, for example, concentrations can become as high as 1 ppm during periods when the quality of the outdoor air is deemed poor, especially during temperature inversions. Normally, concentrations are much lower than 0.5 ppm. However, the concentration of carbon monoxide in the outdoor air will vary with the pollution load, for example, close to an air intake on the side of a building it will vary with the intensity of the traffic on nearby roads. Carbon monoxide is relatively simple and inexpensive to measure and can be used as a general indicator for traffic-related air pollution.

Ozone

Ozone in the air close to the ground is formed when volatile organic compounds in vehicle exhaust gases are subject to strong sunlight. The concentration of ozone in the outdoor air will, therefore, vary during the year, with periods of high concentration when there is a lot of sunshine. Laser printers and copiers can also generate ozone. Air purifiers that rely on the ionisation of the air can also create ozone and this is not beneficial to the quality of the air. Consequently, the indoor environment can be affected both by ozone that is introduced from outdoors and by ozone that is formed indoors.

Ozone is a gas that readily reacts chemically with other substances and can cause irritation of the mucous membranes. A proposed limit level for the concentration of ozone in indoor air is given in Table 1. When ozone reacts chemically new substances can be formed, for example, nitrogen dioxide.

Nitrogen dioxide

Nitrogen dioxide is formed by a chemical reaction between nitrogen monoxide and ozone. This is unfortunate, as nitrogen dioxide has extremely negative health effects, while nitrogen monoxide is practically harmless. Like ozone, nitrogen dioxide affects our respiratory system and mucous membranes. Nitrogen dioxide can, for example, cause problems for asthmatics or people who suffer from pollen allergies. Nitrogen dioxide is one of those pollutants whose concentrations often reach or exceed the stipulated limit level for good air quality. A proposed limit level for the concentration of nitrogen dioxide in indoor air is given in Table 1.

Volatile organic compounds

Volatile organic compounds, or VOCs, are chemical substances in the form of thinners and solvents. Benzene, toluene and xylene are typical traffic-related volatile organic compounds. Toluene is one of a number of types of solvents that can be found in building materials and other products that are used indoors. Other, often unpleasant, VOCs can be formed by chemical reactions when building materials are subject to moisture. Microorganisms also create VOCs, often with their own characteristic smells, due to their own metabolism.

A great deal of research has been carried out since the 1980s to study the sources of VOCs and their health effects. It has been established that a large number of these substances have different forms of negative health effects and effects on well-being, but in a great number of cases there are no criteria for defining limit or target levels. This is one of the reasons why it is seldom possible to use measured levels of VOCs to draw reliable conclusions when investigating indoor environments. Thanks to research in this field, most building materials now contain and emit far smaller amounts of VOCs than 10 or 15 years ago. Nonetheless, building materials should always be handled correctly and kept away from moisture. As a result, the risks associated with this group of air pollutants have been reduced over the years.

Airborne particles

There are numerous sources of airborne particles both indoors and outdoors. The concentration of particles in a building's air intake is determined partly by sources close to the building, for example, road traffic, and partly by particles transported through the air from distant sources. The particle concentration in an air intake can vary rapidly, for example, if there is a change in wind direction. The concentration of particles in the outdoor air is generally higher in large towns than in small towns and country environments. It is quite easy to remove particles from the ventilation air before it is introduced into a building and this is dealt with in Chapter 20, which discusses air filters and air filtration.

Other examples of air pollutants that are particulate, or are bound to other airborne particles, are pollens and their allergens, allergens from domestic pets, bacteria and mould spores.

Among the most prolific sources of airborne particles in indoor air are the occupants themselves: the more active the occupants are, the larger the number of skin flakes and textile fibres released into the air.

A common measure of the particle concentration in air is the mass of all the particles up to 10 μm in size. 1 μm, or 1 micrometre, is equal to

1 thousandth of a millimetre. The measure is called the PM10 value and is expressed using the unit µg/m³. PM stands for 'particulate matter'. Another common measure is PM2.5, which includes all particles up to 2.5 µm in size. This measure is regarded as more relevant from a health point of view, as it reflects the occurrence of much smaller particles, which can more easily reach deeper down into our respiratory system than larger particles. Limit levels, with respect to good air quality, are available for airborne particles. See Table 1 for proposed limits expressed both as PM2.5 and PM10 levels.

In recent years, many researchers have turned their attention to even smaller particles, so-called ultrafine particles or nanoparticles. These particles are, by definition, less than a ten-thousandth of a millimetre, i.e. one tenth of a micrometre or 0.1 µm, in size. The lower limit lies somewhere around a few millionths of a millimetre, that is, around a few nanometres – the size of a molecule. These particles are so small that they are thought to be able to penetrate lung tissue and cause ill-health in other organs, such as the heart [Nemmar A. et al., 2002]. As yet, there are no limit values for concentrations of nanoparticles.

Carbon dioxide

Carbon dioxide is a product of our metabolism and is emitted from our bodies in exhaled air. As mentioned previously, carbon dioxide itself is not dangerous to health in the concentrations that occur in ventilated rooms. It is, however, a good indicator of other substances, for example, unpleasant gases and particles emitted due to human activity. A person sitting down will exhale 18 to 20 litres of carbon dioxide per hour and during moderate physical activity (at an activity level corresponding to about 3 Met, see Chapter 10/Thermal climate) around 50 litres per hour will be exhaled. At very high levels of physical activity, around 10 Met, a gymnast, for example, could exhale up to 170 litres of carbon dioxide per hour.

HYGIENIC AIR FLOWS

Most of the guidelines and standards relating to air quality assume that it can be guaranteed by limiting the indoor production of pollutants and by ventilating with suitable air flows. Since the 1990s, different quality systems have evolved, aimed at limiting emissions from building materials. Examples of such systems can be found in Finland [RTS, 2007], in Denmark (Danish Indoor Climate Labelling) and in Sweden (Swedish Flooring Trade Association). These systems require material manufacturers to provide information about the effects that their products will have on the quality of the indoor air. Information about an increasing number of building materials is now available and data should be consulted when choosing and purchasing materials.

There is a general consensus regarding how large air flows should be with respect hygiene when established practices, now current in numerous countries, are compared. Typical limit levels are 7 to 10 l/s per person. Discussions among researchers, however, indicate a possible future need of increased flows, for reasons of hygiene, of up to 20 or 30 l/s per person.

The concentration of pollutants in indoor air depends on the concentration of pollutants in the supply air, the generation of pollutants in the room in question and the supply air flow rate to the room. The following equation can be used for steady state conditions:

$$C_{room} = C_{sup} + \frac{G}{\dot{V}}$$

where:

C_{room} is the concentration of pollutants in the room air in µg/m³
C_{sup} is the concentration of pollutants in the supply air in µg/m³
G is the rate of generation of pollutants in the room in µg/h
$\dot{V}$ is the supply air flow rate in m³/h

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16. AIR CHANGE AND AIR FLOW

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The term air change refers to the process whereby the air in a ventilated room is replaced by outdoor air. This is also what we mean by the term ventilation, which is therefore synonymous with air change. If the supplied replacement air does not come from outdoors but, instead, from the same room from which it was removed, this is called *circulated air*. If the air introduced into a room comes partly from another room, this is known as *recirculated* or *return air*.

The size of the air change, i.e. the size of the supply air flow, can be standardized in relation to the net volume of a room, i.e. excluding the volume of the furnishings, and expressed as a *specific air flow*, often called the air change rate. The specific air flow is defined as the number of net room volumes of air that are ventilated per unit of time (air changes per hour). The time taken to ventilate a room with a volume corresponding to the net room volume is the time it takes for one air change. This time is often called the time constant of the ventilation system.

The relationships between some of the different ways in which the size of an air flow can be expressed are shown in Figure 1. For example, an air flow of 45 l/s into a room with a floor area of 20 m² corresponds to 2.25 l/s per m² floor area. If the height of the room is 3 m, the air change rate will be 2.7 room volumes per hour (i.e. 2.7 air changes per hour).

AIR FLOW IN A ROOM

The flow of the air through a room, i.e. the path taken by the air, depends on the types of supply air terminal devices and their locations, the locations of the extract air terminal devices, the temperature of the supply air in relation to that of the room air and the activities in the room. The supply air temperature often has to be lower than the room air temperature to create the intended air flow, i.e. the full ventilation of the whole of the occupied zone.

It is also assumed that there are internal sources of heat in the room, which, in turn, means that the air flow will only be correct when all the apparatuses in the room are in use and/or when the room is occupied. Even if measurements in a room that is not in use indicate that the air flow is not sufficient, the room can still have an acceptable air flow when it is in use. If air flow measurements are required, they must be carried out under conditions that are similar to the normal conditions in the room, i.e. carried out when the room is actually in use or in conditions when the heat loads are simulated, for example, by using light bulbs.

If some of the supply air is not directed through the occupied zone in a room, and is more or less directly removed with the extract air, this is called *short-circuiting*. A consequence of short-circuiting is that some parts of a room will be ventilated by an air flow that is smaller than the supply air flow. In turn, this means that some parts of the occupied zone will have higher degrees of contamination than others. These parts are called *stagnation zones*. If these conditions exist when the room is used as intended, the ventilation will be compromised and the air flow process will not be able to provide complete mixing or displacement of the room air.

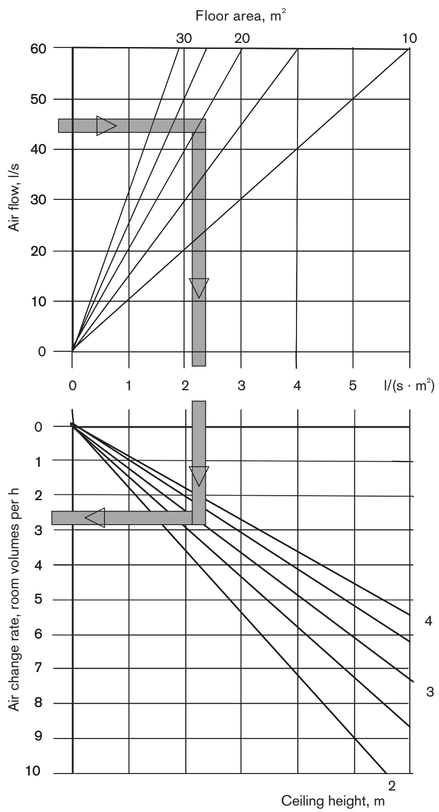


FIGURE 1. The relationships between some of the units used for measuring air flows.

In rooms that are not very large or that have high ceilings, there are seldom any practical problems with unevenly distributed air flows.

Stagnation zones and short-circuiting effects primarily occur when both the supply air and extract air terminal devices are located at ceiling level, where air speeds are low, or if the temperature of the supply air is higher than that of the room air. This latter case, however, is not very common, at least in non-residential buildings, even if heating functions are sometimes integrated into chilled beams and so-called comfort units. In residential buildings, airborne heating systems are sometimes installed and the air is supplied at an over temperature. Here, it would be advisable to carry out measurements to check the air flows.

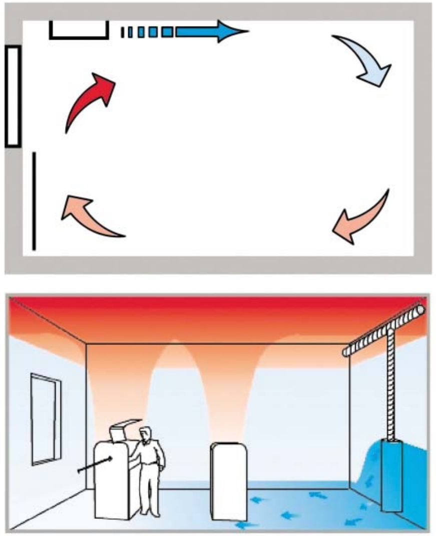
MIXING VENTILATION

Mixing ventilation systems are characterized by the supply air being introduced into a room at high speed. This means that a large proportion of the room air is set in motion by the supply air jet and subsequently mixed with it. The principle for mixing ventilation is shown in Figure 2. The air jet should be able to reach across the whole room, i.e. must have a sufficient throw. Air movements are created in the room, resulting in a mixing effect, and airborne contaminants become evenly distributed throughout the whole room. Even the temperature distribution becomes relatively even. The supply air is normally introduced at ceiling level, so that the occupants are not subjected to draughts caused by the high air speeds. The supply air is normally colder than the room air, to avoid temperature stratification with a warm cushion of air at ceiling level. If there are large cooling requirements, these can be met by letting the supply air have a temperature of 8 to 10 °C. On condition that the supply air terminal devices are suitably designed and located, air as cold as this can be introduced without causing problems with draughts. If the room is to be heated by the ventilation air, the supply air must be warmer than the room air. In this case, it is especially important that the speed of the air jet is high.

DISPLACEMENT VENTILATION

Displacement ventilation systems are characterized by cool supply air being introduced into a room at low speed. In this case, the thermal forces will dominate over the dynamic forces, i.e. this is a form of *thermally controlled ventilation*. The air flows in the room are determined by the difference in density between the supply air and the room air and by the locations of the supply and extract air terminal devices. If the supply air is introduced at floor level, so that there is hardly any mixing of the supply air with the room air, this is called *displacement ventilation*. In this case, the relatively cool supply air spreads across the floor, is heated by internal heat sources and subsequently rises towards the ceiling. The extract air is removed at ceiling level. The principle for displacement ventilation is shown in Figure 2. Both a temperature gradient and a contaminant concentration gradient will exist in the room, with a higher temperature and a higher concentration of contaminants at ceiling level than at floor level. If the displacement ventilation functions correctly, the concentration of contaminants in the occupied zone will be lower than the average level in the room, which results in a high value of the air change efficiency, discussed in more detail below. To avoid draught problems, the supply air must not be allowed to be more than a few degrees colder than the room air. On the other hand, there is always a great risk of experiencing draughts close to the terminal devices used for displacement ventilation, which limits the extent of the occupied zone, i.e. the useful part of the room.

There is also a form of thermally controlled ventilation in which the room air is mixed with the supply air inside the terminal device or close to it. As in normal displacement ventilation, cool supply air is introduced at low speed. The extract air is removed at ceiling level but, unlike in dis-



MIXING VENTILATION
The room air is mixed by being drawn into a high-speed air jet. The air jet reaches all the way across the room.

DISPLACEMENT VENTILATION
Cooled supply air is introduced at floor level. The cooled air spreads across the floor, gains heat from apparatuses and people, and rises towards the ceiling.

FIGURE 2. Air flow principles: mixing and displacement ventilation.

placement ventilation, the supply air terminal devices can also be located close to the ceiling. This method is called *equalizing ventilation*, as an equal temperature distribution is sought, primarily in the occupied area.

PISTON FLOW

Piston flow ventilation is a method that is only used where there are very high demands on air quality, in so-called clean rooms. The method entails distributing the supply air evenly, for example, over the ceiling area and in such a way that there is only one direction of flow, namely, in this case, downwards. The air is said to pass through the room as though it were pushed by a piston. In this case, the extract air terminals are located at floor level. If this method is to function properly, relatively high air speeds, up to 0.35 to 0.40 m/s, will be required to ensure stable piston flow through the room. Because of these high flow speeds, piston flow is never used in connection with comfort ventilation, as this would cause draught problems.

CHECKING THE BEHAVIOUR OF THE AIR FLOWS

Demands are placed on the behaviour of air flows to ensure that the whole comfort zone is properly ventilated, i.e. that there are no short-circuiting effects. Air flows in ventilated rooms can be assessed by measuring one of the following:

- The air change efficiency
- The contaminant removal effectiveness
- The local air change index
- The local air quality index

Determining the air exchange efficiency is the most complicated, time-consuming and expensive way of verifying air flows and measurements require specialist competency. In practical applications, for example, when inspecting installation work, it should always be possible to verify air flow requirements using a reasonable amount of effort and at a reasonable cost. This is not the case if this has to be done by measuring the air change efficiency. The ventilation efficiency and the local air change index are also time-consuming to measure and complicated to verify. On the other hand, it is relatively easy to determine the local air quality index.

AIR CHANGE EFFICIENCY

This is a measure of how well the supply air is distributed throughout a room. High air change efficiency implies that there are no stagnation zones. The air change efficiency is *a mean value for a particular room*. A

high air change efficiency indicates, but does not prove, that there are no stagnation zones.

The air change efficiency is defined as the ratio of the nominal time constant of the ventilation system to twice the mean age of the room air.

The mean age of the room air can be measured using tracer gas methods, in which, for example, sulphur hexafluoride or nitrous oxide (laughing gas) are used as tracers. The methods are based on analyses of natural dilution process measurements, i.e. the process that takes place without the room air being mixed by table fans or similar. However, before measurements of the dilution process can begin, the room air must be thoroughly mixed. Measurements and analyses are then carried out according to the NT VVS 047 Nordtest method [Nordtest, 1985]. The nominal time constant is determined either from the dilution process measurements, when the room air is continuously mixed using mixer fans, or by dividing the net volume of the room by the measured supply air flow. This means that two tracer measurements might have to be carried out: one to measure the mean age of the room air and one to determine the nominal time constant.

Air change efficiency is defined as follows: its value is 1.0 in the case of piston flow, is between 0.5 and 1.0 when displacement ventilation is used and is 0.5 when there is complete mixing.

Determining air change efficiency is time-consuming and includes measurements on site and a following analysis. For example, it normally takes about 3 hours for a qualified technician to carry out measurements in a single room with a specific air flow of 0.5 room volumes per hour. The time required for analysing the measurements must then be added. According to NT VVS 047, this method is primarily suitable for mechanically ventilated buildings.

VENTILATION EFFICIENCY

Ventilation efficiency is defined as the steady state concentration of a tracer gas in the extract air divided by the average concentration of the gas in the room air. Even this measure provides an average for the whole room and a high ventilation efficiency is therefore an indication, but not proof, that there are no stagnation zones. The method is impractical as the average concentration in a room is very difficult to measure. In a handbook published by REHVA, the European HVAC organization, [Mundt et al., 2004] a method is described in which the ventilation is switched off after introducing the tracer gas into the room. The air is then mixed by using mixing fans and the concentrations measured. It is obvious that this is a difficult method to apply in many buildings, and impossible in others, and is therefore not discussed further.

LOCAL AIR CHANGE INDEX

The local air change index is defined as the nominal time constant, i.e. the time required for one change of air divided by the local average age of the room air at a specified measuring point. The method used for measurement and analysis is similar to that used for determining the air change efficiency. The local average age of the room air can be determined by using the NT VVS 019 Nordtest method [Nordtest, 1988]. The local air change index is relevant only for one particular measuring point and it might be necessary to repeat the time-consuming procedure at a number of points in the room. Determining the local air change index is, consequently, at least as time-consuming as determining the air change efficiency. The method is not discussed further.

LOCAL AIR QUALITY INDEX

This is a measure of how effectively contaminants in the air can be removed from a room. It can be based on measurements of the concentration of carbon dioxide in the supply air, in the room air and in the extract air, according to the NT VVS 114 Nordtest method [Nordtest, 1997]. In Nordtest publications, the index is called the local ventilation index, but the definition is still the same:

$$\textit{Local air quality index} = \frac{C_{ext} - C_{sup}}{C_{room} - C_{sup}}$$

where:
C_{ext} is the concentration of tracer gas in the extract air
C_{sup} is the concentration of tracer gas in the supply air
C_{room} is the concentration of tracer gas at a point in the room

The measurements are carried out when the concentrations have reached a steady state and relevant criteria are given in the Nordtest instructions. It might be necessary to carry out measurements at a number of points in the room. The number of points chosen depends on the size of the room, its layout, the location of the supply air terminal devices etc. The method is relatively simple and quick, even if measurements have to be made at a number of points.

A local ventilation index close to or above 1.0 shows that the air flow in a room is satisfactory, i.e. that there are no stagnation zones or short-circuiting effects. If there is complete mixing, the ventilation index will be 1.0 precisely. Values under 0.9 indicate short-circuiting, i.e. that a considerable amount of the supply air is being extracted directly. In a report from the European Committee for Standardization [CEN, 1988], the interval 0.9 to 1.0 is given as typical for mixing ventilation systems with cooled supply air, while a well-functioning displacement ventilation system can have values between 1.2 and 1.4.

CRITERIA FOR A SUITABLE METHOD

Buildings often contain different types of rooms and a large number of measurements would have to be carried out if air flow requirements were needed to be checked, for example, when carrying out an inspection on completing an installation contract. In situations like these, it is much more expedient if a relatively simple and quick method can be used. Furthermore, measurements might have to be carried out while work is still in progress and it should therefore be possible to choose a method that causes as few disruptions as possible. In addition, the method that is chosen should be standardized and quality assured.

Measurements should be simple and quick to carry out
Measurements of air change efficiency and contaminant removal effectiveness are very time-consuming and, therefore, expensive. In practice, only researchers and possibly a few specialist consultants can carry out measurements like these. Measurements of local air quality are relatively simple and ought to be able to fulfil the criteria for simplicity and quickness.

Measurements should be carried out using a standardized method
When requirements are stipulated regarding air flows the method to verify them must also be stated. The simplest way of doing this is to refer to a standardized method. As seen above, Nordtest instructions have been published for a number of the methods that can be used, even for determining the local air quality index.

Measurements should not interrupt ongoing activities
Measurements to determine air change efficiency require the release of a tracer gas and the room air to be mixed using separate mixing fans. Measurements also need to be carried out over a long period of time

(hours), with a sampling tube connected to each supply air terminal device. Measurements of the local air quality index can, on the other hand, be carried out with far fewer disturbances, as the measurements require shorter time and the room air does not have to be mixed. Furthermore, the natural content of carbon dioxide in the air can be used as a tracer gas, as long as the room air has a much higher concentration than the outdoor air. The method is therefore suitable for rooms that are in use.

DESIGNING FOR ACCEPTABLE AIR FLOWS

The design engineer's job is to design the ventilation system to suit the layout of the building and its intended use. Where air flows are concerned, it is a question of ensuring suitable supply air temperatures and choosing suitable supply air terminal devices, i.e. terminal devices that can be adjusted to provide the correct throws. In most cases, it will be sufficient to use the data and diagrams supplied by the terminal device manufacturers.

It is also possible to predict air movements in a room by using computerized calculations, so-called computational fluid dynamics, CFD, calculations. In special applications, for example, in very large rooms, it might be advisable to use techniques like these to ensure acceptable air flows.

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17. DEMAND-CONTROLLED VENTILATION

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INTRODUCTION

What is demand-controlled ventilation? There are a number of answers to this question but, basically, it means that air flows are controlled by the demand for fresh air and depend on the number of people occupying a room or space at a given time. Demand-controlled ventilation is normally denoted by the initials DCV.

The term DCV has been used since the end of the 1980s, when serious consideration was first given to the design of ventilation systems with the aim of improving air quality in both residential and non-residential buildings.

Previously, the term VAV, which stands for variable air volume, was used when referring to systems with variable air flows. This expression was derived from the ventilation systems originating from USA in the 1960s. VAV systems were also a form of demand-controlled ventilation, though the purpose of the system in this case was to supply air depending on the prevailing cooling needs.

Today, however, it ought to be possible to use the term DCV as a general term for systems designed to control the air flows that are required to maintain a good indoor climate.

The two main reasons why demand controlled ventilation is used today are to ensure a sufficiently large flow of fresh air and to maintain a low level of energy use.

Considering the problems encountered with air quality that were very common in the 1970s and 1980s, it is not surprising that there has been increased focus on improved air quality.

Looking back at how we have previously dealt with the subject of ventilation, it can be seen that attention was often turned to the creation of good thermal comfort, neglecting the other important function of the ventilation system, namely, the creation of good indoor air quality.

When we talk about comfort ventilation, the scope of this chapter, it must be emphasized that the primary purpose of a ventilation system is to create good air quality. A secondary, but nonetheless important, purpose is to create a good thermal climate.

One of the most important features of a demand-controlled ventilation system is that it is not necessary to transport more air through the system than is momentarily needed. This feature is important, as it allows systems offering good air quality and good indoor climate to be combined with systems that use a minimum amount of energy.

Today, in Sweden, buildings account for about 40% of all energy use and a great deal of this is used to provide heating and cooling. This is why it is extremely important that future development work in this field is aimed at implementing technologies that favour low energy usage.

In this chapter it is argued why we must focus on system design and develop it in such a way that we will be able to talk about the creation of a new ventilation technology. This is because system solutions must be created that both satisfy our needs for good comfort while reducing energy usage to the lowest possible level. An important ingredient in this new ventilation technology will be demand-controlled ventilation.

WHAT DO RESEARCHERS SAY?

The task of the ventilation industry is to create system solutions that satisfy people's needs for good air quality and good thermal comfort. It is therefore essential that the industry actively participate in research projects at university level, so that they can use new findings in their own development projects. A significant increase in investments in such research has been seen over the last ten years and a report, with a compilation of the research results from projects conducted during the period 1996–2002 in the Nordic countries, has been published [Svensson, 2003]. The report supports industry consensus that there is a large need for improvements in system design to ensure that proposed functional demands can be met in practice. It also concludes that important research projects have been carried out and that a number of general conclusions can be drawn. These include:

- The importance of being able to make continual adjustments, i.e. being able to change air flows according to prevailing needs.
- The need for great attention to be paid to the design of systems to ensure low noise levels.

- The need for increased flexibility within the systems to facilitate compliance to different activities
- The importance of designing systems with low energy needs.

Bearing these factors in mind, there ought to be an increase in the use of DCV systems in the future.

One of the most recent research reports on DCV systems was presented as a licentiate dissertation [Maripuu, 2006] at Chalmers University of Technology, Gothenburg. The report emphasizes the importance of producing simple system solutions that can be implemented both when upgrading existing systems and when installing new ones. Great emphasis is placed on the flow patterns from supply air terminal devices being stable and independent of the supply air flow, and the risk of cold downward draughts in the occupied zone being prevented when air flows are low. The report also mentions the need for better insulation of ducting systems to reduce transmission losses.

Research aimed at improving the functions of demand-controlled ventilation has, however, been carried out for a long time. An important year in this respect was 1974, when the IEA, International Energy Agency, was formed within the OECD. The purpose of the IEA was to implement a new international energy research programme with emphasis on developing methods to reduce the use of energy. An important part of this work was to initiate research and development in the field of energy use in buildings.

A number of different research teams (Annex Teams) were established and one of these was Annex 18, working on demand-controlled ventilation systems. Sweden was designated the so-called operating agent for this team, comprising researchers from Belgium, Canada, Denmark, Finland, Germany, the Netherlands, Norway and Switzerland. A large number of the team's reports have been published and the team's work has been summarized in a report [Månsson et al., 1997].

Although the report confirms that DCV systems offer clear energy savings, no general figures for savings are given, as these are greatly dependent on the type and location of the plant, though savings of up to 60% are mentioned.

One report [ASHRAE Journal, 2003] gives an account of the situation in USA. Here, the average repayment time for DCV systems is given as 2 to 3 years, which makes the system an attractive alternative for many applications. Somewhat better results were obtained in a research project in California [PIER Buildings Program], in which repayment times of 2 years or less were achieved.

Another report [Svensson, 2003] reveals that, according to IEA Annex 35, the requirements specification must be adapted to the ventilation system that is chosen. The Annex maintains that expectations placed on a project cannot be the same in simple cases as in more complicated cases. It is also maintains that the degrees of fulfilment of the aims, for energy use and comfort, cannot be the same.

It ought to be quite obvious that a requirements specification must not be dictated by the choice of ventilation system. The only justifiable approach is to first formulate the requirements and then choose the system solution that can best fulfil them.

In a number of the research projects that were analysed, the ventilation systems were designed for air flows of 5 l/s per person instead of the normally prescribed 8 l/s. Unfortunately, this seems to be part of a deliberate strategy to achieve a minimum possible use of energy.

The trends today, towards design for the lowest possible air flows to achieve low energy use, can be seen as a new opportunity for DCV systems to gain a larger market share. With DCV systems it is possible to combine the desire for good air quality with requirements for low energy use. Using DCV systems, there is no need to compromise on these wishes.

WHAT ARE THE REQUIREMENTS?

When planning a ventilation system it is important that a sufficiently good indoor climate can be achieved in a controllable way for many years to come. There are extensive rules and regulations that the design engineers must take into account.

On the other hand, there are a number of requirements/desires that are not so easily understood. Among other things, there are often requirements that are implied or taken for granted.

Functional safety is also important and this can be improved by designing the system so that shortcomings, inherent in any system, are also accepted and dealt with by the system.

It is therefore quite reasonable to use the term wide-tolerance ventilation systems to denote systems that accept and deal with a number of common operational disturbances or obstacles.

Some of the requirements/desires that should be expressed by the different parties engaged in a building project, so that they will be satisfied with the indoor climate system, can be summarized as follows:

- The users would like a system that does not cause draughts or noise and provides acceptable air quality.
- The proprietor would like low costs and a guarantee that the system will function well.
- The operating and maintenance staff would like to see a simple design configuration.
- Both the contractor and the operating and maintenance staff would like the commissioning work to be as simple as possible. Sometimes the plant does not even fulfil elementary requirements so that it can be correctly commissioned. It is not surprising, however, that problems occur when setting air flows, as working within the small allowable tolerances is an extremely difficult task for the commissioning engineers. Clients should therefore demand better and more commissioning-friendly tolerances when procuring system design.
- The property manager would like to see systems with low energy demands and functional durability, as well as flexibility when premises have to be adapted to new tenants. Flexibility is a key concept in the building industry and this means:
  – Being able to control air flows according to demand and change levels of demand during operation.
  – Being able to easily extend or add to the plant.
  – Being able to change the distribution patterns in individual rooms to suit how they are used.
  – The users being able, to a certain extent, to adjust the air flows and room temperatures, [Hult, 2002].

If these desires and requirements are to be met to a reasonable extent, the focus of the design engineers' attention must be on the users and their needs for sufficiently good indoor climate. This means that the design engineers, besides planning an indoor climate without noise or draughts, must also focus on a simple and reliable system design to achieve long-term functional stability.

Other requirements that have been expressed, in order to reach these goals, include, in the first stage of the building process, i.e. before the shape, size and structural principles have been finalized, being able to assess the consequences of the design and properties of the indoor climate system. This makes it possible to stipulate realistic demands for the indoor climate and to make sure that sufficient space is reserved for the plant room etc.

The tendering procedure for the indoor climate system is also important, if satisfactory results are to be obtained. The areas of responsibility for the system must be made quite clear from the very start, so that it is not apportioned to too many hands. The HVAC consultant must take on overall responsibility for the plant functioning according to the client's requirements.

The need for good air quality

Requirements regarding lowest allowable air flows are normally stipulated in official regulations. However, it should be noted that more recent research [Wargocki et al., 2000] and [Indoor Air, 2000] has shown that the number of days lost through sickness can be reduced, if air flows are increased above minimum levels.

A great advantage of DCV systems is that they can be designed for the highest required air flows without having to compromise on total energy use. The highest flows are normally only required for short periods of time after which they are reduced as the demands on the system are reduced, promoting, in turn, lower energy use.

In the Nordic countries, there are normally no problems when supplying air at the lowest rates. In countries where the use of return air is allowed, problems have been observed when trying to supply sufficiently high air flows during periods of low demand. This is primarily a problem in USA, Canada and the UK.

The indicator that is most often used as the determining parameter in DCV plant is the concentration of carbon dioxide, CO_2. Concentrations of individual or total volatile organic compounds, VOCs or TVOCs, can also be used but the use of CO_2 is recommended, as this is a reliable indicator in premises where people are the primary source of pollutants.

The need for suitable flow patterns to avoid draughts

If a DCV system is chosen, air flows will vary according to need. This means that great demands must be placed on the properties of the supply air terminal devices. To avoid problems with draughts, it is important to maintain suitable distribution patterns that are independent of whether the flows are large or small. It is therefore advantageous if the same type of distribution pattern can be achieved, irrespective of air flow.

A common problem encountered with earlier VAV systems was that the supply air, at low flow rates and at under temperatures, would flow down into the occupied zone and cause cold downdraughts. This downflow must be avoided by creating sufficiently high supply discharge speeds from the air terminal devices.

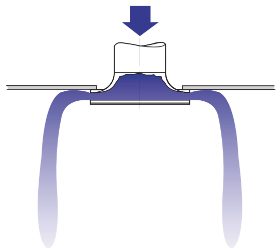

FIGURE 1. If supply air flow speeds are too low in traditional air terminal devices, correct distribution patterns cannot be maintained. The risk of draught problems is obvious when air is supplied slowly at under temperatures.

If air is supplied at over temperatures in a DCV system, it is also important to maintain a suitable distribution pattern. A problem that can occur at low air flow rates is that stagnant zones can be created, if sufficiently high air speeds cannot be maintained at the outflow from the supply air terminal device.

Minimizing noise levels

High noise levels over long periods of time have been one of the most common problems caused by ventilation plant. They are completely unnecessary and steps must be taken to avoid them. It is, therefore, of the greatest importance that design engineers and system solutions address this question seriously.

One cause of high noise levels is pressure drops across some of the system components, such as supply air and extract air terminal devices. There are, therefore, strong reasons why systems should be designed to create the lowest possible pressure rises, so that the risk of noise problems can be eliminated. In DCV systems, in which air flows vary, pressure control devices are required in certain parts of the system to limit pressure build-up. It is worth mentioning that early VAV systems from USA did not sell well in Europe, among other things, because of the high pressure drops and, consequently, high noise levels.

Flexible systems – systems that are easy to adjust or add to

One of the reasons why many ventilation installations in Sweden have not been approved, after compulsory ventilation inspections, was their limited flexibility [Engdahl, 2002]. Inspections often showed that it would not be possible to upgrade the plant, if the use of a building was changed.

Flexibility, in this respect, means being able to make minor changes in the sizes of flows to individual rooms as well as being able to change the room layout, and size and shape of the rooms, without making extensive changes to the ventilation system.

A simple method for achieving the desired flexibility in a ventilation or indoor climate system and, at the same time, for fulfilling the demands and requirements for low energy use and low noise levels, can be found in [Engdahl, 2002] and [Maripuu, 2006] for mechanical ventilation systems.

COMPONENTS IN VAV/DCV SYSTEMS

The components, besides the actual fan unit, that might be used in connection with demand-controlled systems include:

1. Variable flow units/supply air terminal devices
   a. Pressure-dependent
      This means that the air flow through the unit varies with the static pressure before the unit. Special care must be taken to check that flow variations are kept within the stipulated limits.
   b. Pressure-independent
      This means that the air flow through the unit is constant and independent of the pressure variation before the unit. Consequently, the unit includes a device for measuring the air flow, so that the damper can be adjusted to maintain the desired air flow.
      It should be noted that pressure-independent units could cause operational disturbances, if the flow measurements are carried out using sensors that react to static pressure differences. Experience shows that flow sensors require careful and regular service if they are to function properly.

2. Supply air terminal devices – pressure-dependent
   a. Passive
      These air terminal devices are the traditional ones in which the setting, the outlet area of the device, is independent of the air flow.

The flow can be regulated by manually adjusting a damper in the device intake or by employing a motorised damper in the ducting immediately before the device.

b. Active
These are devices in which the outflow is regulated (variable outlet area). This means that the outflow speed is independent of the outlet area, provided that the static pressure of the air before the device is kept constant. No special flow measurements are required if the pressure remains constant, as the flow is determined by the size of the opening (the outlet area) and the operating pressure.

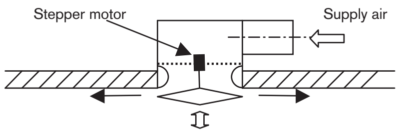

FIGURE 2. A schematic of an active pressure-dependent supply air terminal device.

3. Extract air terminal devices – pressure-dependent
   a. Passive
   These air terminal devices are the traditional ones in which the setting, the inlet area, is independent of the air flow. The flow is regulated by employing a motorised damper in the ducting immediately adjacent to the device.
   b. Active
   These are devices in which the flow is regulated at the intake (variable inlet area). The device requires a constant pressure in the branch duct so that the flow can be controlled and so that noise generation is kept to a minimum.

4. Central control units
These are programmed on site or in the factory before delivery.

5. Air quality sensors
As the purpose of a DVC system is to control the air flows in a building or room according to prevailing needs for good air quality, it is quite natural that there is some form of air quality sensor. There are a number of types of sensor that react to different types of pollutants. The most common are CO_2 sensors, i.e. sensors that react to the concentration of carbon dioxide in a room. As these CO_2 sensors are now used to a far greater extent, it has been possible to reduce their price. Over the last ten years, the price has been halved.

The advantages and disadvantages of different types of sensors are as follows:

a. CO_2 sensors have been shown to be reliable. Long-term stability has increased and measurement accuracy is normally between ± 30 ppm for measurements under 1000 ppm. If these sensors are used, the occupants should be the dominating source of pollution. Previously, sensors like these had to be recalibrated every year. Many of today's CO_2 sensors do not require recalibration during their operational lifetime (10–15 years).
b. Presence sensors are cheap and are suitable for use in office modules. Provided that a presence sensor can determine the number of people in a room, it can also be used to control the air flow according to actual demand. Otherwise, CO_2 sensors are a better choice.
c. Humidity sensors should only be used under very special circumstances. Fibre sensors are less reliable than electrical impedance sensors.
d. Mixed gas sensors can be used when the occupants are not the major source of pollution. They can be used to identify a number of gases and particles such as tobacco smoke, hydrogen, carbon monoxide, ethanol and ammonia. It is not possible to determine which gas is predominant when using a mixed gas sensor.
e. Time-controlled flows might be required in certain circumstances but this solution should not be used when people use a room on an irregular basis.
f. Sensors in which a CO_2 sensor and a temperature sensor are combined in the same unit are available on the market and can be recommended as an economical alternative.

DIFFERENT TYPES OF VAV/DCV SYSTEMS

When planning a demand-controlled system, it is necessary to include some form of pressure control in the system. This is because pressure drops must be avoided, as the flows are variable. Where these pressure-controlling units are placed in the system depends on the properties of the supply air terminal devices and the overall ambition to economize on energy.

Examples of system design:

A. Pressure-independent variable flow units installed before passive supply air terminal devices in combination with a pressure control unit in the main duct.

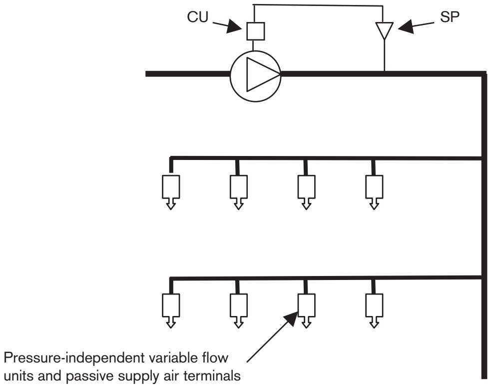

FIGURE 3. A supply air system – type A – with pressure-independent variable flow units connected to passive supply air terminal devices.
CU – Control unit for regulating fan speed using a frequency converter.
SP – Static pressure sensor

When passive supply air terminal devices are used large air flow variations are not normally permissible, if the spray nozzles are required to provide acceptable distribution patterns over the whole range of flow. However, some of the latest terminal devices with advanced nozzle design can operate satisfactorily over large ranges of flow. The devices can be set to provide acceptable distribution patterns even at low flow rates.

To minimize both noise generation and energy use, the pressure in the main duct is limited by regulating the speed of the fan.

The use of this type of demand-controlled system was previously restricted because, among other things, problems caused by noise and draughts were quite common.

B. Pressure-dependent active supply air terminal devices in combination with pressure control units in the main and branch ducts.

The pressures in the branch ducts are kept constant by a regulator installed at the beginning of every branch. The pressure sensor can be placed between the last two supply air terminals in each branch. The pressure in the branch ducts is determined by the properties of the supply air terminal devices.

A pressure sensor is installed in the main duct to limit the pressure after the fan by controlling its speed via a frequency converter. This sensor does not have to be placed immediately after the fan as shown in Figure 3. A more even pressure distribution between the branch ducts can be obtained by placing the sensor further along the main duct, see Figure 4.

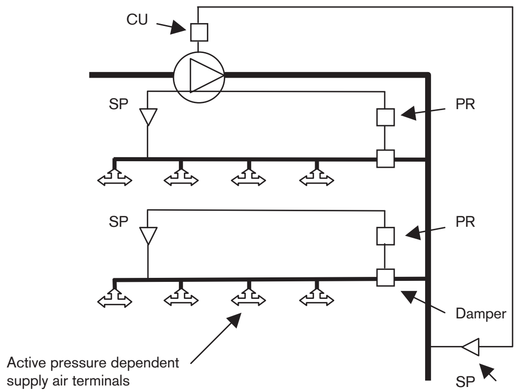

FIGURE 4. A supply air system – type B – with pressure-dependent active terminal devices, in combination with constant pressures in the branch ducts. A pressure regulator regulates the speed of the fan.
CU – Control unit for regulating fan speed using a frequency converter.
SP – Static pressure sensor
PR – Pressure regulator

In normal circumstances, a pressure of 30 to 40 Pa should be sufficient to manage the largest and smallest air flows. To obtain the best possible distribution of air to the terminal devices in each branch duct, the air speed in these ducts should be limited to 3 m/s.

Bearing in mind demands for flexibility and future adjustments, it is recommended to let the branch duct system have the same dimensions along the whole of its length.

A great advantage this type of constant pressure regulation is that passive terminal devices, designed to supply continual and constant air flows, can also be installed in this type of supply air system.

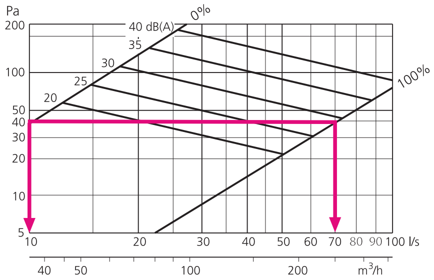

FIGURE 5. The diagram shows one of the advantages of constant pressure technology. Reduced flow, i.e. restricting flow with a constant pressure drop through a device, generates less noise.

C. Pressure-dependent active devices in combination with pressure regulation in branch ducts and fan speed control via a regulator to optimise operational data.

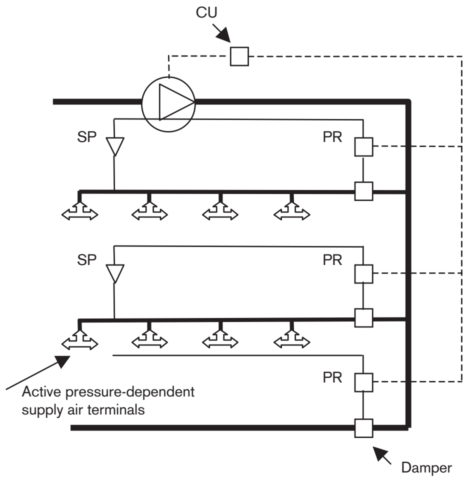

FIGURE 6. Supply air systems – type C – with pressure-dependent active terminal devices, in combination with pressure regulated branch ducts and fan speed control
CU – Control unit for regulating fan speed using a frequency converter
SP – Static pressure sensor
PR – Pressure regulator

The same type of supply air terminal devices (active and pressure-dependent) as in type B can be used. A regulator, PR, communicates with the branch duct dampers and fan units. This means that the opening angles of the branch duct dampers are monitored allowing the fan speed and, consequently, the pressure rise across the fan, to be reduced. This version is a good example of a system solution that minimizes the fan speed and pressure rise and, thereby, minimizes the use of energy and the risk of disturbing noise.

D. Pressure-independent active devices in combination with fan speed regulation via a pressure sensor in the main duct.

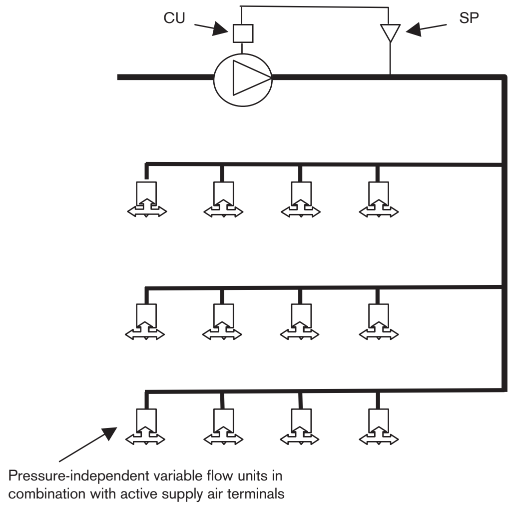

FIGURE 7. A supply air system – type D – with pressure-independent variable flow units, in combination with active devices and fan speed regulation via a pressure sensor in the main duct.
CU – Control unit for regulating fan speed using a frequency converter
SP – Static pressure sensor

In this case, the pressure regulators, PR, and the damper in type C have been replaced by active and pressure-independent supply air terminal devices. This means that a pressure sensor has to be installed in the main duct to regulate the fan speed via a frequency converter. This version is more prone to noise problems, as the pressure drops across the devices are considerably greater than in type C.

A limitation of this type of solution is that passive supply air terminal devices designed for constant flow cannot be connected to the duct system.

A positive aspect of the B, C and D systems is that a large part of the often quite difficult commissioning work can be eliminated.

Experience shows that commissioning is seldom carried out completely correctly. One reason for this is that the building might have been occupied before the commissioning work was even begun, making it difficult to carry out the work accurately. Other practical problems can arise due to measuring techniques.

Comments on the different types of systems

The technical solutions that are normally used when designing CAV systems do not allow the air flows through the supply and extract terminal devices to be varied to any great extent. Only when VAV/DCV systems are chosen is it possible to regulate air flows depending on the room temperature or concentration of CO_2, or by using presence sensors. And this naturally limits the flexibility of CAV systems, as the design air flows through the terminal devices can not be easily changed, as the properties of the devices, when passive, are limiting factors.

When flows were previously regulated in VAV systems this was done by using dampers installed immediately before the supply air terminal device (Type A). A disadvantage here, if low speed devices are not used, is that the speed of the air, on leaving the ventilation device, will be proportional to the air flow. This means that in ceiling devices, when discharging low supply air flows at an under temperature, there will be a risk of the ventilation air flowing down from the ceiling into the occupied zone, causing draught problems, see Figure 1.

If flow regulation is achieved using pressure regulators, as in B, C and D, these disadvantages can be eliminated.

By maintaining constant pressures in the branch ducts, in B and C, there will be a certain degree of flexibility with regard to the flow changes in the terminal devices. The devices can be adjusted individually without having any negative consequences on the overall balance in the system or the commissioning of the plant.

It should be pointed out that maintaining constant pressures is warranted even if the system is designed to function as a CAV system. In this case, we will have a so-called wide-tolerance ventilation system solution, i.e. one that automatically compensates for the operational disturbances or obstacles that can arise in the system.

If active devices are used, the distribution patterns will, to all intents and purposes, be the same even when air flows differ. This property/function is yet another example of a wide-tolerance ventilation system solution. Maintaining constant pressures in the branch ducts automatically compensates for the disturbances that occur in the form of varying air flows.

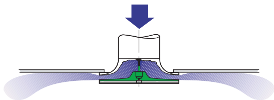

FIGURE 8A. Distribution pattern for nominal air flows.

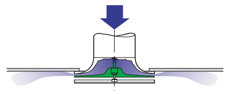

FIGURE 8B. Distribution pattern for reduced air flows, on condition that constant air pressures can be maintained in the branch ducts and that the flow is regulated at the outlet of the terminal device.

In order to differentiate between these two types of supply air terminal devices, the terms passive device and active device are used. The passive device is the traditional type in which the setting is independent of the air flow. Active devices have a flow regulator at the outlet of the device. If the properties of these devices are studied more closely, the major differences that can be seen are:

- Passive devices increase the risk of draughts when the air flow is reduced.
- Active devices reduce the risk of draughts when the air flow is reduced.

This disadvantage of passive devices, when used in flow regulated ventilation systems, is one of the reasons why traditional VAV systems, i.e. type A, have not been used to any great extent in Europe.

The supply air terminal devices used in type D systems offer an alternative solution and they also have the advantageous flow properties of the active devices used in B and C. The difference is that pressures in the branch ducts increase when flows decrease.

Systems B and D can be recommended for small ventilation systems, while system C is a better choice for large systems. The reason for these recommendations is that the pressure variations have to be limited. In large systems, pressure variations can be a problem and it is essential that operating costs are minimized, which system C can ensure.

In systems A, B, C and D described above, the extract air plant has not been included in the diagrams. It is, however, extremely important that the system design can ensure a flow balance between the supply and extract air, no matter whether the flows are large or small.

The principles used for the design of an extract air system can be the same as those for a supply air system. However, simpler extract air system designs can be accepted, for example, for handling transferred air from office modules and for central extraction from corridors. To ensure a balance of flow, both supply and extract air flows must be monitored and the extract air flows regulated according to the variations in the air flow, see Figure 9.

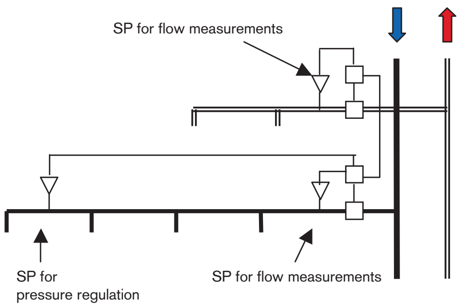

FIGURE 9. The desired balance between the supply and extract air flows is achieved by measuring the supply and extract air flows and regulating the extract air flow according to the flow variations in the supply air duct.
SP – Static pressure sensor

SUMMARY

Over the last decade, it has become obvious that the time has now come for better acceptance of system solutions based on flexibility, demand-controlled ventilation and low energy use. The new generation of ventilation systems focuses on:

- Comfort for the individual user
- Cost-effectiveness and flexibility for the client

Based on experience of different technical solutions, partly via different research projects and partly via the compulsory functionality tests, it is now possible to fulfil all the requirements that were initially described to achieve good ventilation systems. One of the first steps is to use pressure regulation devices at strategic locations in the ducting system. This will make it possible to create a flexible, demand-controlled and low-energy solution. With the help of the research and development work mentioned above, it is clear that it is now quite feasible to have:

1. Simple, pressure-dependent regulatory functions in the supply and extract air terminal devices.

2. Good control over the most commonly occurring noise problems in ventilation systems, as pressure drops can be limited.
3. More user-friendly system devices with which the users can, within certain limits, adjust air flows without disturbing the overall balance in the system.
4. A ventilation system in which it can be guaranteed that the design air flows can be maintained.
5. A ventilation system in which there will always be a balance between the supply and extract air flows, as active components in the system can ensure this balance.
6. A ventilation system in which traditional commissioning of air flows will not be necessary.
7. A reduced use of energy as:
   a) the flow balance in the system can be maintained, despite the static pressure in the branch ducts being limited,
   b) the fan can operate at its lowest possible speed to create the required air flows.
8. A ventilation system in which devices for constant air flows can be mixed with devices for variable air flows.
9. Good room climates without draughts within the whole of the operating ranges of the supply air terminal devices.
10. Flexibility, with regard to future additions to the system.

The term wide-tolerance ventilation system was, at the beginning of the 1990s, a vision – now it is a reality. There is, therefore, no exaggeration when it is said that the research carried out over the past ten years has made a breakthrough in the field of ventilation system design.

Wide-tolerance ventilation systems can, in the future, be seen as ventilation systems that have been designed in a natural way. They will, quite simply, be logically designed and the ventilation problems of yesterday will have been eliminated by careful system design.

Knowledge and quality awareness will have been allowed to prevail.

Quality awareness must embrace all aspects of a ventilation system, if the whole system is to be regarded as satisfactory and not only some of its constituent parts. All parts of a solution must work in harmony to create a functioning whole, so that the desired functional requirements can be fulfilled.

Demand-controlled ventilation is here to stay.

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18. OUTDOOR AIR INTAKES – LOCATION, DESIGN, INSPECTION AND CLEANING

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INTRODUCTION

Good indoor air quality is a must for health, comfort and well-being. Poor indoor air quality, on the other hand, can have negative effects on work performance and productivity. In order to ensure satisfactory indoor environments, we install more or less advanced ventilation and climate systems and, from a purely technical point of view, we can make numerous choices, depending on the complexity and location of the building in question. Common solutions include natural ventilation, extract air systems, balanced mechanical ventilation systems, air conditioning and different types of hybrid solutions.

There are often quite heated arguments about which alternative is best. Common to all solutions, however, is that the quality of the results will never be better than the design engineer's own level of competence and the quality of the components in the finished installation. And discussions as to whether the ventilation system should be natural or mechanical just tend to shift focus from the real challenges. The primary aim of a ventilation system must always be to provide a good indoor environment and one of its central tasks is the removal of pollutants and the regulation of the indoor temperature.

ANSI/ASHRAE 2001 defines ventilation as follows:
Ventilation is the process of supplying or removing air by natural (including infiltration) or mechanical means, to or from a space, for the purpose of controlling air contaminant level, humidity, or temperature within the space.

The choice of ventilation system is, therefore, primarily a choice that is governed by the location of the building, the quality of the outdoor air, how the building is going to be used and the user's needs. Most important, however, is that the quality of the chosen solution is sufficiently high, that the plant is installed correctly and that service and maintenance are not neglected. To meet the most important challenges we must, to a much greater extent:

- Focus on air quality – not on air quantity.
- Realize that the problem is not one of ventilation but one of pollutants – ventilation is a part of the solution.
- Accept that the quality of even the best ventilation systems will, over time, be reduced to a level determined by the competency of the operating staff.

FIGURE 1. Different types of air intakes for public and commercial buildings. Some of the intakes have not been designed or located according to Norwegian regulations and recommendations. Those shows bottom-right are substandard in all respects.

AIR INTAKES – AN IMPORTANT COMPONENT

An important, but often underrated component in most installations, is the air intake. Together with the air filter, its task is to prevent waste products, air pollutants, insects and other creatures from entering the HVAC system. For example, every form of waste or pollutants – such as leaves, road dust, exhaust fumes and combustion products – as well as other pollutants – such as pollen, mould fungi, microorganisms, and organic and inorganic substances from more undefined sources – must be kept out. All these can be removed from the air, to a reasonable extent, by designing and locating air intakes in suitable ways – and by installing efficient filters.

Depending on the type of building and the size of the plant, air intakes can vary from simple trickle ventilators above windows to large, centrally located units integrated into facades and intake towers. The following discussion focuses on large buildings, such as schools, offices, public buildings and commercial buildings. The most common types of air intake are:

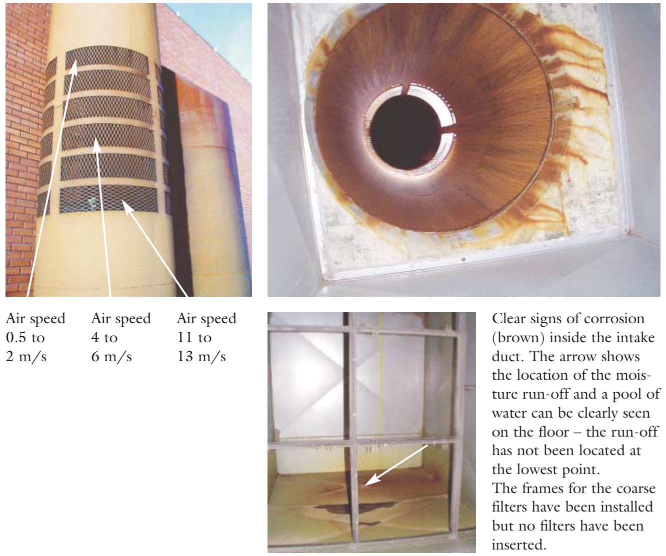
- Intake grilles/fixed louvres in walls
- Swan neck intakes, ribbed hoods and ventilation hoods for roof and ground-level installation
- Combined exhaust and intake hoods
- Wall units
- Horizontal intake grilles at ground level

The locations of air intakes are, unfortunately, seldom included an architect's first sketches. Now and again this will mean that rather unusual air flow solutions have to be adopted, as compromises are nearly always needed when designing air intakes. HVAC engineers want as low speeds as possible at the air intake to prevent precipitation and pollutants from entering the system. This can result in a large intake grille being located in a dominating position on a facade, which could spoil the architecture of the building.

FIGURE 2. Left, a primary school in Trondheim, Norway situated on a hill on the edge of the town close to vegetation and trees. Right, the outdoor air intake and the exhaust air outlet. Rain and snow falling on the intake can be easily sucked into the ventilation system. Photo: Olav Struksnes, Project thesis 2002, EPT-NTNU.

However, a large and correctly located air intake is not a guarantee against problems, as the design of the ducting section behind the intake will affect the flow of the air through the grille. If this section on the inside of the grille is not carefully designed, it could result in an uneven speed distribution across the intake. Measurements carried out in numerous buildings have, unfortunately, shown that this is quite often the case. This is a problem that arises primarily in rectangular and narrow air intakes as well as in air intake towers. A low and even air speed – preferably less than 1 to 1.5 m/s – is often essential to obtain good results.

Figures 2 and 3 are good examples of how well-located air intakes can still give rise to very uneven air speed distributions across the grilles, as



Clear signs of corrosion (brown) inside the intake duct. The arrow shows the location of the moisture run-off and a pool of water can be clearly seen on the floor – the run-off has not been located at the lowest point.
The frames for the coarse filters have been installed but no filters have been inserted.

FIGURE 3. An outdoor air intake for the primary school in Trondheim, Norway, shown in Photo 2. Frames for intake filters can be seen but none have been installed. Photo: Olav Struksnes, Project thesis 2002, EPT-NTNU.

no attention has been paid to the pressure distribution across the intake. The result is that rain and snow are easily sucked into the system.

Moisture, rain and snow entering the ventilation system via the air intake can cause corrosion and operational disruptions. If moisture is retained in the filter and other components, this can also result in the ventilation air having poorer quality. Insufficient cleaning combined with free moisture in the ventilation system can become a source of hygienic and aesthetical problems. There are also indications that microbic activity, combined with normally occurring air pollutants, can lead to serious problems in buildings with complex designs and advanced building services installations.

FIGURE 4. A primary school in Trondheim, Norway, on the outskirts of the town. Note the vegetation and trees nearby. Photo: Olav Struksnes, Project thesis 2002, EPT-NTNU.

In other instances it is obvious that insufficient insight and understanding are the causes of problems in air intakes – well illustrated in Figures 4 and 5. Air intakes built into roofs or located at ground level are bound to fail when it comes to tackling problems caused by moisture and pollutants.

Figure 4 shows a primary school comprising two detached buildings. The air intakes are almost completely integrated into the roof and designed in such a way that snow and other pollutants such as leaves, insects and bird droppings can easily collect in front of them. Their lo-

FIGURE 5. A primary school in Trondheim, Norway. Countryside environment with forest and vegetation close by. Photo: Olav Struksnes, Project thesis 2002, EPT-NTNU.

cations make them very difficult to access for inspection and cleaning. Figure 5 shows a free-standing school building in the countryside. Here the problem is reversed: the air intake has been located at ground level and close to the building, which means that all sorts of pollutants can easily fall into the air intake. Furthermore, there is a narrow concrete passageway with a rough surface finish, making both inspection and cleaning difficult.

The need to de-ice aircraft in wintertime is well-known in those parts of the world where temperatures fall below zero – but this procedure should not be necessary for air intakes in modern school buildings. And if it is, it should be done by using a more advanced piece of equipment than a broom handle! See Figure 6.

FIGURE 6. De-icing the air intake for a school building in Trondheim, Norway. The school was built in 2004 and the photos taken in early January 2006. Photo: Fredrik Hedeman.

Another recurring problem is that air intakes are often placed too close to exhaust air outlets, which increases the risk of short-circuiting the exhaust air and mixing it with the intake air. This can be seen in Figure 7. In this case, the exhaust air outlet has been located so close to the surface of the roof that the relatively warm exhaust air in the winter will melt snow falling onto the roof. Together with pollutants on the surface of the roof, this could cause undesirable microbic growth. Heavy snowfalls might even block the exhaust outlet completely.

FIGURE 7. Unsuitable location of an outdoor air intake in relation to the exhaust air outlet. There is a great danger of short-circuiting between the two systems and undesirable microbic growth on the surface of the roof – microbes could then find their way into the air intake. Photo: Frode Frydenlund, SINTEF Energy Research, 2007, Trondheim, Norway.

AIR INTAKES AND HEALTH PROBLEMS

Moisture in the air intake, dirty mechanical components and fouled ducts are often underestimated problems when air quality and health problems are discussed. The combination of moisture, dust and organic substances can cause both fungal and bacterial growth. Mould fungi are greatly suspected of causing allergies and, not least, of making problems worse for those who are already hypersensitive. Furthermore, air intakes in close proximity to pollen-generating plants and trees are risk factors if the filters that have removed the pollen from the air become wet, allowing soluble allergens to be freely transported through the ventilation system to occupied rooms.

Even individual types of bacteria – especially legionella bacteria – can be a health risk if the air intake is located in the vicinity of an infected cooling tower or scrubber. Aerosols – in the form of small drops of water containing legionella bacteria – can, in unfortunate circumstances, be spread to the outdoor air from the polluted plant and remain in the air for relatively long periods of time. They can then be sucked back into ventilation systems together with the ventilation air and this could constitute a serious health risk – especially for people with reduced immune defence. It is therefore essential that architects and HVAC design engineers have basic insights into microbiology and fundamental knowledge of the possible consequences to health and comfort when people are exposed to harmful microorganisms.

Bacteria are normally divided into two categories: Gram-positive and Gram-negative bacteria. Gram-positive bacteria are the most common type in our general surroundings. They are very resistant to dryness and heat, and are very important ingredients in the skin's defence mechanisms against infections. Gram-negative bacteria, on the other hand, live in water and die quickly in dry environments. This is why it is unusual to find Gram-negative bacteria in the outdoor air and, consequently, in air intakes. If they are found in the supply air in a ventilation system, this could be because there is a source of moisture in the system.

Fungi can be divided into two categories: yeast fungi and mould fungi. The majority of fungi found in the outdoor air are mould fungi. Bacteria and fungi differ in size:

- Bacteria are approximately 0.5 micrometres in diameter and can be classed as either round (coccus) and long (rod-shaped or bacillus)
- Yeast fungi are 2 to 5 micrometres in diameter.
- Mould fungi are 2 to 200 micrometers in diameter.

Physical factors such as moisture, temperature, UV light and pH value can affect the presence of different microbes. The availability of organic substances can be a deciding factor for their survival and will affect the number of microbes in the environment. The type of microbes that actually exist in a certain environment will be finally determined by the mutual competition between them – in which antagonism (mutual opposition) and symbiosis (mutual support) are decisive factors. It is, therefore, easy to understand the importance of avoiding free moisture or condensation in the ventilation system's equipment and ducting. And, if unprotected slabs of mineral wool are used for internal insulation, the risk of problems occurring is acute. Without free moisture, this risk is greatly reduced and it is here that keeping air intakes dry comes into the picture.

The composition of microbial communities and the possibility of viable spores being created determine whether the air flora is regarded as being natural or unpleasant by the occupants in a room. A number of types of mould fungi are thought to be the cause of respiratory problems in indoor environments where microorganisms can be found. This is especially true of the following mould fungi: *Alternaria, Aspergillus, Aureobasidium, Cladosporium, Fusarium, Mucor, Paecilomyces, Penecillium* and *Phoma*. An important question here is whether microorganisms can adapt to the environment in a ventilation system and change character so that they can grow at body temperatures. This could mean that fungi that are normally harmless in their free states could mutate and become harmful in ventilation systems. Microbiologists have been researching this field for many years [ISIAQ Guideline TFI-1996].

In some instances, the ducting system can function as an effective distribution path for pollutants generated far from the occupied rooms. Return air systems are very well known to be problematic where air pollutants and spreading of infections are concerned. Problems occurring at one point in a system have been clearly observed to multiply within the ducting [Hays et al., 1995].

A typical example of a source of infection being spread via air intakes and through ducting systems is the well-known legionella bacteria that cause, among other things, Legionnaires' disease and Pontiac fever. Legionnaires' disease is a specially dangerous type of bacterial pneumonia that was named after the epidemic that broke out among American legionnaires attending a congress in Philadelphia in July 1976. Around 30 of the 180 or so cases were fatal.

The disease is caused by the bacterium called *Legionella pneumophila*. This bacterium is common in HVAC plants, cooling tower sumps and hot water systems (cooler than 55°C), i.e. just about everywhere water can be found. The disease is spread by water droplets in the form of aerosols. Contraction seems to depend on age, sex, general health condition and degree of exposure to the bacteria. Especially vulnerable are old or middle-aged males with reduced immune defence and other people with poor immune defence.

Pontiac fever is caused by a particular type of legionella bacterium and develops as an illness similar to influenza. It is short-termed with symptoms including fever, shivering and headaches. The incubation time is between 4 and 66 hours, most often between 36 and 48 hours. Medical treatment is not necessary as the disease abates by itself – often within 2 to 5 days. It is not known whether there have been any fatal cases of Pontiac fever.

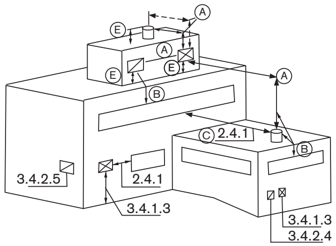
Unlike Legionnaires' disease, Pontiac fever can be contracted by a majority of those subject to the infection source (aerosols containing legionella bacteria) – more than 90% rates have been reported. Pontiac fever first became known after an epidemic outbreak among visitors and employees at the County Health Department in Pontiac, USA in 1966. Since then, epidemic outbreaks have occurred frequently around the world. In addition to cooling towers, scrubbers, evaporative condensers and air humidifiers, it is also known that hot water systems, showers and indoor fountains are possible risk and exposure sources.

Bearing all this in mind, one might imagine that the design and location of air intakes were subjects that were taken seriously – in other words, that components were planned and designed for each separate installation to prevent free moisture in the form of snow and rain from entering ventilation systems. Unfortunately, in far too many buildings, this is still not the case. And this in spite of the fact that, for many years, both national and international recommendations and regulations for locating air intakes in facades, based on knowledge about pressure drops and recommended speed distributions, have been available. No matter whether simple manual calculations or advanced CFD, Computational Fluid Dynamics, simulations are used, it can be seen that there are still differences, sometimes large ones, between theory and practice. When air intakes are planned and designed, the following aspects, at least, should be taken into account:

- Motor traffic and road dust
- Industrial pollutants
- Vegetation (natural and planted)
- Location of exhaust air outlets in own and neighbouring buildings
- Distances to cooling towers
- Precipitation (rain and snow)
- Directions of prevailing winds
- Opportunities for inspection and cleaning
- Choice of materials used
- Conditions around the air intake
- Location in relation to exhaust gases from the building's own heating plant
- The use of wire netting in front of the intake grille, which could become blocked due to build up of freezing rain

REGULATIONS CONCERNING THE DESIGN AND LOCATION OF AIR INTAKES

Based on experience, insight and knowledge, clear and well-founded recommendations have been available for many years regarding the location of air intakes. Figure 8 shows two examples: to the left, a more than 20-year-old Finish recommendation and to the right more recent recommendations from a working draft produced in connection with preparations for a proposal for a CEN standard.



Location of openings, recommended distances

Category (examples)	A m	B m	C m	D m
EHA 1	2	2	1	2
EHA 2	2	3	2	2
EHA 3	4	6	3	5
EHA 4	6	10	5	8

Key:
A = minimum distance above windows which can be opened
B = minimum distance from windows, which can be opened, at the same level or above
C = minimum distances from grades, paving etc
D = minimum distance from adjacent building lot

FIGURE 8. Example of locations of outdoor air intakes from the Finish Building regulations 1987 (left), and from CEN Working Draft prEN 13779, February 2002 (right). Note that the distances A, B, C etc, in the drawing and table, do not refer to the same variables.

In the EN 13779:2004 ventilation standard the following specific and clearly formulated recommendations can now (2007) be read:

- "No air intake should be located closer than 8 metres of horizontal distance from a garbage collection point, a frequently used parking area for three or more cars, driveways, loading areas, sewer vents, chimney heads and other similar polluting sources".
- "Special attention should be paid to the location and shape of openings in the vicinity of evaporative cooling systems in order to minimise the risk of spreading impurities into supply air. No air intake openings should be placed in the main wind directions from evaporative cooling systems. In addition, good maintenance of cooling tower systems is important".
- "No air intake should be positioned on a facade exposed to a busy street. Where this is the only possible location, the opening should be positioned as high above the ground as possible".
- "No air intake should be positioned where a back-flow of exhaust air or a disturbance from other pollutants or smelling emissions is expected".
- "No air intake should be positioned just above the ground. For example, a distance at least 1.5 times the maximum expected thickness of snow between the bottom of the intake and the ground is recommended".
- "On top of the building or when the concentrations on both sides of the building are similar, the intake should be arranged on the windward side of the building".
- "The air intake opening adjacent to unshaded places, roofs or walls should be arranged or protected so that the air will not be excessively heated by the sun in summer".
- "Wherever the risk of penetration of water in any form (snow, rain, mist, etc.) or dust (including leaves) into the system is apparent, an unprotected opening should be dimensioned for a maximum air velocity in the opening of 2 $m \cdot s^{-1}$ (see also EN 13030)".
- "The height of the bottom of an air intake opening over a roof or deck should be at least 1.5 times the maximum yearly expected thickness of snow. The distance can be lower if the formation of a layer of snow is precluded by means of, for example, a snow shield".
- "Consideration should be given to the possibility for cleaning".

On the basis of these general recommendations, it is quite clear which parameters should be considered. Both manual calculation methods and CFD simulation programs are available that can be used for designing and locating air intakes and that indicate the respective consequences of the different choices.

DESIGNING AIR INTAKES

No matter what the design of the actual intake, the inlet area must be large enough to prevent an unnecessarily large pressure drop across the grille. It is also important that the volume of intake air is evenly distributed over the whole of the inlet area. To achieve this, a number of principles must be considered [Berner et al., 2007].

Figure 9 shows an example of a rectangular intake connected to ducting at one end. If the intake is designed in this manner, the speed of the intake air closest to the ducting will be higher than that at the far end of the intake. Although it is practical to design the intake chamber with a uniform square cross-section, the resulting air speed imbalance must be kept within acceptable limits.

The imbalance will depend on the cross-sectional area of the chamber in relation to the grille area and the pressure drop across the grille. The reason for the imbalance is that the air speed in the chamber parallel to

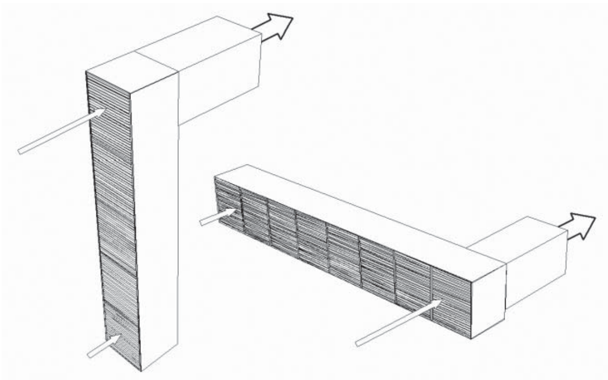


FIGURE 9. Rectangular air intakes with single end ducting connections. Illustration: Luftinntak, Draft report SINTEF Energy Research 2007 [Berner et al., 2007].

the grille becomes so large that the static pressure is significantly reduced. When the static pressure inside the grille is reduced, the speed of the inflowing air increases, as shown in Figure 10. It is therefore important to make sure that the relationship between the highest and lowest speeds through the grille, u_2/u_1, does not exceed a certain limit.

To meet this requirement, the speed of the air (the longitudinal speed behind the intake grille) must not exceed a certain limit, u_{Lmax}. The maximum speed is given by:

$$u_{Lmax} = \sqrt{\left[\left(\frac{u_2}{u_1}\right)^2 - 1\right]\frac{2\Delta p_1}{\rho}} \qquad (1)$$

where:

u_{Lmax} is the maximum longitudinal speed behind the intake grille in m/s

u_1 is the speed of the air through the intake grille furthest from the ducting connection in m/s

u_2 is the speed of the air through the intake grille closest to the ducting connection in m/s

Δp_1 is the pressure drop across the intake grille in Pa

ρ is the density of the outdoor air in kg/m^3

Note: u_1 and u_2 are speeds based on the flow volumes per gross unit area of the grille

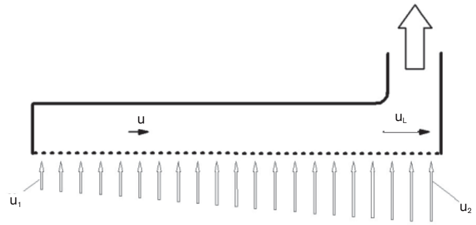


FIGURE 10. The speed of the air behind the grille is greatest closest to the supply air duct. Illustration: Luftinntak, Draft report SINTEF Energy Research 2007 [Berner et al., 2007].

The following example is based on the equation above:

Assume that an outer wall grille is to be designed for an air intake of 70 000 m^3/h. The height of the intake is limited to 1.2 m.

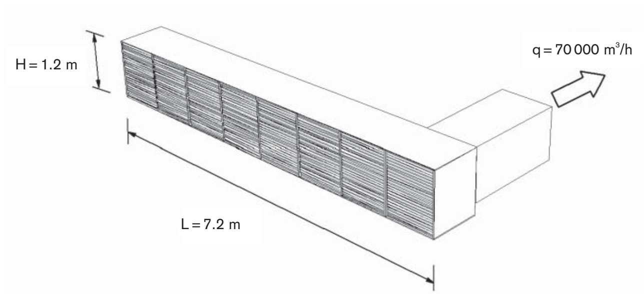


FIGURE 11. The air intake in the example. Illustration: Luftinntak, Draft report SINTEF Energy Research 2007 (Berner et al., 2007).

An intake length of 7.2 m is chosen. This gives the following figures:

Length	L	7.2 m
Height	H	1.2 m
Gross area	A	8.64 m^2
Air flow rate	q	70 000 m^3/h
equivalent to		19.4 m^3/s
Intake speed	u_0	2.3 m/s
Pressure drop	Δp_1	27 Pa

The grille characteristics are shown in Figure 12.

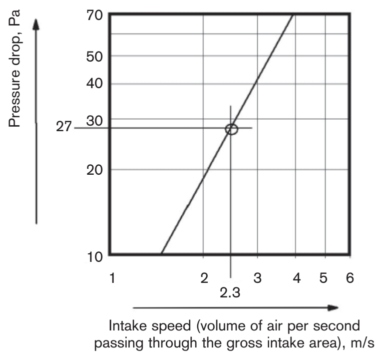


FIGURE 12. Characteristics for an external wall grille.

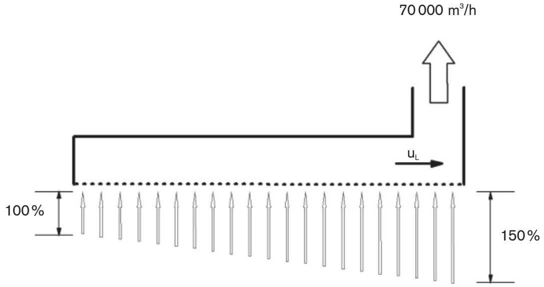


FIGURE 13. Assumptions in the example. Illustration: "Luftinntak", Draft report SINTEF Energy Research 2007 [Berner et al, 2007].

The highest intake speed must not be more than 50 % greater than the lowest intake speed. This means that the longitudinal speed on the inside of the grille must not exceed:

$$u_{L\max} = \sqrt{[1.5^2 - 1]\,\frac{2 \cdot 27\,\text{Pa}}{1.25\ \text{kg/m}^3}} = 7.35\ \text{m/s} \qquad (2)$$

The cross-sectional area of the chamber behind the grille must be at least about:

$$A_{\min} = \frac{19.4\ \text{m}^3/\text{s}}{7.35\ \text{m/s}} = 2.64\ \text{m}^2 \qquad (3)$$

If the height of the chamber is the same as that of the grille, i.e. 1.2 m, the width of the chamber will be:

$$B = \frac{2.64\ \text{m}^2}{1.2\ \text{m}} = 2.2\ \text{m} \qquad (4)$$

The air intake will then have the form shown in Figure 14. The intake ducting has been designed for an air speed of 8 m/s and there is a tapered section between the intake duct and the chamber behind the grille.

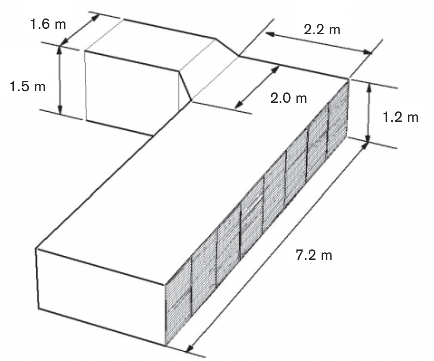


FIGURE 14. The air intake with the calculated dimensions. Illustration: Luftinntak, Draft report SINTEF Energy Research 2007 [Berner et al., 2007].

The solution shown in Figure 14 causes an imbalance of air speeds through the grille but within acceptable limits. A way of completely avoiding this imbalance is to reduce the space behind the intake grille. The reduction must extend from the intake ducting to the far end of the

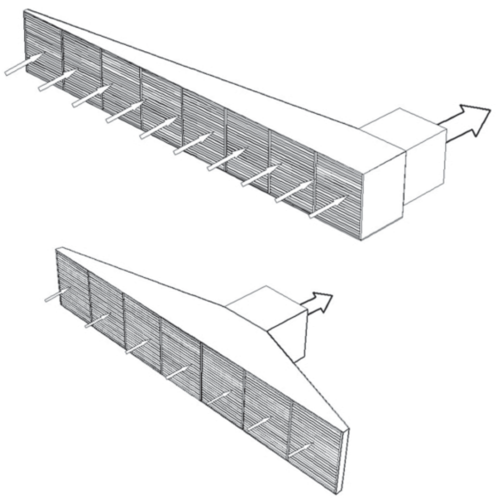


FIGURE 15. Intakes with reduced spaces behind the intake grilles. Illustration: Luftinntak, Draft report SINTEF Energy Research 2007 [Berner et al., 2007].

grille. The top illustration in Figure 15 shows an intake with ducting connected at one end and the lower illustration shows how the intake can be designed if the connection is centred.

Figure 16 shows an air intake designed to reduce the air speeds so that they are the same across the whole of the intake grille. The design data used is similar to that in the example above but with a maximum air speed of 6 m/s in the intake chamber.

More advanced calculations can, of course, be carried out using CFD simulations. A frequently used CFD program is Fluent Air Pak. Figure 17 shows two examples of calculations of speed distribution through an air intake tower, left, and through two facade-mounted rectangular air intakes, right. High air speeds are shown in red. It can be seen that the upper part of the tower has practically no intake function at all. The air flow through the facade-mounted intake, to the right, is 7.5 m^3/s in both cases and the grille area is 3 m^2. This means that the average air speed in both cases is 2.5 m/s. Here, we can see the results of the duct design and their change in direction downstream of the intake. Red denotes high speeds. It is quite clear that the square intake has a considerably more even speed distribution than the rectangular intake. It can also be seen that the design of the ducting connected to the intake affects the speed distribution over the intake area.

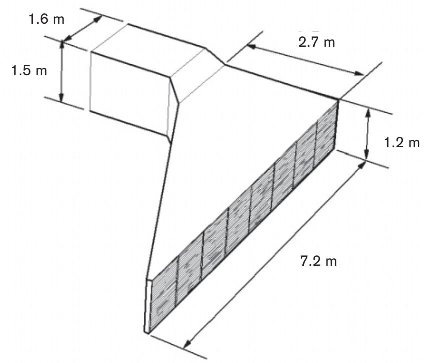


FIGURE 16. Air intake with tapered chamber. Illustration: Luftinntak, Draft report SINTEF Energy Research 2007 [Berner et al., 2007].

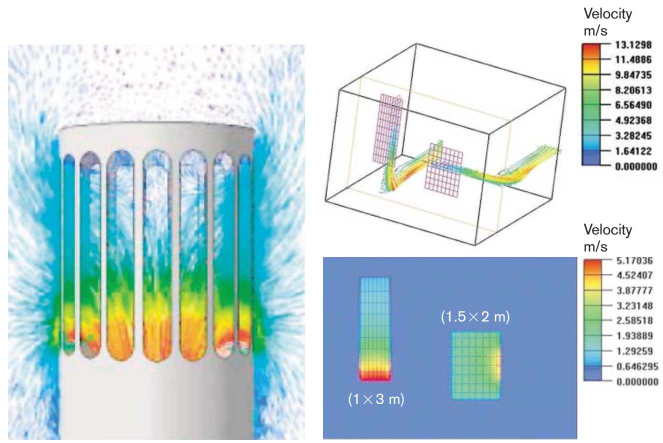


FIGURE 17. Air movements in and around an air intake can be calculated using a Computational Fluid Dynamics program such as Fluent Air Pak. Red – high speed, blue-green – low speed. Illustrations: Left: A.S. Wide, Degernes, Norway. Right: Frode Frydenlund, SINTEF Energy Research, Trondheim, Norway.

Figure 18 shows an example of a CFD simulation of speed distribution through an air intake tower together with a photograph of the duct under the intake. Piles of snow and waste material have been formed as predicted in the simulated air speed illustration.

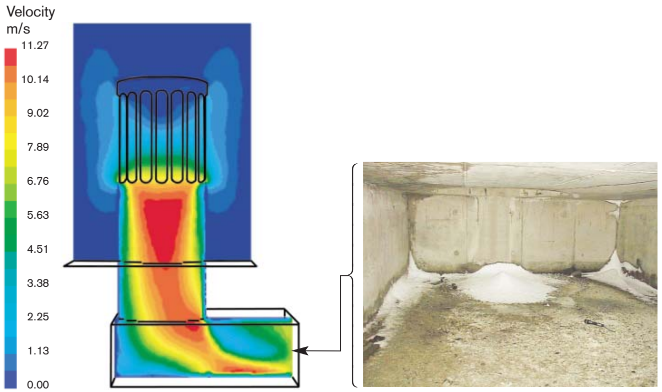


FIGURE 18. Flow simulation for an air intake. The colour scale to the left denotes the air speeds. The photograph to the right was taken in the intake duct on a winter day and shows that a significant amount of snow has been sucked in and allowed to accumulate at the rear of the duct.

COMMENTS AND CONCLUSIONS

Traditionally, engineers have focused on moisture problems in the structures of buildings while moisture in ventilation systems has almost been ignored. This is paradoxical, as the task of the ventilation system is to supply the building with fresh air. Ventilation systems are often installed in closed-off spaces and are generally regarded by both property managers and tenants as having low status.

Moisture, dirt and heat are not harmful as such, but in combination they can create the conditions for growth of harmful microorganisms and the emission of chemical substances from moisture-damaged materials, which in turn can constitute a health risk. Of these three factors, moisture plays a special role, as it is the most essential criterion for mould fungi growth [Bryn et al, 2006].

In this respect, very little attention is paid to air intakes as parts of HVAC systems. Free moisture in the form of mist, rain or snow can enter a system if preventive measures are neglected. Unfortunately, we can see far too many examples of systems – potentially perfect for supplying high quality ventilation air – not reaching up to standard because, right from the very start, their chances of doing so have been ruined by poorly designed and often wrongly located air intakes. This is a serious paradox when we know that the necessary competence for solving problems like these has been available for many years. This is why we can safely say that the cause of many of the problems, and ones that occur year after year, is due to a lack of competency – not a lack of regulations. Competency in this respect means the sum of the knowledge, insights, attitudes and actions of those involved in ventilation projects.

If architects, building engineers and HVAC engineers cannot discuss the designs of air intakes and their locations before detail design work begins, then everything is lost. We risk seeing results like those shown in Figures 19 and 20. Figure 19 shows an intake tower that has been designed without using any common sense and in disregard of current reg-

FIGURE 19. Air intakes at easily accessible heights can become waste bins. Abandoned building waste does not improve the situation. Photo: Frode Frydenlund, EPT-SEfAS.

ulations. It is quite clear here that problems will occur due to rain, snow and different sorts of pollutants. Furthermore, a lot of building waste has been left in the intake well. The resulting hygienic level of the air is very low.

Users, too, must also be made aware of what can happen. When smoking was banned indoors it was still allowed outdoors. This resulted in smokers looking for alternative places to smoke, see Figure 20. This is a secluded spot at the back of a large office building. Chairs and tables have been put out to make it a pleasant venue. However, both the floor and, more significantly, the light well along the wall of the building have been used as ashtrays. Any effects that this might have on the function of the air intake have been completely disregarded. Cigarette ends and other pieces of waste left by smokers have drastically reduced the function of an incorrectly located and poorly designed air intake. The results can be seen in the right-hand photograph: an air intake with piles of waste that are drenched in water every time it rains, causing very unpleasant smells. The intake is also a potential incubator for undesirable microbacterial activity.

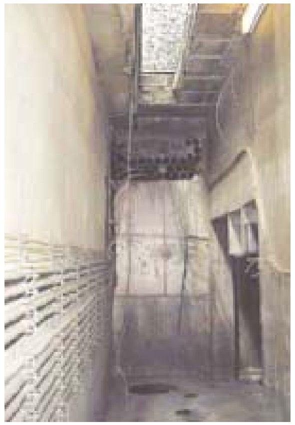


FIGURE 20. Intake grilles located at ground level. The open space is often used by smokers and the grille is partially blocked by cigarette ends and other debris. Photo: Frode Frydenlund, EPT-SEfAS.

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19. DUCTING SYSTEMS – LOCATION, DESIGN, INSPECTION AND CLEANING

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This chapter discusses the importance of designing and installing efficient ducting systems, how they are integrated into buildings, quality requirements, and how function and operating costs are dependent on having an airtight system.

WHY IS IT IMPORTANT TO HAVE AN EFFICIENT DUCTING SYSTEM?

When two or more rooms are connected to a common ventilation system, a ducting system will be required to supply, distribute and remove the air. There are, however, a number of reasons why the system, from a functional point of view, might be unsatisfactory, with problems caused by:

- Air leaking into or out of the ducts because they are poorly sealed.
- High air speeds giving rise to disturbing noise levels.
- Dust and other pollutants in the ducts resulting in health problems.

It is important that duct systems are designed with due care – they will normally be used for a great number of years and will have a significant effect on the amount of energy used by the ventilation system. The total amount of energy used by a ventilation system is often equally divided between the ducting system and air handling plant.

Another important aspect is the role played by the ducts in transmitting, creating and damping noise in a building. Ducts passing through adjacent rooms can transmit sound and impair the sound reduction properties between them.

Careful planning and common sense are always required when aiming for acceptable results. In most cases, ductwork will form an integral part of a ventilation system, installed with the purpose of providing an acceptable climate and good air quality. However, it does not follow that any noise created by the system is acceptable: a great many people experience a true feeling of relief when the ventilation system is switched off at the end of the day and silence returns. Dissatisfaction caused by noisy ventilation systems must be avoided and it is important to pay as much attention to the acoustic design of a system as to other air quality and thermal climate factors that affect well-being and comfort.

Silence – or the absence of noise – is now often a rare commodity and too little can lead to stress and discomfort. And remember, prevention is better than cure – problem-solving at a later stage will be more difficult and expensive, and require more time and effort to provide satisfactory solutions. It is also harder to convince people who have been discontented that they should now be satisfied.

More about this topic can be found in Chapter 26/Sound and sound attenuation, in which the creation and damping of noise in ductwork is also discussed.

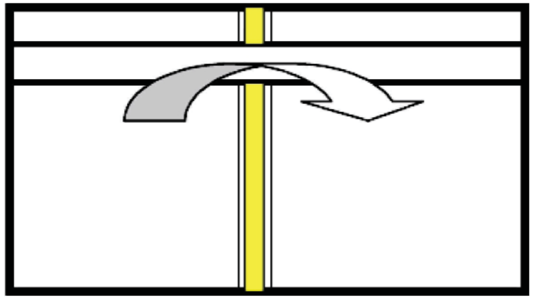

FIGURE 1. Crosstalk – noise transmitted between two rooms via a ventilation duct.

Development and stricter requirements

DUCTS AND COMPONENTS

Manufacturing methods for ventilation ducts and components during recent decades have progressed from manual and time-consuming work, relying on the professional skills of sheet-metal workers, to large-scale industrial production and the assembly of complete systems by fitters on site. At the same time, the system of using tailor-made ducts and components, made to measure on site, gave way to industrially manufactured and stocked ducts, with standardized dimensions and component design, for example of bends and junctions.

A great and important step in this development work was taken when machines were introduced in the 1960s to manufacture cylindrical, spiral-seam ducts in standard sizes. Previously, circular ducts had hardly been used at all.

Demands for airtight ducts were made at the beginning of the 1970s and they resulted in the development of rectangular ducts that were fitted with different types of rubber seals at the joints, replacing putty and tape that had been used before. The circular ducts were also later fitted with rubber seals that were pressed into place when the duct sections were joined together, see Figure 2.

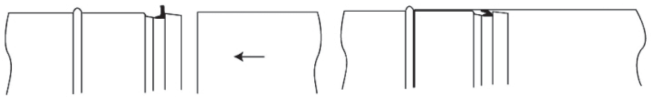

FIGURE 2. Joining circular duct sections fitted with rubber seals.

To prevent the joints from sliding apart they are then fixed in position using pop rivets or special screws. One manufacturer has recently introduced a solution whereby the joints are firmly held together without rivets or screws, which greatly facilitates the installation work.

Greater demands on airtightness were followed by demands to carry out checks when the ventilation system was put into operation. As better design solutions were produced, the demands on airtightness increased, see the section on requirements, tests and inspection below.

Types of ducts

A ducting system normally comprises a combination of rectangular ducts, installed closest to the supply air handling equipment where the air flow is the greatest, and circular ducts further downstream in the system, where the air is distributed to the different parts of the building and the different rooms.

Duct sections are normally made of galvanized sheet steel. In corrosive environments, occurring inside or outside the ductwork, stainless steel, aluminium and sheet-steel coated in an alloy of aluminium and zinc are also used. Even synthetic materials (PVC, polyamide, etc) can be used, for example, for extract air ducts in laboratories.

As mentioned above, duct cross-sections can be circular and rectangular. In some countries, flat-oval ducts are used as well. These are made by either pressing together or drawing apart circular ducts.

Rectangular ducts

Rectangular ducts are manufactured in heights and widths according to the EN 1505 dimensioning standard, in steps of 100 mm up to 600 mm and thereafter in steps of 200 mm. This means that there are a great number of possible combinations for straight duct sections and even more for components and specially formed fittings, such as bends and junctions, which means that it is impossible for suppliers to keep them all in stock. The length of the duct sections is normally limited to 2.4 m, which corresponds to the standard length of the galvanized sheets used in their manufacture.

The sections are normally joined using slide-on cleats, which press the flanges of the duct sections, and the rubber seal in between them, together. The larger the ducts, the stronger the cleat required, see Figure 3. The four corners of the rectangular duct sections are often fitted with bolt holes to ensure that the sections are kept in a straight line and at the same time facilitate the assembly of the cleats. Sufficient space must always be left around the flanges for the cleats and assembly tools.

Pressure variations in the ductwork can cause the flat sides of the ducts to vibrate and this could cause disturbing noise. To prevent this, the sides are normally stiffened by cross-creasing, see Figure 4, by transverse stiffening, see Figure 5, or by attaching exterior ribbing.

Struts inside ductwork should be avoided, as they can cause noise and make cleaning more difficult.

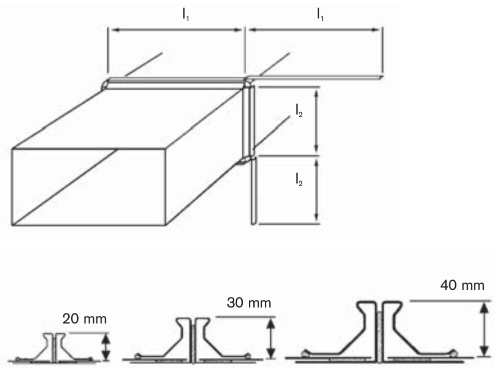

FIGURE 3. The larger the duct, the larger the flanges required. Extra space is also needed so that the cleats can be attached.

FIGURE 4. Cross-creased duct.

FIGURE 5. Transverse-stiffened duct.

Circular ducts

All circular metal ducts manufactured today are made from rolls of galvanized steel sheet, aluminium or stainless steel that are fed into a spiral folding machine. There are a number of different machine manufacturers although all the machines, in principle, are quite similar and are based on the original design concept. The metal strip is rolled into a standard-diameter duct with standardized allowable tolerances with regard to finished dimensions.

Unlike the rectangular ducts, the circular ducts are made in fewer sizes. The standardized dimensions are based on a mathematical series in which successive diameters increase in the ratio of $1{:}\sqrt[3]{2}$ (i.e. by about 27%). Note that the one-third octave bands used in building acoustics, see Chapter 11/Building acoustics, follow the same mathematical series.

The following duct diameters are commonly used in Europe:
63, 80, 100, 125, 160, 200, 250, 315, 400, 500, 630, 800, 1000, 1250 and 1600 mm.
Other diameters, used in other countries, include:
355, 450, 560, 710, 900, 1120 and 1400 mm.

The length of the circular duct sections is only limited by transport regulations – the machine can continue to produce a duct until the strip runs out or the workshop becomes too small! Duct sections are normally cut into 3 m lengths, though lengths up to 6 m have been produced, transported and installed. If long duct sections can be used, fewer joints will be required and this will reduce installation time and leakage risks. In one special case, when a large number of large-diameter (1600 mm) duct sections were needed, the spiral folding machine was moved to the building site to avoid transportation problems.

Components for circular ducting, for example, bends, can be turned in any direction, which means there is only need for one type of bend for each duct dimension, as opposed to the numerous types required for rectangular or flat-oval ducts.

Flat-oval ducts

One disadvantage of circular ducts is that they cannot be made flatter when narrow spaces have to be negotiated. This is where the use of rectangular ducts can be advantageous, even if a number of parallel circular ducts, instead of a single low and wide rectangular duct, can be used, see the sub-section *Planning space carefully* below.

A compromise that is used in a number of countries is the flat-oval duct, mentioned above. The sections are manufactured as circular lengths and then pressed together or drawn apart using special tools to give them their flat-oval shape. They are used instead of rectangular ducts in places where heights are limited. These ducts should only be used where there is an over-pressure in the duct, i.e. in supply air ducts, otherwise there is a risk of them becoming even flatter than intended. On the other hand, the over-pressure must not be so great that it causes the duct to revert to its original round shape.

One of the disadvantages of flat-oval ducts, compared to circular ducts, is that joining the duct sections and components, as well as the design of the components, is much more complicated. Additionally, as in the case of rectangular ducts, there are so many possible combinations of heights and widths that prefabrication and the carrying of stocks becomes impossible.

Strength

Duct systems must be able to meet requirements regarding:

- Mechanical strength
- Corrosion resistance
- Vibration fatigue

The ductwork must also be installed using fixings that are strong enough to withstand the loads that the system might be subject to. Many of these requirements are specified in European standards. Ducts are subject either to an over-pressure (supply air) or an under-pressure (extract air) and must be able to meet specific requirements in these respects, depending on their size.

The dimensions for rectangular ducts and components are stipulated in EN 1505 and the requirements for airtightness and mechanical strength in EN 1507.

The dimensions for circular ducts and components are stipulated in EN 1506 and the requirements for airtightness and mechanical strength in EN 12237.

The importance of correct corrosion protection

To ensure that the ducts will have an acceptably long operational life, it is essential that the correct material qualities be chosen, with respect to the environments in which they will be used. If the ducts are to be installed in corrosive environments, the standard quality – galvanized sheet steel – is not durable enough. Either extra treatment, by applying a sufficiently thick coat of paint of suitable quality, or a more corrosion-resistant material is required, see the sub-section *Types of ducts* above.

Table 1 shows how quickly a zinc layer can corrode in different environments. A common problem that arises in climate systems is caused by condensation dripping from a cold surface onto galvanized steel ductwork. This must be prevented by providing the cold surface with sufficient thermal insulation and a vapour barrier or by moving the condensation source away from the ducting. The reason why the rate of corrosion is so rapid is that condensation, like distilled water, does not contain salts.

TABLE 1. Corrosion rates for zinc in different environments.

Environment	Approximate rate of corrosion, µm/year
Indoors	<0.5
Inland countryside district	<1
Coastal districts	
Towns	1 to 3
Countryside	0.5 to 2
Industrial areas	2 to 10
Sea water	
North Sea	12 to 46
Baltic	about 10
Distilled water	50 to 200
Soil	500

Example:

Standard quality ventilation ducts are made of galvanized sheet steel class Z275. The class designation means that 1 m^2 of the sheet is coated with 275 g of zinc equally applied to both sides. This provides an average thickness of 20 µm of zinc. If the duct is installed indoors in a dry environment, it will take more than 40 years before the zinc layers have corroded to such an extent that the sheet will start to rust. If, on the other hand, the duct is subject to condensation with an approximate corrosion rate of 50 to 200 µm/year, see Table 1, it will start to rust after only a few months.

The importance of correct insulation

There are three reasons why ducts might have to be insulated and three corresponding types of insulation are available:

1. *Thermal insulation,* which creates a thermal barrier between the inside and outside of the duct.
2. *Fire insulation,* which prevents fire from spreading through the wall of the duct.
3. *Acoustic insulation* placed on the inside of the duct, which absorbs sound, or placed on the outside, which increases the sound reduction index of the duct wall, and thereby prevents sound from entering or leaving the duct. See Chapter 26/Sound and sound attenuation.

Sometimes there is a simultaneous need, for example, to meet thermal insulation and fire insulation requirements. The most cost-effective solution might be to combine these two requirements and choose a type and thickness of insulation that satisfies both. The requirement that is most important varies from case to case. Normally, fire protection requirements demand thicker insulation than energy-efficiency requirements.

Common to all applications is that the insulation material must be fireproof and mineral wool or fibreglass is therefore normally used. The insulation material is usually wrapped around the outside of the duct, except when the material is used for sound absorption. The insides of intake ducts, between the intake grilles and the air handling plant, used to be lined with thermal insulation and the metal duct then acted as a vapour barrier between the cold, dry air inside the duct and the warm, humid air on the outside of the duct. The disadvantage of this solution was that the outdoor air that was drawn into the system did not pass through any sort of filter. This meant that the air often contained dust and, together with raindrops and snowflakes, created viable, but highly undesirable, breeding grounds for microbial growth in the insulation material.

If the air in the supply air equipment is cooled to a temperature under the dew point of the air in the building, it is important that the insulation, enclosed in a suitable vapour barrier, is wrapped around the outside of the duct, i.e. where the partial vapour pressure of the air is higher than that of the air next to the cold duct.

It is extremely important that the vapour barrier, for example plastic sheeting or galvanized steel, is completely airtight, otherwise water vapour will penetrate, diffuse, into the insulation material and probably corrode the wall of the duct. Insulating material that becomes wet will loose most of its insulation capacity.

If acceptable from a hygienic point of view, the insulation material can be placed inside the duct wall and the wall will then act as a vapour barrier against the surroundings. When insulation material is used to line a duct, it is important that the material chosen can be cleaned using standard methods, see the section on cleaning ventilation ducts below. It is equally important that the material cannot emit particles into the air, i.e. cannot erode, at the air speeds encountered in the duct.

PLANNING THE DUCTWORK

Decisions about location and design must be made jointly

Although the HVAC design engineer will try to avoid long and difficult routes for the ductwork, unsuitably located shafts will make it difficult to design an efficient system.

The design and location of ductwork should therefore be carried out in consultation with the architect. It is important to remember that sufficient space must be reserved for building services installations. If consultations are carried out at an early stage in the project, the results can be advantageous to both parties. This is discussed further in the sub-section *Will the installations be pleasing to the eye?* below.

One or a number of systems?

The first logical step when planning a ventilation system for a building is to decide on whether it will be served by one single supply and extract air system or a number of systems. The decision should be based on the following factors:

- The size of the building and required air flows – the greater the needs, the more advantageous it could be to divide the system into sub-systems. In a large, low-rise building the ductwork will be extensive, costly and difficult to design, if all the air has to be supplied from a single point.
- The number of users/tenants – will they have different demands regarding operating times for the ventilation system? This could be the case if the building comprises both offices and shops. Separating the systems could reduce energy usage, as it will not be necessary to run the whole system just for the sake of one of the tenants.
- Will the users/tenants have different demands regarding air quality and thermal comfort? This will most probably result in the need for different technical solutions that will be easier to manage if separate systems are installed. If the users/tenants are going to pay for their respective solutions, having separate systems will make it easier to proportion costs between them.
- The creation of fire cells and other safety considerations could make it easier and safer to design the ventilation system with individual fire cells rather than to design a single system for the whole building.

Layout and low pressure drops

The design of the ducting system greatly affects the pressure drops and, consequently, the energy needed to transport the air through the system. The required fan power is given by the following equation:

$$P = \frac{q \cdot p_{tot}}{\eta} \qquad (1)$$

where:
P is the fan power in kW
q is the air flow provided by the fan, i.e. the nominal flow plus leakage, in m^3/s
p_{tot} is the total pressure rise across the fan, i.e. the pressure drop across the plant and in the duct system, in Pa
η the total efficiency of the fan

The fan power P can therefore also be expressed as:

$$P = f(q \cdot p)$$

and, as $p = f(q)^2$, then $P = f(q)^3$ (2)

The pressure in a duct system can be regarded as the energy that has been supplied by the fan and has been converted into kinetic energy (air flow). This is then irreversibly reduced due to friction against the duct walls or turbulence, for example, at bends or sudden duct enlargements. These losses, normally termed pressure drops or flow resistance, must be overcome by the fan so that the design air flows can be supplied via the supply air terminal devices in the system.

Pressure drops cost money, as they are directly connected to the energy used by the fan. It is important, therefore, that the design engineer calculates the pressure drops across plant and in the duct system and tries to reduce unnecessary losses.

The air flow in a duct is dependent on the reduction of pressure in the duct in the direction of flow. Pressure losses are caused by friction and local flow restrictions in the components.

Both types of loss are caused by local changes in speed:
- Friction corresponds to the force that is required to accelerate the air that leaves the low speed zone along the walls of the duct and moves into the high speed zone in the centre of the duct.
- Component losses correspond to the forces that are required to create local increases in average air speeds in the duct system.

To reduce the pressure losses in a system, it must be designed to be as smooth and unrestricted as possible:
- Sudden changes of cross-section and sharp bends without guide vanes should be avoided.
- The placing of duct components at a distance of less than five duct diameters from each other should be avoided.

Locations for fans and air handling plant

There are a number of pointers to be followed when deciding where to install fans and air handling equipment:
1. Avoid locations next to sound-sensitive spaces, such as conference rooms.
2. Choose locations close to the spaces that they are going to serve to minimize the length of ductwork required. This will save costs, energy and space.
3. Locate air handling equipment and supply air fans close to suitably located outdoor air intakes.

4. Fans and air handling equipment require regular service and maintenance and it must be possible to replace worn-out parts when required. Locate them to facilitate this work. Avoid locations that are difficult to access, for example, in lofts and on roofs, especially in cold climates and in high buildings. Carefully consider how maintenance work will be carried out and what will be required. Do not forget that spaces like these are working spaces for the maintenance staff and they should therefore be planned as such.

Location of outdoor air intakes and exhaust air outlets

Outdoor air intakes must be located where it can be presumed that the surrounding air is clean. It is best to locate them:
- High up on the rear side of a building away from traffic exhaust fumes.
- On the north side rather than on the sunny east, south and west sides.
- At a safe distance from exhaust air outlets – in the same or neighbouring buildings. Note prevailing wind directions, vertical and horizontal distances to other intakes and outlets.
- At a safe distance from cooling towers and evaporative condensers, to reduce the risk of legionella spreading to the building via the supply air system. *Legionella pneumophila* bacteria can be found in the small water droplets emitted, for example, from cooling towers.

The location of exhaust air outlets must not create problems in your own or any neighbouring building.

Location of duct shafts

Study the different floors in the building and how the supply air and extract air flows are to be distributed. Try to choose shafts that are located as centrally as possible. The more symmetrically the ducts can join up at the shafts, the lower the costs and space requirements for the ductwork.

Symmetrical branch systems in the riser ducts and in the connecting ductwork on each floor will reduce pressure drops and, consequently, the energy required to convey the air.

In large buildings that are divided up into a number of fire cells, it is often advantageous to separate the supply air and extract air ducts and locate them in separate shafts. The shafts can then be regarded as individual fire cells, provided that the shaft walls have been designed appropriately.

For structural reasons, the duct shafts are often placed next to the lift shafts. In long buildings, with a lift shaft at each end, it is often a good idea to place the supply air ducts in one of them and the extract air ducts in the other.

Note that the shafts must be accessible from every floor, both for installation work and future modifications. In large buildings, with a number of ducts in each shaft, the shafts often have their own lighting systems and are fitted with steel working platforms and inspection doors on each floor.

Symmetrical design

Symmetrical duct systems have a number of advantages and should be used wherever possible. When the design air flow for every terminal device is the same, a symmetrical layout will provide an equal pressure drop in all the terminals and installation work will also be much easier. Figure 6 shows a symmetrically designed sub-system, in which the supply air is introduced at point 1 and its flow is halved at every subsequent T-joint, until it reaches the terminal devices at the ends of the stub ducts. The air flow at each device is one eighth of the original flow at point 1.

On reaching each terminal device, the air will have passed identical duct components with identical dimensions and pressure drops. When symmetrical, branched sub-systems are used the commissioning work is

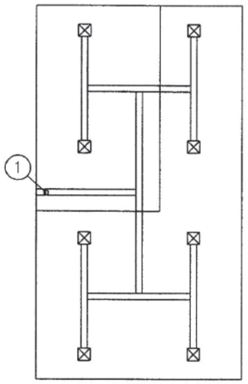

FIGURE 6. A symmetrical sub-system in which the supply air, introduced at point 1, is distributed via identical duct components on its way to the terminal devices.

simplified, as the pressure drops in the different terminal devices are the same and their dampers can be given the same setting. There will be no need for balancing dampers except, possibly, to distribute the air between different sub-systems, see Chapter 29/Balancing ventilation systems.

Designing installations to be flexible

A building is normally designed for a long operational life, often considerably longer than the operational life of the original building services installations. Demands made on the building and the building services can change over time. New tenants and changes in use of a building often mean that new demands are made on the services installations. It is therefore important to consider the demands for installation flexibility and the future uses of the building:

- Will the original installation provide reasonable margins – with regard to plant and ductwork – to be able to cope with moderate increases in air flows? Having low air flow rates in the ducts will make it possible to cope with higher loads whereas high initial levels of energy use and of noise in ducts, components and terminal devices could hinder future changes. Higher investment costs can be regarded as taking out an insurance policy, one that will pay off very generously, if and when functional changes have to be made in the system.
- Are the plant rooms, shafts, suspended ceilings etc sufficiently large to accommodate new plant and installations that will fulfil higher demands? Spaces and access routes that are too narrow will otherwise make it difficult or even prohibit future changes, which could cause the value of the property to fall. Even here, the initial investment in extra space could prove to be very profitable when future changes in building use or HVAC functions are required.

When choosing between these two alternatives – flexibility or more available space – the former should be chosen, if changes can be expected in the near future, i.e. within about ten years, and the latter if changes are first expected later on.

Special care must be taken when solutions are chosen that entail integrating the installations with the structural design of a building. This could lead to difficult and expensive rebuilding work if the ducts, for some reason, need replacing in the future. For environmental reasons, materials that are recyclable should always be used if possible and sandwich solutions, in which a number of materials are combined, should be avoided, as these prevent rational sorting when they have to be replaced at some point in the future.

Plan space carefully
As mentioned above, it is important to ensure that the installations have enough space, both for the original solution and possible future changes. Careful space planning should aim at ensuring that the plant and equipment can be:

- Easily moved within the building
- Easily installed – are shafts and other spaces sufficiently large so that joining and insulation work can be carried out correctly?
- Tested
- Commissioned
- Maintained
- Repaired
- Removed from the building when they are no longer needed

Ducts, both rectangular and circular, require a lot of space when compared to other installations such as cables and pipes. They also require space for handling and are difficult to manoeuvre, especially when there is a collision risk with other installations. To prevent collisions, a common problem when different services are installed in corridor ceilings, it is important that the project engineers and contractors carefully study potential congestion points at an early stage and deal with the questions concerning what should be installed where and in what order.

Drawing up cross-sections of difficult areas, so that everyone knows what space is available, is a practical way of tackling situations like these and will save both time and money. This might be regarded as a rather laborious task but it can pay big dividends. It will speed up the installation process and reduce the number of disagreements between the different contractors on site. It can also be referred to if one of the contractors uses space outside the agreed limits or installs pieces of equipment that make it difficult for other contractors to carry out their work. The person in the wrong will be obliged to make amends!

In the sub-section *Rectangular ducts* above, it was shown how duct sections could be joined using cleats. A relatively large amount of space is normally required for fitting the cleats onto flanges, between 20 mm and 40 mm high on every side of the two duct ends. This must be taken into consideration when choosing type of duct, dimensions and suitable shafts. An inexperienced design engineer might assume that the logical solution for ductwork in rectangular shafts is rectangular ductwork, where circular ductwork might, in fact, be easier to install.

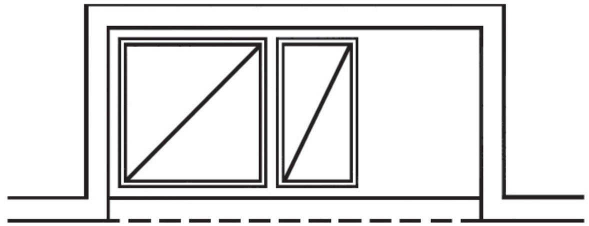

FIGURE 7. So far so good – it then becomes difficult to utilize the rest of the space in the shaft!

The space needed to assemble circular ducts is often less than for rectangular ducts designed to accommodate the same pressure drops. When ducts are installed in the ceiling of an office corridor or in a duct shaft, where the ducts, in both cases, are only accessible from one side, this often causes major problems, as the flanges on the far side are difficult to reach.

Even here, the cost of the finished ductwork might be lower if the circular alternative were used and, at the same time, this might facilitate balancing and zoning.

When the critical areas of a building are designed in detail, it is important to consider the installation methods that are going to be used. As an example, we can look at the installation of ducts to be wrapped in insulation. When the duct itself has been installed – which requires space for joining and suspension – the insulator has to start work. Free space will be needed around the duct for positioning and attaching the insulation material, and possibly a vapour barrier as well. If the ducts have been placed too close to the ceiling, walls or other installations, it will not be possible to carry out this work to the required standards.

Access routes and installation space must be designed so that workers' protection and safety are taken into consideration. Maintenance personnel often have to carry heavy tools and very large spare parts, for example, replacement filters. They must be able to carry out their work in a safe and ergonomic way. Fixed ladders are difficult to negotiate if equipment has to be carried at the same time, as both hands are needed for climbing.

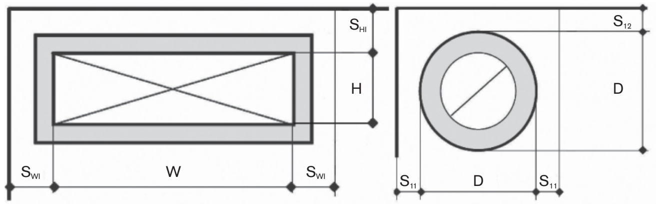

FIGURE 8. Free-space requirements for external insulation. Distances are shown in Table 3.

TABLE 2. Spaces required for 100 mm of external insulation.

Duct size, mm		**Rectangular ducts**		**Circular ducts**	
Circular D (incl. 100 mm insulation)	Rectangular W or H	S_{W1} mm	S_{H1} mm	S_{11} mm	S_{12} mm
≤ 160				≥ 100	≥ 50
$> 160 \leq 300$				≥ 200	≥ 100
$> 300 \leq 500$				≥ 300	≥ 100
$> 500 \leq 800$				≥ 400	≥ 100
> 800				≥ 500	≥ 150
	W, H ≤ 700	≥ 400	≥ 400		
	$700 <$ W, H ≤ 1200	≥ 600	≥ 400		
	W, H > 1200	≥ 600	≥ 600		

Can the duct shaft be accessed so that changes and additions can be made? Can lifting gear be used to lift heavy equipment, such as plant units and fans? Are access routes, doors and service shafts sufficiently wide and high for moving equipment etc?

Will the installations be pleasing to the eye?

A building is a system and, if it is to function as intended, the building services installations must also function properly. There is a popular trend among architects to allow installations to be part of the design of the building. This means that ductwork is not hidden above suspended ceilings or behind screens but is completely visible, forming part of the interior design.

FIGURE 9. Ventilation ducts in two of the atriums at Ramböll's Gothenburg office.

FIGURE 10. Ventilation ducts in a Stockholm restaurant.

Figures 9 and 10 above show examples from buildings where the architect has chosen to put the ventilation ducts on view. In Figure 9, the ductwork in the building's four atriums has been painted in different colours to improve orientation within the building. Even on the different floors the circular ducts are painted and on view and not hidden above suspended ceilings, as is normally the case in offices buildings. In Figure 10, it can be seen how ducts have been painted and on view, in different colours as part of the interior design. The restaurant is situated in a large office building.

This requires close cooperation between the architect and the HVAC engineer and their work must start at an early stage in the planning process if both parties are to benefit.

Figure 11 shows how different ducting systems can be colour-coded to facilitate recognition.

FIGURE 11. Colour-coded ducting.

If room heights are not limited by installing suspended ceilings, this could mean lower building costs. At the same time, the greater free height of the room and its larger ventilated volume will normally contribute to an improved indoor climate. This is because the extra space is created high up in the room where the concentration of pollutants is normally higher and this means that the ventilation air can be used more effectively. Having direct contact between the ventilation air and the ceiling makes it possible to use cool outdoor air to cool the building. This does not mean that suspended ceilings are not necessary, but they should be used primarily for acoustic reasons.

If installations are planned to be visible, they must be of very high standard, professionally designed and installed, and live up to expectations, otherwise they should not be chosen.

REQUIREMENTS, TESTS AND INSPECTIONS

Requirements must be expressed in verifiable terms

Requirements concerning duct systems are normally specified in tender and contract documents. The findings in EU projects [Andersson et al., 1999] and [Andersson et al., 2002] showed that differences when specifying requirements and checking them varied greatly in different countries, and this resulted in great variations in quality. For example, airtightness requirements for ventilation ducts in Sweden were considerably stricter than in two other countries that were investigated.

A probable explanation was that the Swedish AMA (General Material and Workmanship Specifications) system had been in use since 1950 and requirements concerning air handling systems had been expressed in verifiable terms. For example, requirements for airtightness in duct systems had been included in the Swedish 'HVAC' AMA since 1968. See the sub-section *Testing a duct system for airtightness* below.

Airtightness requirements for duct systems

Different studies have shown that air leakage from duct systems can result in a greatly increased use of energy. There are two reasons for this:

1. Fans have to work harder.
   The air flow through a fan is directly affected by any air leakages from the ductwork. If the design air flow is to reach the terminal devices, the fan must be dimensioned for and provide an air flow that is the sum of the nominal flow, i.e. the combined air flows to the terminal devices, and the leakage that occurs on the way to and from them.

As the power required by the fan is proportional to the cube of the air flow, Equation (2), the power requirement will increase by:

$$\Delta P = \left(\frac{q_{tot}}{q_{tot} - q_{leak}}\right)^3 \text{ kW} \qquad (3)$$

where:
q_{tot} is the total air flow through the fan in m^3/s
q_{leak} is the air leakage in the ductwork in m^3/s

This means that a 5% air leakage will require a fan power increase of nearly 20%.

$$\Delta P = \left(\frac{1}{1-0.05}\right)^3 \approx 1.20$$

2. Increased thermal losses.
   Treated supply air – heated or cooled – that leaks on its way to a room will also lose the energy that was used for treating it.
   If the supply air and extract air ducts are placed in the same suspended ceiling space, air that leaks out of the supply air ducts will be sucked into the extract air duct, without having first passed through any of the rooms.
   Leaking ducts are detrimental to ventilation systems as they have a negative effect on energy efficiency, thermal comfort and air quality.

Upper limits for allowable air leakage are therefore required to:
- Minimize costs and energy losses that would otherwise be a result in an over-dimensioned and inefficient plant.
- Simplify commissioning.
- Minimize noise occurring at leakage points.
- Limit infiltration from and leakage into spaces that do not require air treatment.

A duct system will never be absolutely airtight and such a requirement cannot be stipulated. Instead, limits are stipulated regarding how much air is allowed to leak at a given pressure, normalized to the duct system's total surface area.

Air leakage classes for duct systems

The following classification, which came into force in Sweden in connection with the revised 'HVAC' AMA in 1972, is also used as a basis for classifications according to Eurovent, the European air handling and refrigeration equipment manufacturer's association. The classification stipulates the maximum leakage per m^2 duct surface area expressed as a leakage factor K:

$$K = \frac{q_v}{A \cdot \Delta p_{ref}^{\,0,65}} \qquad (4)$$

where:

q_v is the leakage flow in m^3/s

A is the surface area of the tested duct in m^2

Δp_{ref} is the reference pressure used during testing in Pa

TABLE 3. Airtightness classes and leakage flows according to Eurovent 2/2 ('HVAC' AMA) and ASHRAE.

Airtightness class according to Eurovent 2/2 $l/(s \cdot m^2 \cdot Pa^{0.65})$	**Leakage flow at 100 Pa l/s per m^2**	**Leakage flow at 400 Pa l/s per m^2**	**Leakage classes acc. to ASHRAE $ml/(s \cdot m^2 \cdot Pa^{0.65})$**
Class A $(K < K_A = 0.027)$	0.54	**1.33**	27.0
Class B $(K < K_B = 0.009)$	0.18	**0.44**	9.0
Class C $(K < K_C = 0.003)$	0.06	**0.15**	3.0
Class D $(K < K_D = 0.001)$	0.02	**0.05**	1.0

Although Airtightness Class D is not defined in Eurovent 2/2, it is used in a number of European countries, including Sweden. A test pressure of 400 Pa is used in Sweden ('HVAC' AMA 98).

Testing a duct system for airtightness

The reasons for having airtight ducts have been discussed in the sub-section *Airtightness requirements for duct systems* above. This means that the client must express requirements in verifiable terms – terms which the contractor can understand, accept and price – and carry out random checks to verify the airtightness in connection with the final building inspection and when taking possession of the building.

An EU project [Andersson et al., 1999] showed that this was not often the case. When no requirements had been specified and no testing had been carried out, it could be seen that the quality of the installed

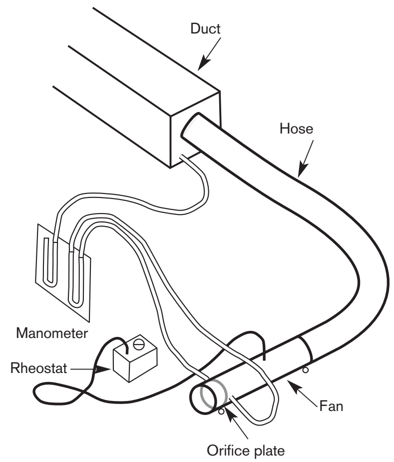

FIGURE 12. Typical measuring equipment for testing airtightness in ducts.

duct systems was very low. This is relatively self-evident – if quality is not demanded and checks not made, quality cannot be provided.

When comparing the measured leakages in duct systems in Belgium, France and Sweden, it was shown that duct systems in Sweden were, on average, 25 to 50 times more airtight than corresponding systems in the other two countries. The reason for this lies in the fact that, as pointed out earlier, airtightness in ducts has been stipulated in contract documents according to the 'HVAC' AMA since 1968, and that these demands have gradually become stricter as different technologies have developed and, in turn, given rise to even higher demands.

The airtightness classifications in the latest version of the 'HVAC' AMA, published in 1998, are as follows:

- *Airtightness class A,* the lowest requirement level, which defines requirements for visibly installed ducts in the space being served. A leakage here will not have any real significance, as any extra air flows, besides those through the supply and extract air terminal devices, will be beneficial to the space.
- *Airtightness class B* (three times tighter than A), which defines requirements for rectangular ducts and circular ducts in duct systems with surface areas ≤20 m^2.

- *Airtightness class C* (three times tighter than B), which defines requirements for circular duct systems with surface areas >20 m^2.
- *Airtightness class D* (three times tighter than C) is not a standard classification in the 'HVAC' AMA 98, but can be specified for systems in which airtightness is essential. Circular duct systems that comply with these requirements are now available.

Airtightness requirements are verified by random testing and the results are reported on special forms as part of the contractor's assignment.

The number of the random tests varies depending on type of ducts used: 10% of the circular ductwork in a contract and 20% of the rectangular ductwork are standard proportions according to the 'HVAC' AMA. The normal testing pressure is 400 Pa.

If it is shown that these tested sections meet the airtightness requirements, then the ductwork will be approved. If it is shown that a tested duct leaks more than allowed, depending on its class, it will have to be sealed and retested together with another 10% of the circular ducts and 20% of the rectangular ducts. If retesting shows that the ducts meet the requirements, then the ductwork will be approved. If it is shown that the retested ducts leak more than their classification allows, the ducts must be sealed and the tests extended to include all ducts included in the contract.

These requirements mean that it will be very expensive for contractors, if they are forced to carry out more than the nominal testing included in their contracts and that has been included in their costs. They will have to pay for all further testing – something that ought to be an incentive to make sure that work is carried out correctly from the start.

A three-country comparison of airtightness in ducts

In the EU SAVE-DUCT project [Andersson et al., 1999], carried out in Belgium, France and Sweden, the airtightness of a large number of duct systems was investigated. It was shown that the leakages in the Belgian and French systems were, on average, three times higher than those allowed in Sweden for Class A ducts. The corresponding average leakages in Sweden were between Class B and Class C, see Figure 13.

Balancing air flows

It is important that the air flows to different rooms in a building are measured and adjusted to comply with stipulated levels. This means that

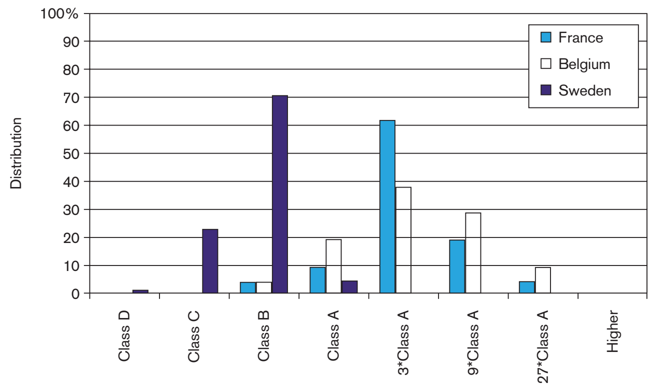

FIGURE 13. Distribution of the different airtightness classes based on measurements carried out in 21 systems in Belgium, 21 Systems in France and 69 systems in Sweden. The bars represent the relative number of systems fulfilling the requirements for the specified airtightness class.

a duct system must be planned and installed in such a way that this work can be carried out with acceptable accuracy and at the lowest possible costs.

The shorter the distance that the air has to be transported between the fan and the terminal devices, the less energy will be required and the simpler the balancing work will be. The balancing work will also be simplified, and the energy use less, if the system is designed so that the air flows to the terminal devices are supplied via symmetrically branched distribution networks rather than via terminal devices placed in series.

This means that ducts supplying several terminal devices should be divided into branch ducts and stub ducts. If several terminal devices are connected to one and the same duct, then the distance between the first and the last terminal device must be made as short as possible, to avoid excessive damping and noise from the devices.

Placing the devices symmetrically is something that will greatly simplify and shorten the time required for balancing. The ductwork between the main duct and the terminal devices will be built up in the same way, using the same types of duct components and duct lengths, which means that the air pressure at each terminal device will be the same. See Chapter 29/Balancing ventilation systems.

Why clean them?
There are three main reasons why ducts should be kept clean:

1. Pollutants can restrict air flows to such an extent that the ducts cease to function properly; pressure drops will rise and air flows decrease.
2. The inside walls of the ducts can become coated with a layer of combustible pollutants; these might ignite and cause fires or explosions.
3. Pollutants that are irritants, or that are otherwise dangerous, can collect in the ducts and cause health damage if introduced into occupied rooms.

The first reason is especially applicable to extract air ducts from wet rooms, where small dimension ductwork can easily become blocked. Extract air is warm and humid, and vapour will condense on the inside of the cold metal ductwork. Fibre particles in the extract air, from washing hung to dry and from wet towels, will then stick to this wet surface. Extract air terminal devices are normally connected to 80 mm ducts and these cannot cope with any large build-ups of pollutants before the air flows become insufficient. As these fibre particles collect along the first 50 cm of a duct, they are easy to remove.

The second reason concerns ducts used to remove inflammable or explosive pollutants, i.e. ducts that must be cleaned regularly according to the requirements stipulated in national fire prevention regulations. There are numerous uses of ducts where these regulations are applicable: in extraction systems for paint spraying booths, bakery ovens, kitchen ranges and deep-frying units where the primary objective is to prevent pollutants from entering the ducts, for example, by using paint and fat filters.

When designing and installing ducts like these, it is important to follow the national fire regulations very carefully. Plans must be drawn up which describe how cleaning is to be carried out, where the inspection and cleaning hatches are to be placed and whether the ducts are to be fitted with internal cleaning devices. Care must be taken to choose the correct duct materials, thicknesses and types of insulation, and to stipulate safe distances to combustible parts of the building.

The third reason must be explained in more detail. It is the most recent of the three reasons and has been discussed for the last two decades as a way of reducing the risk of SBS, Sick Building Syndrome, and improving the quality of the indoor air.

It goes without saying that pollutants from soiled ducts must not be allowed to foul the supply air. If there is a risk of this happening, it must be eliminated by cleaning the ducts. These risks can occur in supply air ducts if the filters are of inferior quality, if they are overloaded or if they are installed so that unfiltered air can bypass the filter.

There is a risk that ducts can become breeding grounds for mould and fungi. The best way to prevent this is to:

- Avoid letting the ducts become wet by:
  - using the right types and designs of air intake grilles
  - limiting the speed of the air through the intake to a maximum of 2.5 m/s.

  Both these measure will reduce the risk of the supply air drawing rain drops and snow flakes into the ventilation system, see Chapter 18/Outdoor air intakes – location, design, inspection and cleaning.
- Locate the outdoor air intake where the air is cleanest, see sub-section *Locations for fans and air handling plant* above.
- Refrain from insulating the ductwork on the inside.
- Inspect the intake ducts regularly to check whether they need to be cleaned. Inspection and cleaning hatches will be needed.
- Design the intake duct so that any moisture can run off to a drainage point.

Cleanliness checks, and cleaning when necessary, are carried out regularly in many countries, even if they are not stipulated in official regulations. It is therefore a good idea to mark out the size and position of inspection and cleaning hatches on the relevant drawings.

Which ducts need cleaning?
If the ducts have to be cleaned for reasons of health and comfort, then it should be possible to limit cleaning to the supply air ducts, at least in countries where, for reasons of hygiene, return air is not used.

If return air is used, it must be ensured that the extract air does not carry tobacco smoke or undesirable smells to the air handling unit and thereby impair the quality of the supply air. If return air is used, then the extract air and return air ducts should, of course, be inspected and, if necessary, cleaned.

The reasons for cleaning ducts, as shown above, apply to all pollutants introduced into the ducts after installation. However, it is not always necessary to clean all the duct systems in a building, if only one of the systems is shown to require cleaning. Table 5 summarizes the most common and important reasons for cleaning different types of duct systems.

TABLE 5. Which ducts should be cleaned and why?

Which ducts should be cleaned?	Why should they be cleaned?		
	Function	**Fire risk**	**Health**
Extract air ducts in dwellings, offices and schools	x	–	–
Return air ducts in dwellings, offices and schools	x	–	x
Supply air ducts in dwellings, offices and schools	–	–	x
Supply air ducts in dwellings, offices and schools with return air	x	–	x
Extract air and special extraction ducts in industrial premises	x	–	–
Cleaning required by law, because of fire risks	x	x	–

x normally required. – seldom required.

Make sure that all ducts are clean from the start!
It is important to keep ventilation ducts clean at every stage in the building process, i.e. while transporting them to the building site, during the installation work and until the system is put into operation. It is now becoming increasingly more common to protect the open ends of the ducts with tight-fitting covers of plastic or cardboard.

When protective solutions are needed, it is important that the requirements are stipulated in the contract documents, for example, according to one of the following alternatives:

TABLE 6. Protection levels and end cover requirements.

Protection level	After manufacture	During transportation	While storing on site	During installation
0	No	No	No	Yes, but only vertical ducts*
1	No	No	Yes	Yes
2	Yes	Yes	Yes	Yes

* In winter, vertical ducts can act as flues, allowing thermal forces to draw polluted air up through the building.

If ducts are not protected against interior soiling, the system must be cleaned before being put into operation for the first time. Checking duct cleanliness should be a natural part of the final building inspection.

When is cleaning needed?
The need for cleaning is normally determined by visual inspection. This can be done by using TV inspection equipment, or manually and visually by using torches and mirrors inserted into the ducts via the inspection hatches, which means that these should be situated relatively close together.

When TV inspections are carried out, a small camera is mounted on a remotely controlled robot that can work its way through the duct system. The camera then transmits a signal to a monitor and video recorder. The distance that the camera has travelled is shown on the screen to pinpoint where any special steps need be taken. As this type of equipment is relatively expensive and requires skilled operators, the work is normally carried out by a specialist contractor.

Duct cleaning methods
The methods used are:

- Dry cleaning
- Wet cleaning
- Disinfection
- Encapsulation
- Removal of insulation linings

Dry cleaning is used when the pollutants can be removed using simple mechanical tools or when the use of water is not appropriate.

Manual cleaning and swabbing is used when it is easy to access the insides of the ducts or when these are so large that the cleaner can get inside them. If this is possible, it is extremely important that the strength of the suspension system is checked first. Supports must be designed so that they can carry the extra weight of the cleaner as well as tools and equipment. For manual cleaning, from both the outside and the inside, sufficiently large cleaning hatches or manholes will be required.

Small ducts can be cleaned using tools fitted with rotary brushes and special mouthpieces for cleaning agents. Chemicals can then be used to kill off organisms or limit their growth.

This method normally entails isolating a section of the duct system and subjecting it to an under-pressure by attaching a vacuum cleaner to one end of the section. The vacuum cleaner used to create the under-pressure and collect the pollutants must be fitted with a HEPA filter. The cleaning process starts at the far end of the section and progresses towards the extraction end. Different types of visual monitoring are used to check the results.

If the ducts are lined with insulation, then encapsulation, i.e. internal duct sealing or re-lining, can be used to prevent the erosion of fibres from the insulation material and thereby contain the organic substances that could form a breeding ground for microorganism growth. However, for best results, it is often best to remove the insulation material, if this is at all possible.

Occupants in a building must be protected during the cleaning work by completely isolating the duct sections that are being cleaned from the air handling equipment and from the rest of the building.

The use of decontaminants and chemicals when encapsulating is more difficult. The substances must be approved for use and the cleaners must be dressed in suitable protective clothing and equipped with breathing apparatus or, in less dangerous cases, with protective face masks.

The effectiveness of different cleaning methods is not very well documented. The methods used to judge the results vary – from scraping samples from the cleaned surfaces to the use of agar plates to check for microbial presence – and the results are not mutually comparable.

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20. AIR FILTERS AND AIR FILTRATION

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Indoor climate solutions have progressed from natural ventilation to airborne heating. But what actually happens when we use natural ventilation and why can't this principle be used in modern buildings? Building techniques have changed: draughty floors and gaps around windows and doors belong to the past. What has not changed, however, is that buildings must still be subject to under-pressures to prevent warm, moist indoor air from leaking into cold walls and condensing. Natural ventilation, which is based on dynamic pressure differences, was helped along by the heat from the flue adjacent to the ventilation duct in the chimney. There were seldom any filters in the outdoor air intakes, only fly nets. Air change rates were small and little thought was given to what the incoming air might contain in the way of pollutants.

Natural ventilation has been replaced, to a great extent, by mechanical ventilation, as flues are no longer used in this age of district heating and heat pumps. Fans used today also create much greater pressure rises than natural ventilation, which has meant that simple filters can now be used in outdoor air intakes. These coarse filters can remove a lot of the pollutants, including natural pollutants, i.e. particles down to about 5 to 10 μm in size (1 μm = 1/1000th of a mm). However, particles that are even smaller can also occur naturally and these, together with particles created by human activities – exhaust gases from vehicle and aeroplane engines and fumes from district heating plants, wood burning, waste combustion etc – will pass straight through a coarse filter.

Naturally occurring particles and pollutants come from soil erosion, forest fires and volcanic eruptions or comprise salt grains from the sea, and pollens etc. A volcanic eruption will often create a really beautiful sunset as the pollutants and particles are lit up by the sun, colouring the sky red.

Our own built-in air filters, the mucous membrane and hairs in our nostrils, can cope with particles down to about 1 μm, while anything smaller will be drawn into our lungs. When particles are smaller than 0.1 μm they can even penetrate lung tissue and reach the circulatory system, and they are suspected causes of heart diseases and vascular disorders [Geiser 2005, Nemmar et al., 2002]. These extremely small particles could, for example, comprise uncombusted hydrocarbons, a by-product of all imperfect combustion processes.

Most of us spend up to 90% of our time indoors and it is important that polluted indoor air be removed and replaced by filtered outdoor air. Natural ventilation systems could only create low rates of air change, especially in summer when the temperature differences between the indoor air and the outdoor air were small or non-existent, and did not cause any noticeable transportation of pollutants. Particles and gases created indoors and emissions from synthetic materials used to have less effect on people than they do now. Today, in most countries, there are laws and regulations governing the air change rates in modern buildings. This means that air flows are greater than in natural ventilation systems but, at the same time, larger quantities of pollutants are now being introduced into the indoor environment from outdoors. This means, in turn, that more attention is now paid to the need for efficient filtering of small particles, which are generally regarded as being dangerous to health.

Originally installed to protect parts of the ventilation system, filters are now optimized to provide special functions, for example, to protect people's health or sensitive manufacturing processes and clean room activities.

In Sweden and most of Europe, the testing methods defined in EN 779:2002 are used to test filters. The standard divides coarse filters into four groups, G1 to G4, by measuring the percentage arrestance, or collecting efficiency, based on weight measurements of the concentrations of standardized test dust before and after filtering. To determine the filter class of a fine filter, F5 to F9, the filtering efficiency is based on measurements of the concentrations of particles, for example, in particles per cubic foot of air, of particles 0.4 μm in size. Measurements are carried out in a test cycle in which the filter is also subject to the standardized test dust to simulate the larger particles in the air, which, over time, cause the pressure drop across the filter.

Coarse filters are seldom used now, and F5 and F6 filters do not differ significantly from G4 filters.

As shown in Table 1, the average efficiency of 0.4 µm particles for an F7 filter lies between 80 and 90%, which makes this filter suitable for use as a supply air filter.

Table 1. Filter classes according to EN 779:2002. Tests continue until the final pressure drop has been reached. In practice, the final pressure drop is considerably lower.

Filter class: EN 779:2002	Average arrestance by weight %	Average particle arrestance efficiency %*	Final pressure drop Pa
Coarse filter			
G1	< 65		250
G2	≥ 65 – < 80		250
G3	≥ 85 – < 90		250
G4	≥ 90		250
Fine filter			
F5		≥ 40 – < 60	450
F6		≥ 60 – < 80	450
F7		≥ 80 – < 90	450
F8		≥ 90 – < 95	450
F9		≥ 95	450

* proportion of 0.4 µm particles.

If we want to remove even smaller particles, then F8 and F9 filters should be chosen. These filters not only provide better efficiency results for the 0.4 µm test size particles, retention of even smaller particles is also better than for F7 filters. The fibres making up the filter media are even narrower, which makes it possible to stop and retain very small particles.

The problem is that it is not possible to make functional demands with regard to the highest acceptable concentration of particles, neither in the supply air nor in the room air. The filters are classed by their percentage arrestance efficiency, while the quality of the filtered air is related to the composition of the outdoor air, and there is a great difference between town air and country air.

Should we protect ourselves to unreasonable lengths? We can breathe outdoor air, so why can't we breathe the same air indoors when it is supplied by the ventilation system?

We even think that 'fresh air' makes us feel good when we go outdoors. This is sometimes because the air is colder but, above all, because we are generally more active outdoors.

Compare this to when we come indoors, slightly frozen after a nice long walk on a cold winter's day. It's pleasant feeling the heat and we perhaps don't pay so much attention to the quality of the air being breathed.

The situation in Sweden, however, is much better than in many other countries in the world where face masks are sometimes needed in large towns and inhabitants might have to buy a few lungfuls of oxygen in a local kiosk to be able to stay outdoors at all. Demands regarding exhaust emission control are lower than ours, but the exhaust gases, on the other hand, are visible and people do realize that they have to protect themselves.

The world is continually striving to increase the efficiency of combustion processes by using atomized fuels under high pressure, though these actually result in even more finely sized exhaust particles. For instance, a modern injection system for a car engine will atomize fuel to a particle size of about 0.3 µm. Although combustion products like these are invisible – particles smaller than 10 µm cannot be seen by the naked eye – our bodies and our lungs will react to them when we spend time outdoors in heavily populated areas.

A highly efficient filtration process can, therefore, reduce health risks and let a person feel much better indoors than outdoors.

F7 filters are now used without any special afterthought, but might it not be advisable to consider changing to F9 filters? Their filtration efficiency is better, the degree of pollution indoors will become lower and we will most probably feel a lot better. Whatever the reason, we will subject ourselves to fewer health risks caused by outdoor pollutants. If the market were to make stronger demands for better filters, it would certainly stimulate their development.

A higher class of filter will, however, require more stringent demands to be made on the filter mountings, as leakages and gaps between the filter and mountings will adversely affect its efficiency. Conventional outdoor air intake grilles should also be replaced by weather-protected intakes. This will prevent water from reaching the supply air filters, which cannot function optimally if they are repeatedly subject to moisture. In some instances it might even be advisable to use a pre-filter in front of the F9 filter. Some people say that this is not viable in modern ventilation systems with the latest energy demands, but should we just leave it at that? And what about future systems and filters – what are they going to have in store?

Figure 1. Pollen and combustion products trapped in an air filter.
© Lennart Nilsson Photography AB/Camfil AB

Future developments will, of course, also include new filter testing methods. Advanced testing methods will then quickly eliminate poorly performing filters and filters that are only designed to pass laboratory tests.

DIFFERENT FILTER MEDIA

Three different materials are used when manufacturing fine filters:

- Rough, carded and electrostatically charged electret fibres
- Melt-blown polymer fibre
- Micro-fibreglass

These different materials behave in one way in a laboratory and in a completely different way when they are subject to outdoor air and outdoor pollutants.

Rough, carded electret fibres must be statically charged if they are to stop small particles. They can remain charged for a day in a testing laboratory where the air is clean, dry and kept at an even temperature, but the charge soon diminishes and eventually disappears completely when the filter is mounted in a ventilation unit and subject to pollutants and moisture in the outdoor air. This greatly reduces the functional lifetime of this type of filter.

The arrestance efficiency of filters with melt-blown fibres is also partially dependent on the so-called electret effect but also on the van der Waals forces between small particles.

The function of micro-fibreglass does not depend on static charges and this material can be used to manufacture the thinnest of fibres, which provide the best arrestance efficiency. The narrower the fibres, the smaller the particles that can be caught.

FILTER CLASSIFICATION AND LABORATORY TESTING

Filters are tested and classified in Sweden and Europe according to EN 779:2002. This test standard stipulates what test equipment is to be used, how the tests are to be carried out, how filters are classified and what must be included in the test report.

During testing, filters are subject to a standardized test dust to measure the particle arrestance performance properties and to an aerosol to measure the filtration efficiency. The arrestance capacity of a filter is the weight of test dust in grams that the filter collects before its final pressure drop, 450 Pa for fine filters, is reached. Repeated tests are carried out and these take about a day to complete, and they all take place in ideal laboratory conditions. The air that is used has a constant humidity and temperature, and the rate at which the dust is fed into the filter is very high.

Conditions like these are not found anywhere in practical applications and therefore a laboratory test can never predict how a filter will perform in practice. On the other hand, it is possible to compare different filters and these are classed according to the test results obtained. The fine filter classification is determined by the average value of the efficiency for 0.4 µm particles. This is the size of the majority of the combustion products derived from human activities and is therefore especially relevant. To be classed as an F7 filter, the efficiency must lie between 80 and 90% during the whole of the test cycle.

To emulate practical conditions, when testing according to EN 779:2002, the arrestance efficiency is determined when the filter medium is completely electrostatically discharged. The discharging of the filter is accomplished either by using diesel fumes or by dipping the filter in isopropanol, drying it for 24 hours. Its efficiency is then tested for 0.4 µm particles.

A new F7 filter has an initial efficiency of about 60%. When a rough, carded statically charged filter is discharged this rating for falls to about 10% while for melt-blown fibres the corresponding figure is about 40%. The reduction for fibreglass media is only marginal, as it is not charged from the beginning.

Information like this, about how the efficiency of a filter can be expected to change in practical applications, is very important, as no attention is paid to this fact when classifying a filter. This means that a filter can be classed as an F7 filter even if the discharge test shows that the filtering efficiency of the filter can be expected to fall in practical use.

This is especially important to note if the filter has not been quality assured, for example, according to the Swedish P-Marking system.

QUALITY ASSURANCE OF AIR FILTERS

P-Marking is a form of quality assurance certification issued by an independent testing laboratory, the SP Technical Research Institute of Sweden.

Air filters are P-marked according to special certification rules [SP 2000]. To obtain certification, the manufacturer must supply comprehensive details about the filter material and the design of the filter. The manufacturer must also ensure that production and quality controls have been carried out according to the ISO 9001:2000 Quality Management System or equivalent. The SP Technical Research Institute of Sweden carries out annual quality audits at manufacturer's production facilities.

Filters are tested in a laboratory according to EN 779:2002, but they are also subject to long-term testing in the field for six months and there is a minimum filtration efficiency requirement that has to be fulfilled. This long-term test is carried out so that the function of the filter can be guaranteed in practical applications.

EUROVENT CERTIFICATION

The Eurovent Certification Company, an independent institute in Europe, certifies performance ratings for air-conditioning and refrigeration products according to European and international standards. This means that manufacturers can verify the technical properties of their products stated in the product documentation. Certification is available for bag, compact and panel filters in the F5 to F9 fine filter range. Eurovent Certification chooses four different types of fine filters from a manufacturer's range for testing. The filters are blind-tested according to EN 779 at testing laboratories that have been accredited according to ISO/IEC 17025. If a discrepancy is discovered, new tests are carried out on a new filter. If the discrepancy is found again, the manufacturer is

obliged to change the technical specifications given in the product documentation within two months or redesign the product within six months. A third alternative is to withdraw the product from the market – if no steps are taken, the manufacturer will not be able to refer to its Eurovent Certification for any of its products.

On approval, the manufacturer will receive a certificate from Eurovent Certification and it will be allowed to use the Eurovent Certification logotype on the approved product and in marketing material. The tests are repeated every year when Eurovent Certification chooses four new filters from other product groups. These are subject to the same testing, and the same test parameters as in previous tests and the products must be approved if the manufacturer is to retain its certification.

Unlike the P-Marking certification process, Eurovent Certification only carries out tests according to EN 779, but without investigating the arrestance efficiency when the filter medium is discharged. This means that a filter's efficiency, in practical applications, cannot be determined.

FILTER CLASSES ACCORDING TO EN13779

EN 13779 is a European standard for air handling systems in non-residential buildings. The standard provides guidelines for choices of filter class, but without stipulating specific limits for the purity of the filtered air. EN 13779 has become the national standard in many countries and focuses on providing healthy and comfortable indoor climates throughout the year at reasonable installation and running costs.

Filter classes are no longer chosen arbitrarily: the standard specifies the filter classes that are required to achieve the desired indoor air quality, IAQ, in relation to the level of pollution indoors. It should be noted that to fulfil the highest requirements, gas filters for molecular filtering of gases and vapours are required.

Indoor air, IDA, is classified from IDA 1, the highest requirement level, to IDA 4, the lowest, see Table 2.

TABLE 2. Classification of indoor air quality according to EN 13779.

Category	Description	Typical range of CO_2 level above outdoor air level ppm	Typical range of outdoor air rate, non-smoking area m^3/h per person
IDA 1	High IAQ	≤ 400	> 54
IDA 2	Medium IAQ	400 to 600	36 to 54
IDA 3	Moderate IAQ	600 to 1000	22 to 36
IDA 4	Low IAQ	> 1000	< 22

If the table were interpreted based on what we regard as being a normal concentration of CO_2 and the size of normal air flows, most non-residential buildings in Sweden would be classed as IDA 1 or IDA 2.

Outdoor air is divided into five different levels, from ODA 1, in which the air is clean and only contains temporary pollutants such as pollen, to ODA 5, in which there are high concentrations of both gases and particles, see Table 3.

When determining the quality of the outdoor air on a future building site it is quite often possible to obtain data from municipal monitoring systems that measure the concentrations of the most important air pollutants.

TABLE 3. Typical concentrations of pollutants in outdoor air and the corresponding air quality classes according to EN 13779.

Descripion of air quality	Concentration levels*					Outdoor air quality
	CO_2 ppm	CO mg/m³	NO_2 µg/m³	SO_2 µg/m³	PM_{10} µg/m³	
Rural areas with no significant sources	350	< 1	5 to 35	< 5	< 20	ODA 1
Smaller towns	400	1 to 3	15 to 40	5 to 15	10 to 30	ODA 2/3
City centres	450	2 to 6	30 to 80	10 to 50	20 to 50	ODA 4/5

* Most European cities publish daily pollution reports on the Internet.

Figures from the table above can be used as reference levels in conjunction with data published by other organizations. For example, the World Health Organization, WHO, has set a target limit for the mean annual PM10 level at less than 40 µg/m³ and the daily mean level of 50 µg/m³ must not be exceeded for more than 36 days per year.

PM10 is a measure of air pollutants in the form of small particles that can affect the respiratory system and heart/blood vessels. A PM10 value is a measure of the quantity of particles that can pass through a size-selective intake that with 50 % efficiency removes particles with an aerodynamic diameter of 10 µm. PM stands for *particulate matter*, i.e. materia in the form of particles.

In many major cities, the outdoor air is usually classed as ODA 4 or ODA 5, which makes it difficult to reach these goals.

G4 and F6 to F9 particle filters, classified according to EN 779:2002, are normally chosen and for molecular (gaseous) pollutants, gas filters for this type of pollutant only, or combination filters for both particle and gas filtering, are chosen.

TABLE 4. Recommended filter classes according to EN 13779 based on the quality classes of the indoor and outdoor air.

Outdoor Air Quality	Indoor Air Quality (IAQ)			
	IDA 1 (High)	IDA 2 (Medium)	IDA 3 (Moderate)	IDA 4 (Low)
ODA 1 (pure air)	F9	F8	F7	F6
ODA 2 (dust)	F7/F9	F6/F8	F6/F7	G4/F6
ODA 3 (gases)	F7/F9	F8	F7	F6
ODA 4 (dust + gases)	F7/F9	F6/F8	F6/F7	G4/F6
ODA 5 (very high conc.)	F6/GF*/F9	F6/GF*/F9	F6/F7	G4/F6

* GF = Gas filter (carbon filter) and/or chemical filter.

It can be difficult to determine an IDA and an ODA class as their definitions are rather vague.

The present standard use of an F7 filter should, according to Table 4, be sufficient to ensure an IDA 3 in smaller towns where the pollutants are mostly comprised of gaseous substances (ODA 3). If the outdoor air primarily contains particulate substances (ODA 2 or ODA 4/5), the air should be prefiltered using an F6 filter and then filtered again using an F7 filter. To achieve the next highest level of indoor air quality, the final filter would have to be changed to an F8 or higher class filter. Note that there is only a marginal difference in filtering efficiency between a solution that uses only one filter, for example an F7 filter, and a solution in which an F6 prefilter is used followed by an F7 filter. In other words, the marginal effect of an F6 filter is, in practice, small.

The definitions given in the standard do not seem to have been fully discussed as, for example, the filtering requirements are the same in an ODA 2 or smaller town environment as in an ODA 4 or city centre environment.

What happens to a filter after installation?

REPLACING FILTERS

Filters have to be changed at regular intervals and there is an important basic rule: To avoid microbial growth in the dirt trapped by a filter, it should be replaced at least once a year in one-stage filter units and at least once every two years in two-stage filter units with a prefilter.

Bacteria and fungi can give rise to microbial growth, if the conditions are right. Basically, this requires the presence of microorganisms that are capable of survival and an environment with correct pH value and temperature. Most important, however, is the presence of moisture, if growth is to occur at all. Moisture can collect in a filter in different ways, for example, when rain falls directly onto the air intake grille or when the humidity of the air is sufficiently high. This can happen when the relative humidity temporarily exceeds 90% or if it remains above 80% for more than three days.

The risk of microorganisms entering the ventilation air in this way is very small but, to be on the safe side, intervals between changes should not exceed those given above.

In general, filters are replaced when the final pressure drop is reached or according to the service schedule. But the question is, when is it most economical to change a filter?

To put costs into the right perspective, it is important that we first look at what happens to the pressure drops in two different types of plant.

CONSTANT SPEED FANS

Far fewer than half of the ventilation systems installed today have fans that operate at constant speeds. Fans can have either belt drives or direct drives.

B type impellers with backward curved blades are the most common type of impeller. The power required for these fans is practically constant, even when the pressure drop across the filter gradually increases and the air flow rate falls.

When new filters are installed the plant will produce its greatest air flow, as the pressure drop across the filter is low. When the pressure drop increases, the flow will decrease.

This reduction in flow will also mean that the pressure drops across other components in the system will also be reduced. The operating point will move along the fan curve, which means that the total efficiency of the fan will be changed. In addition, it is important to determine how much of the total pressure drop occurs across the filter and the lowest acceptable air flow rate will then determine when the filter's final pressure drop has been reached.

VARIABLE SPEED FANS AND CONSTANT FLOWS

Most new ventilation systems have variable speed fans and constant air flows. The pressure drop across the filter increases as the filter becomes soiled and, to compensate for this increase in pressure drop, the speed of the fan is increased.

The pressure drop across the filter determines the energy required by the fan motor. As the constant air flow in the system creates a constant

pressure drop across the other components in the system, only the degree of soiling will affect the fan's change in speed during the lifetime of the filter. In the fan diagram, the operating point for the system moves along a constant flow line when the fan speed is changed.

The speed of the fan can increase as long as the motor has sufficient power. If the pressure drop across the filter continues to increase, the system will start to behave as a constant speed system – and the air flow will fall depending on the degree of soiling of the filter.

VARIABLE AIR FLOW SYSTEMS

In variable air flow systems, the flow rates are reduced during most of the operational time and the pressure drops across the filter are, consequently, considerably lower than the nominal value. Pressure drop readings might then be misleading, though these can be corrected for, if the corresponding air flows are known. When a fine filter is tested according to EN 779, the relationships between pressure drops and air flows can be clearly seen.

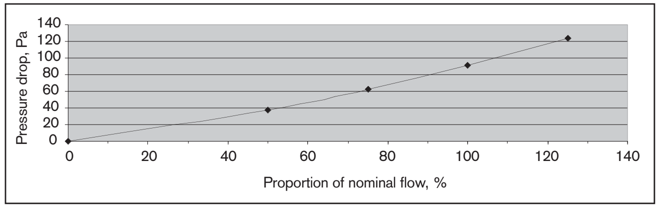

Figure 2. The diagram shows the relationship between the pressure drop across an F7 filter and the air flow through it when tested according to EN 779:2002. The nominal air flow is 3400 m^3/h.

SPRING OR AUTUMN FILTER CHANGES...?

As a rule, filters are changed in the spring and the autumn to approximately the same extent.

In plant with constant speed fans it is often advantageous to replace filters in the spring, after the heating season, to allow large air flows in the summer. This is also a good point in time if the system is fitted with comfort cooling devices with cooling batteries, so that the cooled air can be distributed more efficiently.

When filters are changed in the autumn, after the pollen season, pollen and other organic pollutants that have been trapped during the spring and summer are removed from the system. The risk of microbial growth in the filters is avoided and also the risk of having moist pollen breaking down and emitting allergens. These are so small that an H10 class filter (the micro filter class above F9) would be required to prevent them from being spread into the building.

...OR CHANGES WHEN IT IS CHEAPEST?

Most modern ventilation units are efficient to run, system losses are small and the fans, generally, have satisfactory total efficiencies. Although the cost of running a fan increases as the pressure drop across a filter increases, the filter costs will fall the longer it is in use.

It is therefore economical to replace filters at the point where the curve describing the fan's energy use intersects the curve for filter costs, as this is where the total costs will be the lowest.

Serious filter manufacturers focus on the most important task that the filter has to do, namely, to filter the air. They also strive to manufacture filters that create low initial and operating pressure drops in order to save energy.

Filter users are becoming more cost-conscious, carefully calculating when it is cheaper to put in a new filter rather than to continue using a soiled one with its high pressure drop and high energy demand.

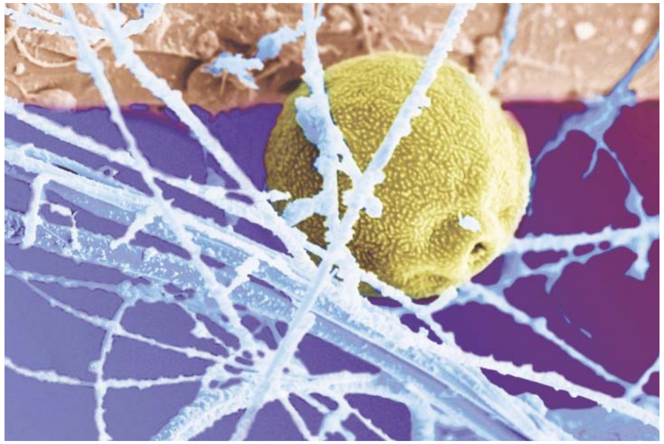

Figure 3. Pollen caught in an air filter.

LIFE CYCLE COST ANALYSIS

If the cost of heating supply air is ignored, the cost of running a fan is a major outlay. This can be shown by carrying out an LCC, Life Cycle Cost, analysis.

An analysis can be performed in two different ways: by using the test results from an EN 779:2002 test in which synthetic dust is used, or by carrying out a more realistic analysis in which the pressure drops are measured over a long period of time while the fan is in operation. Unfortunately, data from such long-term tests is rare.

When the first method is used, the tests are carried out to classify and compare filters. Test data includes details about the dust holding capacity of the filter and these laboratory values are then used to calculate the theoretical operational life of the filter.

If results from long-term testing are used, the LCC calculation will most probably correspond more closely to actual operational results.

An LCC calculation is a form of economic analysis in which all costs and incomes over the operational life of a system or product are taken into account and given a present value.

It is important to note that the analysis results shown in Figure 4 are only valid when variable speed fans are used in systems with constant air flows.

An LCC calculation does not take into account the most important function of the filter – the filtration efficiency, which means that a filter with high dust holding capacity, which often has a poor arrestance efficiency, is normally chosen.

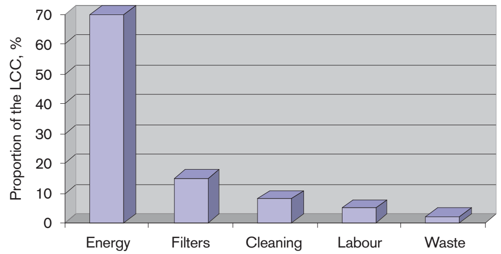

Figure 4. Results of an LCC analysis of an installation with a variable speed fan and constant air flow. The energy used to run the fan is the largest cost.

SAFETY WHEN CHANGING FILTERS

What sort of pollutants will be found in a filter when it has to be changed? Besides the particles created naturally, there will also be particles created by people. Fungi spores, bacteria, heavy metals, uncombusted hydrocarbons, salt crystals, ground dust and pollen – all these might be found in a used filter.

The most dangerous stage when handling filters is when a used filter, full of pollutants, is removed from a ventilation unit and before it is sealed inside an impermeable cardboard box or airtight plastic sack. Only then can the used filter can be handled without risk.

The individual particles that are released from a filter when it is being replaced cannot be seen, they are quite simply too small. It is important to bear in mind that these minute particles are always released to some extent during this step and it is therefore essential to make sure that the correct personal safety equipment is used. This means using:

1. Class FFP2 respiratory protection or higher (offers approximately the same degree of filtration as an F7 filter).
2. Gloves to protect against pollutants that could penetrate the skin or cause infection via a sore.
3. Disposable overalls.

This personal safety equipment must then be thrown away together with the soiled filter, classing both as combustible waste.

Last but not least: make a note of the date and the final pressure drop across a soiled filter before it is removed. These details could be important later on.

WHEN PARTICLES AND POLLUTANTS ARE VERY, VERY SMALL

Nano- is a prefix that means a billionth part and comes from the Greek word for dwarf. A nanometre, nm, is one billionth of a metre, one millionth of a millimetre, one thousandth of a micrometre, i.e. comparable with the size of a typical atom at around 0.1 nm in diameter.

Nanotechnology is sometimes referred to as atomic engineering, a field in which new methods and materials are created by modifications carried out at atomic or molecular levels.

Today, there are numerous examples of industrial applications in which the core technology is in fact nanotechnology. Nanoparticles are used in wear-resistant tyres, the manufacture of paints and cements, UV filters in sun lotions and – perhaps the oldest example – coloured glass.

Every day, enormous quantities of nanoparticles of titanium dioxide and magnetic nanoparticles, for use in hard disks and magnetic tapes, are produced. Surfaces with nanostructures that cannot be scratched or become dirty are now used for frying pans and even trousers. And windows are now available that do not have to be cleaned, as they are coated with a thin self-cleaning film of titanium dioxide. UV radiation from the sun in combination with the titanium dioxide breaks down any organic dirt that has collected on the surface of the glass.

Magnetic nanoparticles are used in surgery for special procedures and the extremely narrow needles used for eye surgery can be made thanks to this technology. Nanotechniques also used in medical diagnostics and to create new materials for implants and prosthetics. They are also used in advanced computer designs that are completely dependent on speed and therefore the number and size of the transistors.

At the same time as nanotechnology offers great opportunities, it can also lead to problems. Nano-sized particles have the same dimensions as biological molecules and therefore affect the human body in ways that no other substances can.

Authorities around the world have now begun to realize that nano-sized particles cannot be just released into nature without being subject to further investigation.

Discussions are now taking place regarding their possible effects on the environment and health. Better methods for measuring concentrations of nanoparticles will definitely be needed in the future.

The dangers inherent in nanoparticles depend more on how small they are rather than on what substances they are made of. Researchers will have to investigate whether different sizes of nanoparticles affect our health to different degrees.

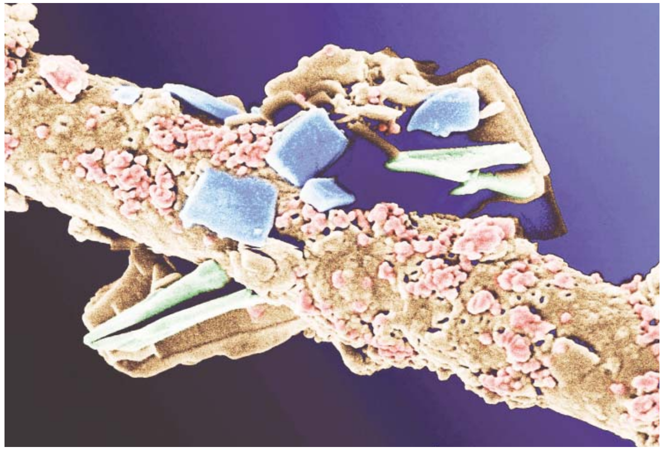

FIGURE 5. Nanoparticles on a strand of filter fibre.
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FILTERING NANOPARTICLES

The filtration efficiency of a filter is measured in a laboratory according to EN 779:2002. Tests are also carried out in the field and the results with regard to filtration efficiency follow the same patterns in both cases. The total filtration efficiency comprises of four filtration mechanisms: the sieve effect, the inertia effect, interception and diffusion. The combined effect of these mechanisms is such that particles in the 0.1 to 0.2 µm size range are the most difficult to catch. In other words, this is the size of particles that can most easily penetrate a filter medium. This is known as the MPPS, the Most Penetrating Particle Size.

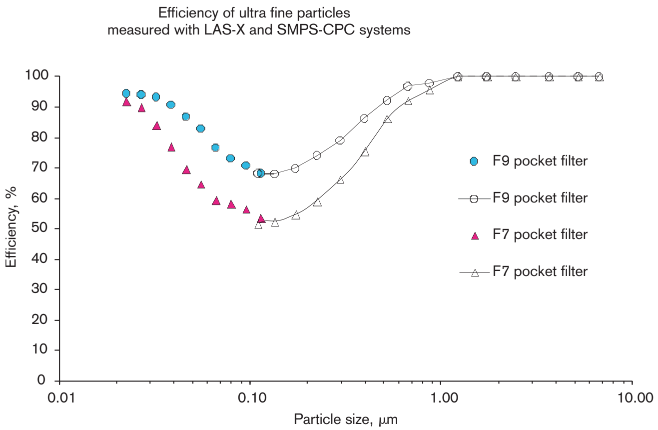

Figure 6. Filtration efficiencies for different classes of fine filters (Camfil Farr, Research and Development).

How good is an F7 filter at catching nanoparticles? Unfortunately, this is very difficult to measure, but equipment is available that can measure particles as small as about 20 nm or 0.02 µm. Examples of such measurements are shown in Figure 6. If we look at the filtering effect in the diagram, we can see that the curve dips at 0.1 to 0.2 µm (MPPS) to rise again for particles smaller than 0.1 µm. The diagram shows the filtration efficiencies for an F7 and an F9 filter. In fact, the curves are similarly shaped no matter what filter class is chosen: only the levels of the curves differ. Penetration can be reduced by about 25 to 30 % by choosing an F9 filter instead of an F7 filter.

There are many different types of nanoparticles and one area that particularly interests researchers concerns uncombusted hydrocarbons (PAHs, polycyclic aromatic hydrocarbons) because they occur abundantly and they are the result of different types of incomplete combustion in, for example, combustion engines and wood-fired furnaces. When normally polluted town air is analysed it is not unusual to find different types of PAH pollutants and a number of these are known to be carcinogenic.

Experiments and tests have been carried out to measure the efficiencies of P-marked F7 filters with regard to PM10 particles at the same time as the PAH content and filtration of ultra-fine and fine particles (20 to 1000 nm) have been measured and analysed. If the retention of PM10 particles in an F7 filter is about 85%, the retention of PAHs is about 70% and of particles from 20 to 1000 nm about 75%.

P-marked F7 filters and, to some extent, even F8 and F9 filters are now in common use and these provide good protection against health problems caused by pollutants, based on what we now know about fine and ultra-fine particles. However, it is important to point out that it is not a question of guaranteeing that we will be healthier and will feel better but rather a question of reducing potential risks.

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Buildings are ventilated for a number of reasons, though most importantly to maintain an air quality level that meets the occupants' requirements with regard to health and well-being. Other reasons include fulfilling requirements regarding temperature, humidity, dehumidification, etc or requirements based on specific usage, for example, for laboratories.

When a building is ventilated energy is used – partly to run the fans and partly to treat the air. Air treatment processes can include heating, cooling, humidification, dehumidification and filtering. And all these processes require energy. Normally, during a whole year of operation, very large amounts of energy are required to heat and cool the ventilation air. How much is needed is determined by the requirements stipulated for the indoor climate, the geographical location of the building and the design of the ventilation system and how it is run. This chapter focuses on the opportunities for recovering energy from the heating and cooling processes to which the supply air is subjected.

The plant used to heat and cool the ventilation air could be the largest end-user of energy in a building and much larger, for instance, than the energy used by the tenants or used to heat the building during the heating season (even in such cold climates as those encountered in Scandinavia). Consequently, there is every reason why the opportunities for energy-efficient operation of ventilation systems should be thoroughly investigated.

ENERGY FOR HEATING VENTILATION AIR

When determining how much energy is used to heat the air before it is distributed within a building, a methodology must be found that makes it possible to understand how energy use depends on system design and operational strategies.

The energy used to change the temperature of ventilation air depends on the mass flow $\dot{m}$ and the change in energy content Δh. This can be written as:

$$Q = \int \dot{m} \Delta h d\tau \text{ kWh} \qquad (1)$$

where:
$\dot{m}$ is the mass flow in kg/s
Δh is the change in enthalpy in kJ
τ is the time in h

If no moisture is added to or removed from the ventilation air, the change in enthalpy will only depend on the change in temperature of the air. If temperature data is available, for instance, as hourly averages, the calculation can be carried out as a summation over the whole year. The relationship can then be written as:

$$Q = \sum_{\tau} \dot{V} \rho c_p \Delta t \Delta \tau \text{ kWh} \qquad (2)$$

where:
$\dot{V}$ is the air flow volume in m^3/s
ρ is the density of the air in kg/m^3
c_p is the specific thermal capacity of the air in J/(kg K)
Δt is the temperature rise in K
τ is time in h

Depending on whether or not the air flow is constant during the year, the expression can be written as:

$$Q = K_1 \sum_{\tau} \Delta t \Delta \tau \qquad (3)$$

for constant air flow

$$Q = K_2 \sum_{\tau} \dot{V} \Delta t \Delta \tau \qquad (4)$$

for variable air flow

where:
$K_1 = \dot{V} \rho c_p$
$K_2 = \rho c_p$

The amount of heat used to change the temperature of the ventilation air is therefore proportional to the temperature change in constant flow systems and proportional to the temperature change *and* the flow change in variable flow systems. By summing up this data for the whole year, the annual energy demand can be obtained. An elegant and illustrative way of doing this is to plot the outdoor temperature data, normally comprising hourly readings, on a graph. To make the results manageable, they are also plotted according to magnitude in a duration diagram. Figure 1 shows a duration diagram for a particular location, based on the hourly outdoor temperatures that were originally arranged in chronological order.

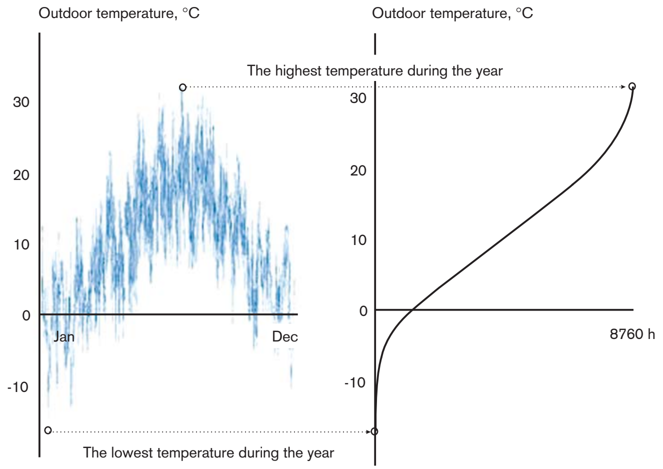

Figure 1. The duration diagram for the outdoor temperature, right, is created by arranging the different temperatures in order of magnitude. Hourly outdoor temperature data for given locations can be obtained from meteorological institutes or similar. There are 8760 hours in a year.

Irrespective of whether the outdoor temperature is plotted in chronological order or in ascending order, as a duration diagram, all the hourly outdoor temperature readings are included in both cases.

If a constant outdoor air flow is to be heated to room temperature, then the amount of energy required can be derived from a duration diagram, as shown in Figure 2. In this case, an indoor temperature of +22 °C has been chosen.

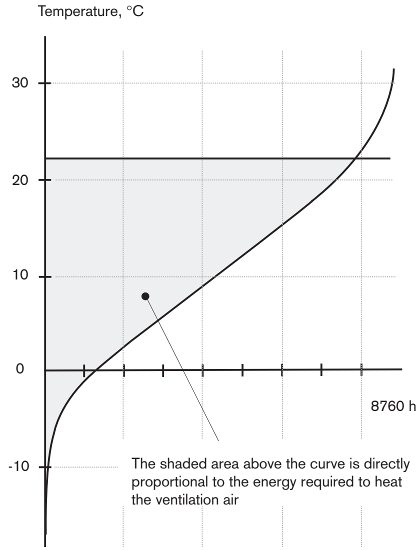

Figure 2. A graphical illustration of the energy required to heat ventilation air. The shaded area has the units *degree hours*. The data is for Berlin, Germany, where the mean annual temperature is 9.3 °C.

The area shown in the diagram, which is directly proportional to the heat energy required, is expressed in degree hours. To convert degree hours into energy, in kWh, a scale factor is used [Abel, Elmroth, 2006]. If the air flow varies during the year, this must be taken into account when calculating the number of degree hours. A method for doing this, using the outdoor temperature duration diagram, is given in [Nilsson, 1994]. For the sake of clarity, constant flow systems are assumed in the following discussion.

Every vertical distance in the diagram, the difference between the outdoor and indoor temperature, is proportional to a power requirement. The relationship between power requirement and temperature difference is given by:

$$\dot{Q} = \dot{V} \rho c_p \Delta t \qquad (5)$$

This means that for any given Δt, the heat power can be calculated if the air flow is known.

Depending on the design of the building in question and on how it is used, the energy needed to heat the ventilation air, as mentioned previously, can constitute a substantial proportion of the building's total heat energy demands. In well-insulated and airtight buildings, heating the ventilation air often constitutes the dominating heat requirement. This is why it is important to investigate whether it is possible to reduce the heating needs for ventilation air.

In supply and extract systems, the most direct way of reducing heating needs for the ventilation air is to pre-heat it by recovering heat from the extract air. Another way is to recirculate the extract air. Both methods are illustrated in Figure 3.

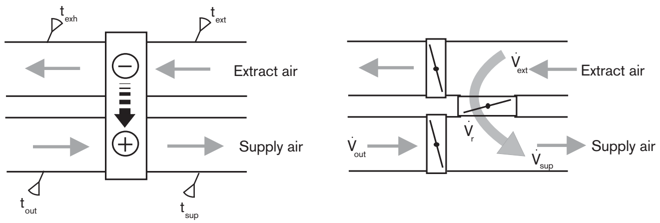

FIGURE 3. A heat recovery system and a return air system.

Nowadays, return air systems are unusual in Scandinavia and the trend in Europe is towards heat recovery using heat exchangers.

The potential for saving heat, by recovering it using heat exchangers, is usually expressed in terms of temperature efficiency. Temperature efficiency is defined as:

$$\eta_t = \frac{t_{sup} - t_{out}}{t_{ext} - t_{out}} \qquad (6)$$

If η is known, then the temperature of the air after passing through the heat exchanger can be calculated from the equation:

$$t_{sup} = t_{out} + \eta_t \, (t_{ext} - t_{out}) \qquad (7)$$

In practice, it is sometimes difficult to measure the temperature of the supply air directly after the heat recovery unit (there is often very little space in the air handling plant or it could be difficult to turn off the re-

heater that is often positioned directly after the heat recovery unit, etc). In cases like these, the temperature of the exhaust air can be used to determine η*t*. By creating a heat balance across the heat recovery unit, it is easy to show that t can also be defined as:

$$\eta_t = \frac{\dot{V}_{ext}}{\dot{V}_t} \left(\frac{t_{ext} - t_{exh}}{t_{ext} - t_{out}} \right) \qquad (8)$$

This means that the flow ratio $\frac{\dot{V}_{ext}}{\dot{V}_{sup}}$ must also be known, if the exhaust air temperature is to be used instead of the temperature after the heat recovery unit on the supply air side when calculating the temperature efficiency.

However, when return air is used, the term return air efficiency is used instead of temperature efficiency and is derived from the following relationship:

$$\kappa = \frac{\dot{V}_r}{\dot{V}_{sup}} = \frac{\dot{V}_{sup} - \dot{V}_{out}}{\dot{V}_{sup}} \qquad (9)$$

When a heat balance is established across the heat recovery unit, the following relationship holds true:

$$t_{sup} \approx t_{out} + \kappa \, (t_{ext} - t_{out}) \qquad (10)$$

The less the change in the absolute vapour content of the air during its change of state, the more correct the expression becomes. The relationship will still provide a good approximation of the return air efficiency, even if the difference in vapour content between the extract air and the outdoor air is in the region of 30 to 40 g/kg.

When the expressions for the temperature after heat recovery and the temperature of the return air are compared, there is a notable similarity. When the temperature efficiency and the return air efficiency have the same magnitude, the expressions will be identical. It must be remembered, however, that there is often a minimum volume of outdoor air that must be supplied to a building or space for reasons of air quality. In a ventilation system with variable air flow, a VAV system, a heat exchanger can provide substantially better heat recovery than a system with return air, when air flows are low [Nilsson, 1994].

By connecting a heat recovery unit to the ventilation system, a considerable proportion of the energy, otherwise solely used to heat the supply air, can be saved. This is illustrated in Figure 4.

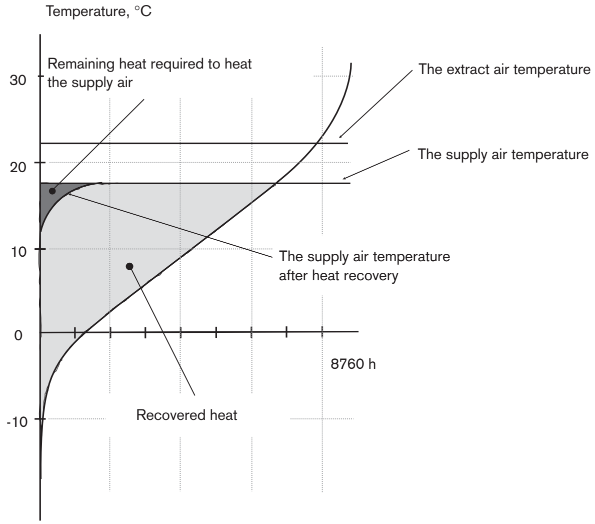

FIGURE 4. Heat recovery in a ventilation system shown in a duration diagram for the outdoor air temperature. The temperature data is for Berlin, Germany. The temperature efficiency of the system is 75%. The indoor temperature is 22 °C and the supply air temperature is 17 °C.

The lightly shaded area in the diagram represents the amount of heat saved for heating the supply air. The size of the area depends on the supply air temperature, the extract air temperature and the efficiency of the heat recovery system. It can be seen in Figure 4 that the temperature efficiency of the heat recovery unit will be gradually reduced over the year and will be zero when the outdoor temperature is the same as or higher than the supply air temperature. The proportion of recovered heat will therefore be larger than that indicated by the temperature efficiency of the recovery unit. The curves shown in Figure 5 are applicable to a ventilation system with constant air flow, a CAV system, in Stockholm, Sweden [Abel, Elmroth, 2006].

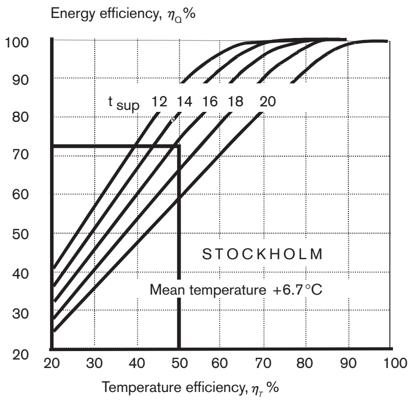

FIGURE 5. Energy efficiency as a function of temperature efficiency for a heat recovery unit installed in a ventilation system in Stockholm, Sweden. The energy efficiency Q, is defined as the ratio of the annual heat savings when using heat recovery to the annual heat requirements without heat recovery. The extract air temperature is 25 °C.

Example: A heat recovery system with a temperature efficiency of 50% is installed in a ventilation system in Stockholm. The supply air temperature is 16 °C. The energy efficiency of the system will be 74%, i.e. significantly higher than its temperature efficiency.

Both Figure 5 and the example illustrate how extremely important it is to know the difference between, and not confuse the use of, the terms temperature efficiency and energy efficiency. If they are confused, then completely inaccurate conclusions might be drawn regarding the operations of a ventilation system. It will be even more problematical, if comparisons are made between quotes for plant from two different suppliers and one of them states the temperature efficiency of the heat recovery unit and the other its energy efficiency. If the performance details given are not presented clearly, or if a mistake is made when making comparisons, and the efficiencies are regarded as being synonymous, the comparison will be totally misleading.

ENERGY NEEDS FOR COOLING VENTILATION AIR

In addition to ventilating a building to maintain a specified air quality, air is also often used to remove surplus heat, i.e. to cool the building. To do this, air has to be supplied to the building at a temperature lower than the indoor air temperature. In a ventilation system that removes heat from buildings, it is often the amount of air that has to be removed that determines the design air flow. This air flow, in turn, determines the dimensions of the ducting, the size of the air handling plant, etc. A system with variable air flow, installed in a building cooled by air, can be schematically illustrated as in Figure 6.

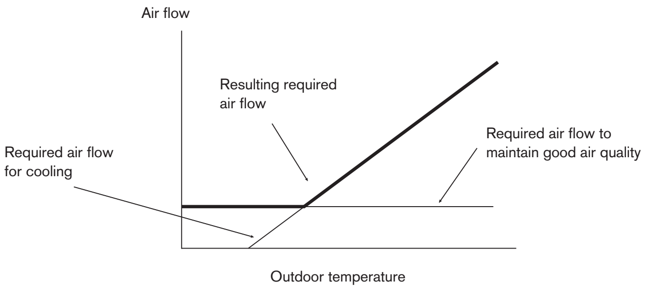

FIGURE 6. Air flows required to maintain air quality and supply sufficient cooling in a VAV ventilation system.

The required capacity of the cooling coils is determined by the state of the air before and after reaching the coils. If the heat content of the air before reaching the coils is denoted by h_{before} and the heat content after by h_{after}, the cooling energy required to change the state of the air can be expressed as:

$$\dot{Q}_{cool} = \dot{V}\rho(h_{before} - h_{after}) = \dot{V}\rho(\Delta h_{dry} + \Delta h_{wet}) \qquad (11)$$

When air is cooled by cooling coils the surface area of the coils is often colder than the dew point of the air. This means that when the air passes over the coils its temperature will be reduced and condensation will occur. Condensation will increase because of the increased amount of cooling energy required, relative to a dry cooling process, as the latent heat of the air also has to be removed by cooling. These two steps can be illustrated with the help of an hx or psychrometric diagram, see Figure 7.

The ratio $\frac{\Delta h_{wet}}{\Delta h_{wet} + \Delta h_{dry}}$ in Berlin is 0.38, which means that just under 40% of the cooling energy is used for wet cooling and just over 60% for dry cooling.

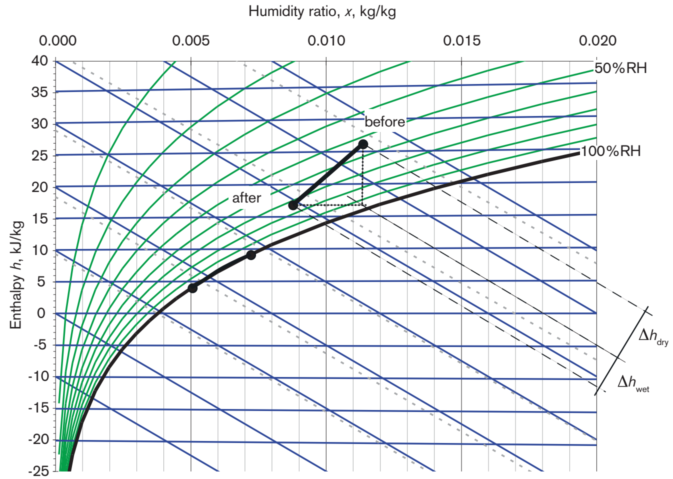

FIGURE 7. The change of state of the air passing over a cooling coil with a temperature lower than the dew point of the air. The state of the air before reaching the cooling coils corresponds to the design state for Berlin. The entry and exit temperatures of the coolant are 4 °C and 9 °C respectively.

In Europe, this ratio is between about 30 and 40%. In very warm and humid climates, for example in Orlando, USA, the wet proportion is nearly 60 %, while in very warm and dry climates, for example in Riyadh, Saudi Arabia, the wet proportion is practically negligible.

As a general rule, this means that to obtain the total energy required by the cooling coils, the dry proportion of the cooling energy is multiplied by a factor of 1.3 to 1.5 in climates similar to those in most of Europe (temperate zones) while the corresponding factor can be as high as 2.5 in warm and humid climates (tropical zones). In warm and dry climates (sub-tropical zones) the factor is close to 1.

The more humid the climate, the more important it is to investigate the possibility of reducing the moisture content of the ventilation air before it reaches the coiling coils. In some climate zones (tropical), return air systems are essential to keep the cooling energy demands low. Here, the extract air that is mixed with the supply air contributes to reducing its moisture content, subsequently reducing the amount of cooling energy that has to be circulated through the cooling coils.

In systems with heat recovery units, during the non-heating season, the function of the units can be reversed and they can, instead, be used for cooling purposes. When a building is cooled using an airborne system, outdoor air can be used as long as its temperature is lower than that of the supply air used to cool the building. Systems are often designed for supply air temperatures of 15 to 17 °C. This means that as long as the outdoor temperature remains below these levels no heat will have to be removed from the air, i.e. the cooling coils will not have to be in operation. In addition, the supply of coolant, normally produced in a refrigerating machine, will not be required.

The amount of heat that can be removed from the ventilation air can be illustrated by using an outdoor temperature duration diagram.

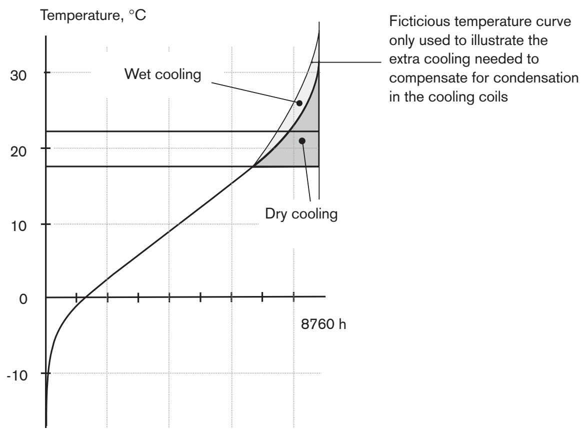

FIGURE 8. Heat removed from the supply air by the cooling coils. The total cooling required is sum of the dry and the wet cooling needs. The data is for Berlin, Germany. The indoor temperature is 22 °C and the supply air temperature is 17 °C.

As mentioned above, it is also possible to use the plant units intended for heat recovery for recovering cooling energy. The duration diagram for the outdoor temperature can be used to study the potential for recovering cooling energy. Figure 9 shows how much cooling energy can be recovered with the help of a non-hygroscopic heat recovery process.

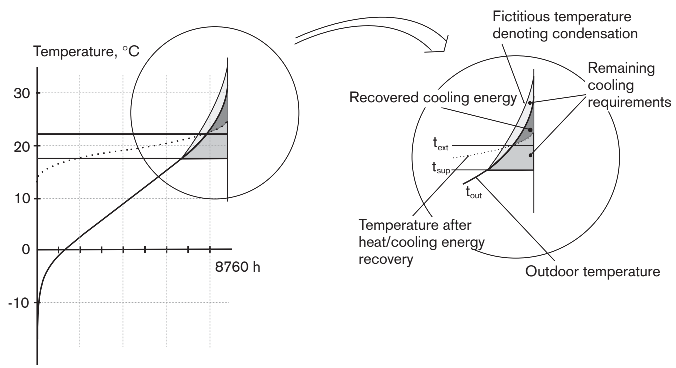

FIGURE 9. Cooling energy recovery in a non-hygroscopic heat recovery process with a temperature efficiency of 75%.

Cooling energy recovery is possible when the temperature of the extract air text is lower than the temperature of the outdoor air t_{out}. As shown in the diagram, cooling energy recovery is only possible when the outdoor temperature exceeds 22 °C.

In Figure 9, the supply air temperature curve after heat recovery has been plotted for the case in which the recovery process is in full operation all year round. In practice, however, the heat and cooling energy recovery processes will have to be regulated, so that the energy transfer is always correct, see Figure 10.

Relatively speaking, cooling energy recovery does not save as much energy as heat recovery in temperate climates, such as those found in Europe. The greater the difference between the supply air temperature t_{sup} and the extract air temperature t_{ext}, the smaller the relative opportunity for savings. While the recovered heat in these climates during the heating season can constitute as much as 90% of the heating requirements, the corresponding figure for the cooling requirements is in the region of 20%.

In other climate conditions, the effect of cooling energy recovery can be considerably greater. The warmer the climate, the greater the proportion of the total energy requirements needed to cool the ventilation air to room temperature. In sub-tropical and tropical climates, cooling energy recovery is becoming increasingly important. In tropical climates, with high temperatures and high humidity, there is every reason to con-

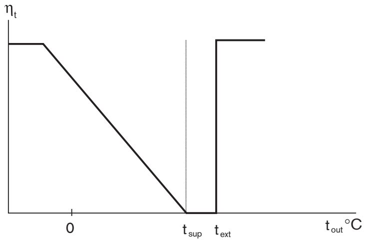

FIGURE 10. Change in temperature efficiency when both heat and cooling energy recovery is used.

sider using recovery solutions with hygroscopic units, for example, rotary hygroscopic heat recovery units.

The potential for using cooling energy recovery in a sub-tropical, i.e. warm and dry, climate is shown in Figure 11.

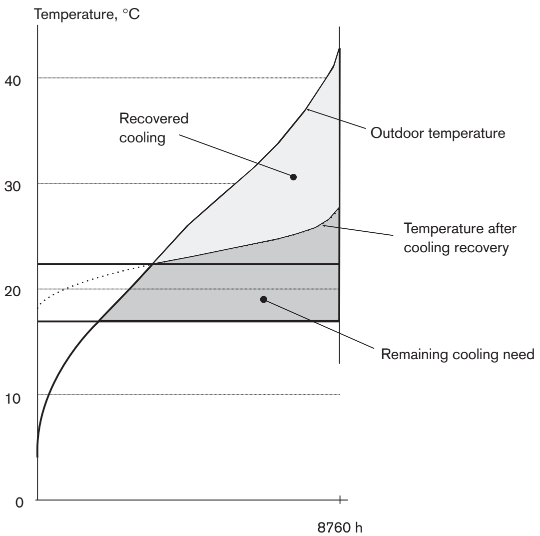

FIGURE 11. Cooling energy recovery in a warm and dry climate. The climate data is for Riyadh, Saudi Arabia where the mean annual temperature is 25.6 °C. The indoor temperature is 22 °C and the supply air temperature 17 °C. The temperature efficiency of the recovery system is 75 %.

As the climate is dry, the cooling required will only comprise a dry cooling process. There will be no need to heat the air at all, if heat recovery is used. Here, the cooling energy required is substantially greater than the heating energy required and the significance of cooling energy recovery is considerably higher than in cooler climates, see Figure 9. If cooling energy recovery is used, this process can supply about 50 % of the total cooling requirement.

If a hygroscopic heat recovery process is used, this will mean that both dry and wet cooling heat contents can be recovered. This is illustrated in Figure 12 for a warm and humid climate, where this type of recovery process is especially applicable.

In this case, the enthalpy efficiency is used instead of the temperature efficiency. Enthalpy efficiency is expressed as follows:

$$\eta_h = \frac{h_{sup}-h_{out}}{h_{ext}-h_{out}} \qquad (12)$$

The change in enthalpy efficiency is similar to that of the temperature efficiency shown in Figure 10, with the corresponding enthalpies used instead of temperatures.

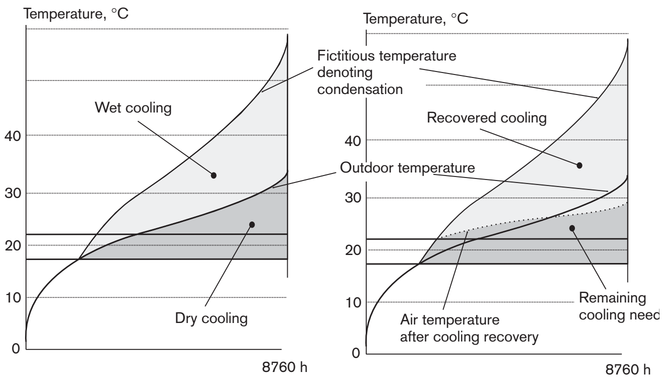

FIGURE 12. Cooling energy recovery using a hygroscopic unit in a warm and humid climate. The climate data is for Orlando, USA where the mean annual temperature is 23 °C. The indoor temperature is 22 °C and the supply air temperature is 17 °C. The enthalpy efficiency of the recovery system is 75 %.

Cooling energy recovery using a hygroscopic heat recovery unit offers considerably greater savings than recovery using a non-hygroscopic recovery solution.

Together, these figures and examples illustrate the importance of using energy recovery units in a ventilation system. In some climates, the recovered heat is an important factor while in others the recovered cooling energy is important. Important, in this context, means the potential of the solution to achieve considerable energy savings when heating and cooling the ventilation air. In warm climates, that are also humid, it might be advisable to use cooling recovery processes with hygroscopic units, whereby the lower humidity of the indoor air can be utilized and the need to dehumidify the incoming supply air can be avoided.

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22. HEATING SUPPLY AIR

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This chapter is all about heating supply air, primarily in air handling units but also in individual rooms. How to calculate the amount of energy required for heating air with the help of duration diagrams for the outdoor air temperature is discussed in the preceding chapter, Energy recovery. Energy calculations are therefore not discussed in this chapter.

WHY DOES SUPPLY AIR HAVE TO BE HEATED?

Air supplied to the different rooms in a building normally has to be heated to avoid causing any discomfort or disturbances in the occupied zones. A typical disturbance is draughts, i.e. air movements at speeds greater than about 0.15 to 0.20 m/s. If the supply air flow need is less than about 0.5 to 0.6 air changes per hour, the outdoor air in northern European climates can often be introduced unheated through outdoor air vents or gaps/holes in the building envelope, usually found around windows. When air flows are higher the outdoor air normally has to be heated. This can take place in an air handling unit, usually by using heat recovery units or return air, or by using preheaters. In non-residential buildings with a heat surplus, the supply air temperature is normally lower than the room air temperature while in residential buildings, with mechanical supply and extract air ventilation systems, the supply air is normally at the same temperature as the room air.

For most of the year, the supply air will have to be heated to reach the room temperature

In many buildings, the heating process is driven by internal heat sources, e.g. heat emitted from apparatuses, lighting and the occupants in the room and heat due to solar radiation through windows. This heating takes place after the supply air has been mixed with the room air.

Supply air is often preheated in a central air handling unit to a temperature below the desired room temperature. When the internal heat surplus is not sufficient to achieve the desired room temperature heat must be supplied via so-called zone heaters. These can be installed in individual rooms or in the supply air ducts leading to the rooms. The zone heater is adjusted so that the set point for individual rooms or groups of rooms is reached. Room heaters can be in the form of radiators, fan convectors and induction units. As the internal heat surplus can vary both in time and space, and between the different rooms in a building, it must always be possible to heat individual rooms to their desired temperatures.

Heating rooms using supply air

In this case, air heaters heat the supply air to a temperature higher than the desired room temperature and thus compensate for the heat losses caused by transmission and leakage through the building envelope. The air heaters here are usually duct heaters (reheaters) installed in the supply air duct just before the rooms.

The different stages for heating supply air in an air handling system are described below. Most of the heating takes place in the air handling unit.

HEATING SUPPLY AIR IN AN AIR HANDLING SYSTEM

When air is heated only its temperature is changed; its moisture content remains constant. This means that the heating power required to heat the air can be easily calculated as follows:

$$\dot{Q} = \dot{V} \cdot \rho \cdot c_p \; (t_{after} - t_{before}) \qquad (1)$$

where:
$\dot{Q}$ is the heating power in kW
$\dot{V}$ is the air flow in m^3/s
ρ is the density of the air in kg/m^3
c_p is the specific heat capacity of the air in kJ/(kg · °C)
t_{before} is the temperature of the air before heating in °C
t_{after} is the temperature of the air after heating in °C

In an air handling system, air can be heated in a number of stages as shown in Figure 1. This can be done by using:

1. Preheaters
2. Heat recovery units or return air functions
3. Reheaters
4. Zone heaters – duct heaters (reheaters)
5. Zone heaters – room heaters

The first three of these stages are often included in the air handling unit.

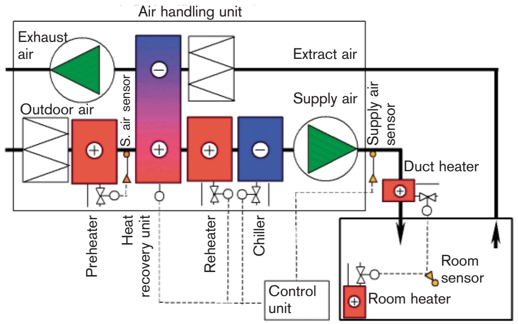


Figure 1. The different stages for heating supply air in an air handling system. The locations of the temperature sensors required to regulate the different stages are also shown.

If the supply air is required to be cooled when the outdoor temperature exceeds the desired set point temperature for the supply air, then a chiller will have to be installed. The supply air might also have to be dehumidified. Cooling is discussed in the next chapter, Cooling supply air. When cooling is required, the heat recovery unit, or the return air function, can also be used as a cooling recovery unit and this is discussed in the previous chapter, Energy recovery.

The different stages in an air handling unit, which all work together to maintain the desired supply air temperature downstream of the unit, must be regulated sequentially. However, if a preheater is used, this would be set to maintain a certain air temperature before the heat recovery unit, allowing the preheater to be regulated independently. Other stages must be regulated in the following order when the actual temperature of the supply air is less than the desired temperature:

A. The capacity of the chiller is slowly reduced to zero
B. Any heat recovery via the heat recovery unit is slowly reduced to zero

C. The capacity of the heat recovery unit is slowly increased
D. The capacity of the air heater is slowly increased

When the actual temperature of the supply air exceeds the desired temperature the regulating sequence is reversed, i.e. the reheater is turned down first and the chiller turned up last.

1. Preheaters

A preheater, i.e. heating coils placed before the heat recovery unit, is only needed in special circumstances and this is to prevent malfunction of the heat recovery unit due to freezing, caused by condensation and icing-up on the heat recovery unit's exhaust side when the state of the exhaust air is lower than its freezing point. Preheaters are normally only required in cold climates when:

- The design outdoor temperature is lower than about –10 °C.
- A lowest relative humidity has been stipulated in wintertime, which means that the supply air has to be humidified.
- The air handling unit is in operation 24 hours a day.

Problems caused by freezing on the exhaust side of heat recovery units arise primarily when units with high temperature efficiencies are used, especially non-moisture transferring rotary heat exchangers and plate heat exchangers. When high humidity levels are required indoors problems can also occur when fluid heat recovery units are used, even if these have lower temperature efficiencies. There are fewer problems when enthalpy exchangers (rotary, moisture transferring heat exchangers) are used. The extent of the problems can be easily seen by studying the processes with the help of Mollier diagrams (psychrometric charts).

When a return air function is used instead of a heat exchanger drops of water and ice crystals might form in the exhaust air and these could cause ice to build-up on the dampers and cold duct walls.

Preheaters mean extra initial investments but cost little to run in northern and central European climates, as they are only required for a short time. Because of the extra investment needed and the increase in the complexity of the air handling equipment, preheaters are not used in continuous operation. Instead, defrosting is achieved by reducing the degree of heat recovery.

Solutions like these, however, will require the reheater to be designed so that it can heat the supply air during the short periods when the heat recovery unit has been turned down to a minimum.

If the air handling system is only in operation during the daytime in winter, only a small amount of ice will build up on the exhaust side and no special defrosting measures will be required.

If the outdoor climate is such that frost can form and the equipment is in continuous operation, and required to provide quite high levels of humidity, it will be often difficult to avoid having a preheater. The preheater will then have to heat the outdoor air to about – 10 °C before it enters the heat recovery unit. The required temperature rise across the preheater as a function of the outdoor air temperature can be seen in Figure 2. An air temperature sensor installed before the heat recovery unit, as shown in Figure 1, is used to regulate the preheater.

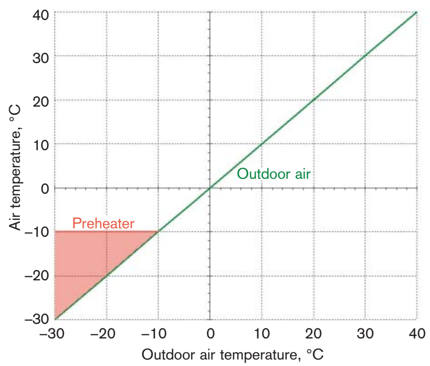


Figure 2. The temperature rise across a preheater as a function of the temperature of the outdoor air.

As the preheater heats air from a temperature below the freezing point of water, the coils will require a separate fluid circuit with antifreeze. This circuit will then require its own water/fluid heat exchanger. An electric preheater is simpler and costs less, although both electrical energy and power can be quite expensive in winter.

2. Heat recovery and return air

Heat recovery and the use of return air are two different solutions used to recover energy from the extract air and to transfer it to the supply air. Both solutions are discussed in the previous chapter, Energy recovery.

Table 1 lists the different types of heat recovery units normally used

Table 1. Common types of heat recovery units used in air handling systems [ASHRAE 2004].

Type of heat recovery unit	Typical temperature efficiency range	Type of capacity regulation
Rotary heat exchanger: – moisture transferring (hygroscopic enthalpy exchanger) – non moisture transferring	 50 to 85 % (same as moisture efficiency) 50 to 85 %	Rotor speed
Plate heat exchanger	50 to 80 %	Bypass valve
Fluid heat exchanger	55 to 65 %	Flow rate through supply air heating coils – often via a three-way valve
Tube heat exchanger	45 to 65 %	Partly self-regulating Bypass valve Angling of the tubes
Return air valve	70 to 90 %	Damper to regulate the proportion of return air, depending on required outdoor air flow

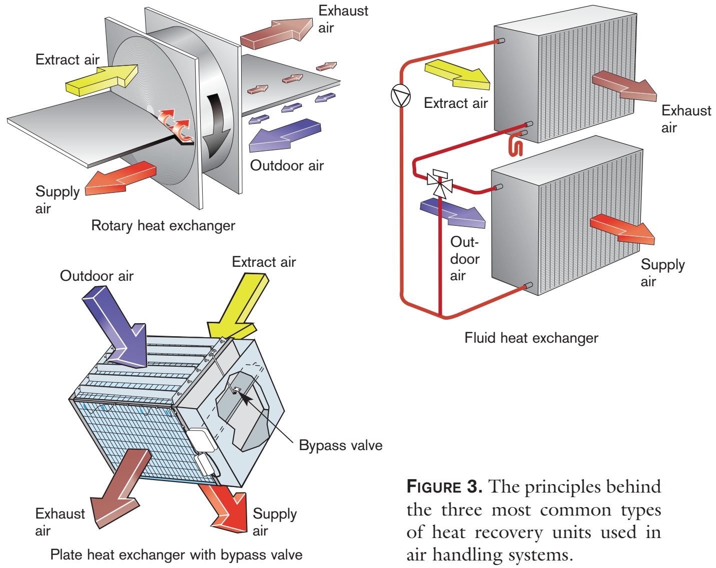


Figure 3. The principles behind the three most common types of heat recovery units used in air handling systems.

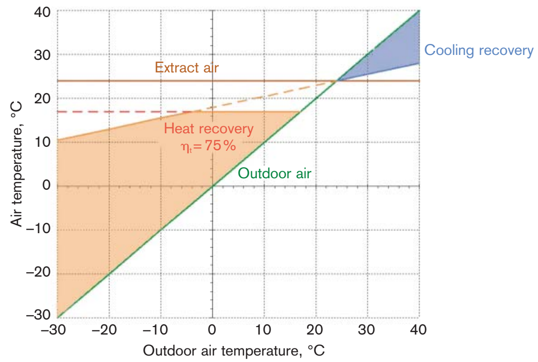


FIGURE 4. The temperature rise across the heat recovery unit as a function of the outdoor air temperature when preheating is not used.

in air handling systems. The table also shows the typical intervals for the design temperature efficiencies and how the capacities of the units are regulated.

Figure 4 shows the required temperature rise as a function of the outdoor air temperature across a heat recovery unit with a temperature efficiency of 75%, an extract air temperature of 24 °C and a constant supply air temperature of 17 °C. The air flows are constant and there is no preheater in the system.

It can be seen in the diagram that when the outdoor temperature is above about –5 °C the heat recovery unit can generate all the heat required to heat the supply air. To maintain the desired temperature of the supply air, the setting of the recovery unit must be reduced when the outdoor temperature exceeds –5 °C. When the outdoor temperature exceeds the extract air temperature the recovery unit must be run at full capacity to provide maximum cooling.

Figure 4 can, however, give the wrong impression with regard to how much heat can be recovered, as low outdoor temperatures only occur for a short time during the year. A more applicable picture is provided in Figure 4 in the preceding chapter, Energy recovery, where it can be seen that a heat recovery unit with high temperature efficiency is, in fact, run at reduced capacity for nearly 90% of the time that the supply air has to be heated. In warmer climates than those in northern Europe, the need for a preheater is therefore much less.

Figure 5, in the preceding chapter, also shows that the energy efficiency is higher than the maximum temperature efficiency when the supply air temperature is less than the extract air temperature. In the example in Figure 4 above, the energy efficiency is about 95%.

3. Preheaters

At very low outdoor temperatures, the heating capacity of a heat recovery unit is seldom sufficient to heat the supply air to the desired level. This means that a preheater will be required in the air handling unit.

Figure 5 shows the rise in required temperature across the heater as a function of the outdoor air temperature. It can be seen in the diagram that the air heater is only in operation when the outdoor air temperature falls below about – 5 °C. Thanks to efficient heat recovery, the output from the heat recovery unit is sufficient to cover the entire heating demand for most of the year.

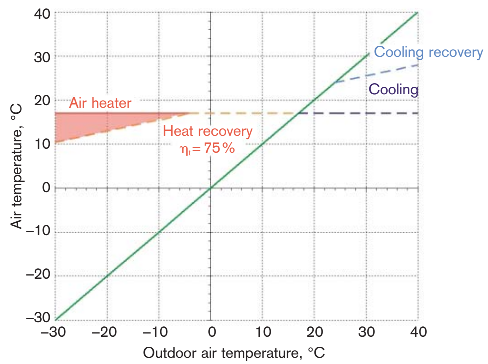


FIGURE 5. The temperature rise in an air heater as a function of the outdoor air temperature.

When heat recovery units with high maximum temperature efficiencies are used the annual operation time of the preheater will be short. In the example in Figure 5, based on the climate in Stockholm, the operational time is just less than 700 hours or just over 10% of the time that the supply air has to be heated during the year when the heat recovery unit is in continual operation.

4. Zone heaters: Duct and room heaters

Individual rooms or groups of rooms (zones), in which little or no heat is generated by, for example, solar radiation or internal heat sources, must be supplied with heat by using heaters in the supply air duct or heaters in the rooms. Although the required heating power will be the same no matter how the air is heated, the actual heating processes are quite different. When duct heaters are used the supply air is heated before it is introduced into a room or zone. When room heaters are used the supply air first cools the room air and the mixed air is then heated.

As shown in Figure 1, zone heaters like these, i.e. duct heaters or room heaters, are regulated by sensors that measure the room temperatures in the zone in question.

5. Zone heaters: Room heaters

In rooms where there is a heat surplus, cooling can be achieved by introducing supply air at a temperature lower than the room temperature, which means that the duct air heaters can be switched off. When the temperature of the supply air is lower than the temperature of the room air and the internal heat sources are not sufficient the room air will have to be heated to the desired room temperature. In rooms without any heat surplus at all and that are not heated by room heaters, the supply air might have to be heated right up to room temperature.

Figure 6 illustrates the heating requirements in an office, in which the supply air is heated by a zone heater, as a function of the outdoor temperature. In this case, the zone heater is a duct heater in the supply air duct before the room. No room heater is used.

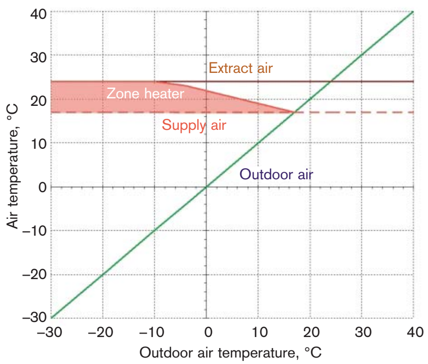


Figure 6. The temperature rise in a zone heater in the supply air duct as a function of the outdoor air temperature when the supply air is not sufficiently warm to heat the room.

If we assume that the temperature of the extract air and the room temperature are the same, then the maximum heat power rating of the duct heater in the example above will be:

$$\dot{Q} = \dot{V} \cdot \rho \cdot c_p \ (t_{room} - t_{sup}) \qquad (2)$$

where:

$\dot{Q}$ is the heating power in kW
$\dot{V}$ is the air flow in m^3/s
ρ is the density of the air in kg/m^3
c_p is the specific heat capacity of the air in kJ/(kg · °C)
t_{room} is the room temperature in °C
t_{sup} is the supply air temperature before the zone heater in °C

If, on the other hand, the supply air temperature is lower than the room temperature, even when it is cold outside, heat will have to be supplied via a room heater. The maximum heating power required is the same as in the example above, i.e. according to Equation (2). This means that it can also be illustrated as a temperature rise in the supply air as in Figure 6.

Maximum temperatures when heating rooms using supply air

When supply air is used to heat rooms (zones) a duct heater must be able to heat the supply air so that the heat losses due to transmission and leakages through the building envelope are also compensated for. This will be the case when no room heaters are used. Depending on how the individual heat surplus in each room varies, the supply air temperature will vary between a maximum value and no temperature rise at all, i.e. when air is supplied to the room at the temperature determined by the central air handling plant, see Figure 7.

The temperature of the supply air at the design outdoor temperature is, for comfort reasons, not normally chosen to be more than about 30 °C to 35 °C.

In USA-inspired so-called “all-air systems”, the room air is always heated via the supply air and no room heaters are used. In northern Europe, these systems are not normally used and rooms are heated using room heaters, such as radiators and induction units.

Heating room air using supply air has a couple of disadvantages:

1. The air handling system must be in operation 24 hours a day as rooms need to be heated, primarily at night, when no or few internal heat sources are available. This entails an increased use of fans compared to when room heaters are used.
2. When rooms are heated via warm air it can be difficult to ensure good mixing of the room air. If the supply air temperature exceeds the room temperature, the supply air can easily form a cushion of warm air near the ceiling and this is avoided by installing specially designed terminal devices.

Figure 7 shows how the maximum supply air temperature after a preheater varies with outdoor air temperature. The supply air temperature shown is only applicable to rooms where there is no heat surplus. On the other hand, in zones where a heat surplus has to be removed, the supply air will be at the main supply air temperature, i.e. the temperature before the preheater. This is shown as a dashed line in Figure 7.

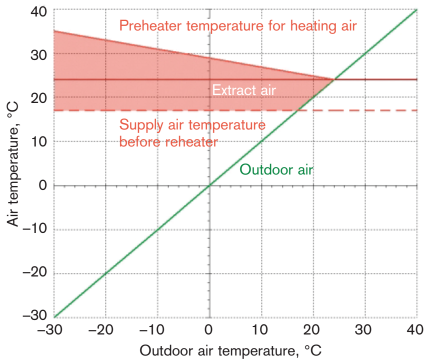


FIGURE 7. The temperature rise in a preheater as a function of the outdoor air temperature when supply air is used to warm rooms in a building.

UNITS FOR HEATING SUPPLY AIR

The most common way of heating supply air is by using heaters with hot water coils in the air handling unit. Fins on the air side increase the heat transmission. In other solutions, the hot water is replaced by low-pressure steam that condenses in the piping. In northern Europe, steam heaters are mainly used in industrial applications, where steam is often readily available. In countries where American technology is prevalent, steam heaters are quite common. Air heating using steam is not discussed further in the following.

Air heaters can be fitted with electrical heating elements instead of hot water pipes. The elements are not normally finned, but heaters with finned elements are sometimes used.

In countries where natural gas is available, the air can be heated using a natural gas burner in the air handling unit. Some solutions involve direct combustion, in which the gas burner is placed in the unit on the supply air side, or indirect combustion, in which the supply air is heated via flue gas pipes. Natural gas heaters require greater investments than hot water heaters but the total solution can be cheaper, as no extra water pipes have to be connected to the air handling unit, no shunt circuit is needed and fumes from natural gas combustion are easy to deal with. This is especially true in direct combustion solutions in which there are no special arrangements for dealing with fumes.

Supply air can also be heated directly using the heat from a refrigeration unit by placing the condenser coils in the supply air flow. In a solution like this, there must be a simultaneous need for cooling the refrigeration unit and heating the supply air. Another common way to use the heat from the condenser coils is to heat the water in the hot water system that is used, among other things, for the air heating coils. As a hot water system normally serves more than just the air heaters this will increase the chances that both cooling and heating will be needed at the same time. Solutions like these are described in the last sub-section of this chapter, which discusses systems for heat recovery from refrigeration units.

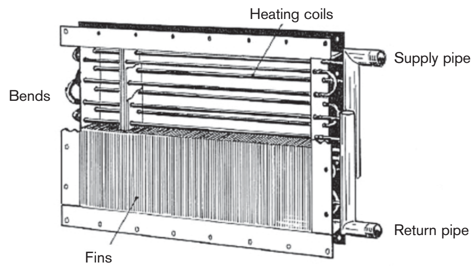


FIGURE 8. An air heater with two rows of heating coils and fins [Adapted from Stampe, 1992].

Air heaters with hot water coils
Air heaters that use hot water normally use 8 to 16 mm o.d. copper piping and aluminium fins spaced at 1.4 to 6.4 mm. Distances between pipes are normally 20 to 45 mm.

Air heaters can have one or more rows of coils. Figure 8 shows an air heater with two rows of coils, partly as a cutaway.

Other materials can also be used and fins of copper are often fitted where the atmosphere would corrode aluminium, for example, in coastal climates. Steel piping can also be used, if the outdoor climate allows, and steel fins are sometimes used as well.

Figure 9 shows an air heating battery for installation in an air handling unit. The feed and return pipes are hidden behind protective metal sheeting, which means that all the air will flow over the finned coils. The frontal area of the air heater is therefore somewhat smaller than the cross-sectional area of the air handling unit.

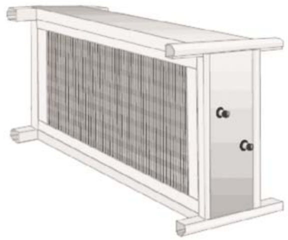
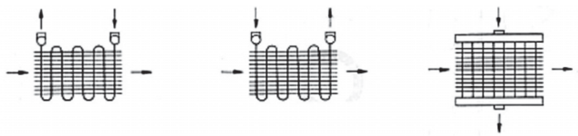


FIGURE 9. Air heater with finned hot water coils for installation in an air handling unit.

The ways in which the coils are connected together determine the heating power of the coils. There are three main ways in which the coils can be connected in the heater and these are defined by how the two media, air and water, flow in relation to each other.

Counter flow connections create the greatest average temperature difference across the heater and this means that the heating power per unit front area is also the greatest. A parallel flow connection creates the lowest average temperature difference while the cross flow connection is in between. Consequently, counter flow connections are the most common in air heaters. However, air heaters are always cross flow connected to some degree, as the coils always cross the air flow. The greater the number of coil rows the clearer the effect of the parallel or counter flow connections will be.



Counter flow Parallel flow Cross flow

[Illustrations from Lilja et al., 1971]

The detail design of an air heater, i.e. the pipe diameters, the fin separation, the number of coil rows and how they are connected, depends on the manufacturer. For a given size of air handling unit there are usually a number of air heaters with different power ratings to choose from. Their fin arrangements, coil separation distances and number of coil rows will therefore differ, depending on their power ratings.

The pressure drop on the water side of the coils will depend on how the coils are connected to the feed and return pipes. A common design criterion for air heaters is a maximum pressure drop on the water side of 20 to 30 kPa. The pressure drop is partly determined by the speed of the water. This should normally be greater than 0.2 m/s to avoid venting problems but it must not be too high, i.e. greater than about 1.5 m/s in copper pipes, to avoid corrosion problems. Traditionally, the speed of the air is chosen to lie between 2 and 5 m/s with respect to the front area of the air handling unit. If air handling units are designed for optimal power efficiency when using fans rated at 1.5 to 2.5 kW/(m^3/s), the air speed often has to be limited to 1 to 2 m/s.

In climates where the outdoor air can fall below zero, water heated coils must contain antifreeze. The most effective way to avoid freezing is to have an evenly distributed flow of water through the coils even when heating is not required. A pump is therefore normally needed in the circulation circuit. Otherwise, the most common way to avoid freezing is to measure the temperature of the water at the coldest point in the coils. If it falls below about 7 °C, a warning signal will be given, the supply air fan will be stopped, the outdoor air intake closed and the regulating valve for the air heater adjusted to maintain a minimum temperature of 7 °C in the return water. The temperature sensor is normally located in one of the coils but can also be located on the collecting pipe for the return water or in the coldest part of the air flow after the heater. In this latter case the alarm limit is normally set to 10 °C. Air heaters without conventional freeze-up protection are also available on the market. In these versions, the coil bends are fitted with small pipes to accommodate the expansion of the water in the battery if ice forms in the heating pipes.

The expanded water is either carried to the main return pipe or released into the surroundings via a safety valve. This solution is sometimes known as a “ThermoGuard”.

Electric air heaters

Electricity is simple to use for heating air in air handling units or in reheaters in the supply air ducts upstream of the heating zones. The elements in an air heater do not, as a rule, have fins and are sometimes coated in an electrically insulating layer of, for example, magnesium oxide. Finned versions, i.e. with insulated elements placed in finned steel pipes, are also available. Figure 10 shows a typical electrical heater with unfinned elements in an air handling unit.

FIGURE 10. An air heater with unfinned elements for installation in an air handling unit.

If unfinned heaters are used, the surface temperatures of the elements will be higher than in finned versions and the power rating can be as high as 320 kW/m^2 front area. Elements in finned piping might only have a power rating of 130 kW/m^2 front area. This latter solution is used when heating supply air containing a lot of dust (which would otherwise be burnt at high temperatures), in reheaters placed after air coolers, in which drops of condensation can occur in the air and might cause short circuiting of uninsulated elements, or when the elements are easily accessible.

In order to regulate an electric air heater, the elements must be switched on in steps. Elements with binary power ratings are the easiest to regulate, the steps then being 1, 2, 4, 8, 16, 32, 64, … kW. If the supply air temperature can be allowed to vary a little, the smallest power step could still keep any variation within about 2 °C. If more precise regulation of the supply air temperature is required, then an extra continuously variable 1 kW element will be needed and a thyristor can be used to regulate the power. If this solution is adopted, then an electric heater can manage all the continuously variable heating powers required just as well as an air heater that uses hot water.

Natural gas air heaters

In many countries natural gas is a common energy carrier. This type of fuel has the advantage of being easy to handle and, as it does not contain sulphur, it can also be used in highly efficient condensing boilers. The supply air can be heated in two ways in a natural gas burner in an air handling unit:

1. By direct combustion in the supply air.
2. By indirect heating of the supply air via a heat exchanger with flue gas pipes.

As natural gas is a clean fuel, it can be used for direct combustion in the supply air. Cooking stoves that use natural gas are common in a lot of countries and here natural gas fired ceiling radiators (infrared heaters) are often used in industrial buildings, the combustion products being allowed to exhaust into the premises. This means that it must be acceptable to allow supply air with somewhat higher pollution levels of combustion products. One group of such products comprises the nitrous oxides that are formed by the nitrogen and oxygen in the air when combustion temperatures in the flames exceed 1000 to 1200 °C. It is important to make sure that combustion is complete, so that no poisonous carbon monoxide is formed. When natural gas is combusted carbon dioxide and a considerable amount of water vapour are, of course, also created.

If direct combustion heating is not possible, gas-fired air heaters can be used in which the fumes flow through finned or unfinned fume pipes and then via a flue into the open air.

Natural gas heating means that a very simple type of flue can be used.

Air heated by using natural gas requires a burner that can be finely regulated.

Heating supply air via condenser coils in a refrigeration unit

A simple way of making use of the heat derived from condensation in refrigeration units is to allow the supply air to cool the condenser. This means that there has to be a demand for heating the supply air at the same time as there is a need for cooling the refrigeration unit. The principle can be applied, for example, in refrigerated and frozen food displays. However, if the refrigeration unit is only used for comfort heating, the cooling need will mostly occur in the summer when there is no corresponding need to heat the supply air and it will be also more difficult to use the heat from the condenser.

One way of utilising the condenser heat would be to let the hot water circuit, used for the air heater, cool the condenser. This will most probably increase the likelihood of needing simultaneous cooling and heating, especially if the system also supplies hot tap water and, quite often, an accumulator tank. This means that there will always be some way of using at least some of the heat from the condenser. This type of system is discussed in the final sub-section below on systems for heat recovery from refrigeration units.

SUPPLY AIR HEATING SYSTEMS

Air heaters can be supplied with heat in a number of different ways:

- Via a waterborne heating system based on
  - – a central heating boiler
  - – a district heating system
  - – heat recovery from refrigeration units and district heating systems as well as central heating units with condensing boilers
- Direct acting electrical heating, i.e. electric air heaters, as described above.
- Natural gas heaters, i.e. air heaters with direct or indirect natural gas burners, as described above.

The following sub-sections look at the principles behind how air heaters are connected to the first three systems mentioned above. Only the essential components for the primary functions are included, which means that shut-off valves and adjustment valves are not discussed. For detailed design of pipe systems for convenient commissioning and control, [Nilsson, 2003] and [Petitjean, 2004] are recommended.

The capacities of air heaters that use hot water are regulated in two principally different ways: by varying the temperature of the water and by varying its flow. These two solutions require the air heating batteries to be connected in different ways on the water side.

Waterborne heating systems with a central heating boiler

A central heating boiler can be run on a number of different fuels. The traditional fuel since the Second World War has been oil, but this is now becoming much less common. In a number of countries, natural gas is a common fuel while bio-fuels in other countries are now often regarded as being carbon dioxide neutral. For small-scale use, wood pellets are common while wood chips are used in large-scale set-ups.

When a fuel that generates corrosive fumes is used the temperature in the furnace must be maintained at a high level so that the water vapour in the fumes cannot condense. This requires specially designed shunt groups, so that the water returned to the boiler is not too cold.

If the fuel has a low sulphur content, like natural gas, or is a bio-fuel with quite a high water content, condensing boilers can be used and these often provide a desirable solution as the condensation process raises the efficiency of the boiler. To achieve effective condensation it is desirable to have a low return temperature and this means that the heating system can be designed in a similar way to a district heating system, in which as low as possible return temperatures are aimed for. See the sub-section below on waterborne heating systems based on district heating.

Figure 11 shows the connection diagram for air heating in a waterborne heating system based on a central boiler. Radiators and other heat users, such as hot water systems, are also connected to the heating system.

If there is a danger of an air heater freezing, it is usually connected so that it can be supplied with a constant flow of water. The temperature of the water supplied to the air heater is then regulated using a three-way valve. In small air heaters, under about 5 kW design power rating and that are used as duct heaters in which there is no danger of freezing, a variable water flow can be used through the air heater. This means that there will be no need for a circulation pump with a bypass circuit and a non-return valve.

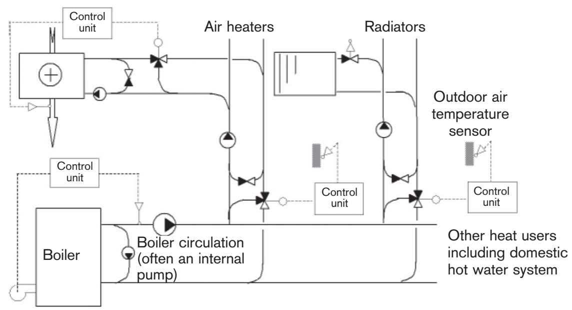


FIGURE 11. Connecting an air heater to a non-condensing central heating boiler.

The flow temperature in the pipes that supply the air heaters can be regulated via a shunt, set according to the outdoor air temperature. The mains that supply radiators can be regulated in a similar way. In large buildings, radiator mains are normally fitted with a shunt per facade, so that the radiators can be regulated independently. Room temperatures are then regulated by using self-acting thermostat valves fitted to each radiator. This means that the water flow in the radiator circuit will vary and, when there are large heat surpluses in many of the rooms, for example when there is intense sunshine through windows in spring, the flow of water will be greatly reduced, as nearly all the valves connected to the supply pipe will close.

The flow temperature from the boiler is kept constant by regulating the capacity of the burner in steps or by switching it on and off.

Waterborne heating systems connected to district heating

Figure 12 shows the connection diagram for air heating in a waterborne heating system connected to district heating. Radiators and other heat users, such as hot water systems, are also connected to the heating system.

When a building is heated via a district heating system the piping system is always designed to give the lowest possible return temperature. This is done by allowing variable water flows in the piping which supplies the heat transferring apparatuses as well as in the flow pipes from the heat exchanger in the district heating unit. This is achieved by using two-way valves. The flow through the heat transferring apparatuses can, however, be kept constant and this is achieved by using a two-way valve and a bypass pipe, see Figure 12. This connection is sometimes replaced by a three-way valve, which will then allow an approximately constant flow of water through the air heater or radiator circuits.

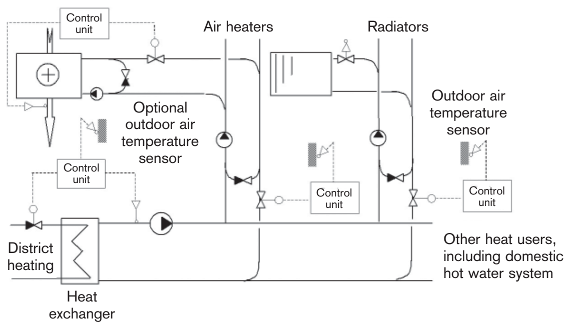


FIGURE 12. Connecting air heaters to district heating.

The flow temperature from the district heating unit's heat exchanger is regulated by varying the flow of the water in the district heating system. The flow temperature is normally regulated according to the outdoor air temperature. In a similar way to a system with a central boiler, it is advisable to have separate shunt groups for the radiator and air heater systems. Connections as shown in Figure 12 are also suitable for a central condensing boiler. The boiler then replaces the heat exchanger in the district heating system.

Waterborne heating systems with heat recovery from refrigeration units and a district heating system or a central condensing boiler

When recovering heat from a condenser in a refrigeration unit via a hot water circuit there must be a simultaneous need for cooling the refrigeration unit and heating the water system. If the water system supplies heat to the domestic hot water system it is more probable that these needs will occur simultaneously.

If the refrigeration unit is to contribute energy to the heating system, its return temperature must not be too high, preferably not more than about 50 to 60 °C. This means that it is quite often viable to recover heat from the condenser and use it for a heating system designed to run on district heating or a central condensing boiler. The refrigeration unit must have a coolant that is suitable for quite high condensation temperatures, around 50 to 60 °C.

District heating plant owners are not always keen to allow heat recovery from refrigeration units if this means that the temperature of the return water will be raised. To prevent this, the condenser in the refrigeration unit must be connected in parallel to the district heating heat exchanger.

Figure 13 shows an example of how heat recovery from the condenser in a refrigeration unit can be connected to a hot water system that derives heat from a district heating system. In this example the condenser is connected in series with the district heating heat exchanger.

To ensure that the refrigeration unit can always be cooled, a separate connection to a coolant cooler or a cooling tower is needed. At the same time, a connection is needed so that the condenser, when possible, can preheat the return water from the heating system before it reaches the district heating heat exchanger or the central boiler. A simple but slightly more expensive solution is to use a refrigeration unit with a double bundle chiller. In this case, a pipe circuit is connected to the coolant cooler and a circuit to the heating system, as shown in Figure 13.

To prevent the coolant from being heated when the refrigeration unit is in operation, two two-way valves or one three-way valve, have to be installed in the heating circuit connection to the condenser. When the refrigeration unit is switched off the return water will be returned straight to the district heating heat exchanger and, when heat recovery is possible, it will flow through the condenser. When the cooling of the refrigeration unit is insufficient the pump in the coolant circuit will start.

The condenser can, of course, be fitted with a single pipe circuit, but this will mean that the connections and regulation will be more complicated, and the risk of problems due to poor function will be greater, due to leakages in valves etc.

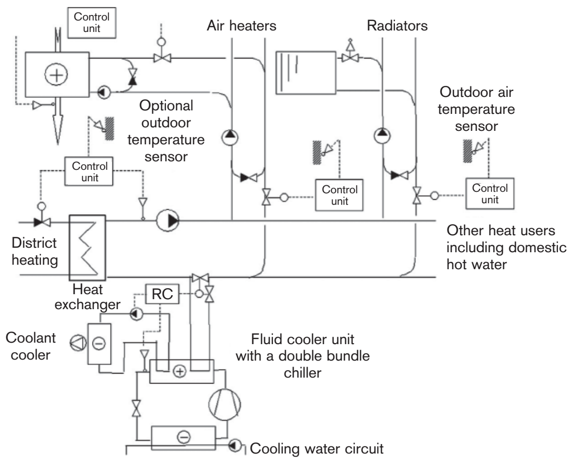


FIGURE 13. Heat recovery connections to the condenser side of a refrigeration unit when using district heating or a central condensing boiler. The condenser in the refrigeration unit is connected in series with the district heating heat exchanger.

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23. COOLING SUPPLY AIR

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WHY COOL SUPPLY AIR?

In many buildings, the indoor climate requirements stipulate that, during a normal year, the indoor temperature must not be allowed to exceed a given maximum for more than a limited number of hours and that it must never exceed a specified upper limit. How requirements regarding indoor climate are specified are shown in Chapter 14/The Client and the building process and in Chapter 10/Thermal climate. It is important that requirements like these are formulated with great care and judgement: letting the temperature of the room air rise for a few hours during the warm season, means that the size of the HVAC system and its annual energy use can be kept within reasonable limits.

This means that, in order to fulfil the indoor thermal requirements, the heat surplus created in every individual zone in a building must be removed. This is the job of the HVAC system and cooling is achieved using air or water. These two solutions are discussed in detail in Chapter 27/Airborne indoor climate systems, and Chapter 28/Waterborne indoor climate systems.

When the cooling medium is air it will have to be cooled when the outdoor temperature is higher than the desired supply air temperature. This process takes place in the air handling unit. The lowest supply air temperature is normally determined by thermal comfort considerations, i.e. the supply air must not be too cold, as this could cause draught problems. The air flow to an individual zone is determined by the design heat surplus in each particular zone.

In humid climates, which are also often warm, the supply air must also be dehumidified so that the relative humidity of the room air lies between the limits used to define acceptable thermal comfort. This dehumidification is often achieved by a cooling process in the air handling unit.

When the cooling medium is water it is used to chill surfaces in the different building zones, and, in turn, cool the space or rooms by convection and/or radiation. This can be achieved by using active or passive chilled beams, induction units (often as window units), fan convectors and chilled ceilings. The temperature of the water is determined by two opposing factors: the cooling needed to cool the room air, i.e. the temperature difference between the room air and the chilled surfaces, and the risk of condensation of the moisture contained in the room air on the chilled surfaces. The water temperature is required to be as cold as possible to reduce the size of the units in each room and as high as possible to avoid condensation on the chilled surfaces. In many buildings, in the summer, moisture is introduced via the supply air. The internal generation of moisture is limited and normally comes from the air exhaled by the occupants. In cases like these, the moisture content can be reduced by dehumidifying the supply air by cooling it in the air handling unit. It will then be possible to lower the temperature of the cooling water in the room units. This is sometimes even necessary in dryer climates, such as in Europe.

In buildings with considerable generation of internal moisture, the air can be dehumidified by placing dehumidifying units in each room or by recirculating dehumidified extract air via a centrally located circulation unit.

HOW IS SUPPLY AIR COOLED?

Supply air is often cooled in the air handling unit by chilled water or an evaporative cooling medium flowing through finned cooling coils. The water is normally chilled in an electrically driven refrigeration unit working on a vapour compression cycle. This process, and the absorption process, is described below in the section on systems for generating cooling.

Besides cooling and dehumidifying air using cold surfaces in cooling batteries, the air can also be first dried and then cooled via humidification in a heat-driven sorptive cooling process in the air handling unit. A brief description of this method is also given below in the sub-section on sorptive cooling.

In the following, the principles behind the cooling of air are shown in Mollier diagrams (psychrometric charts). The processes illustrated are cooling via evaporative refrigerants and cold water, and the sorptive cooling process.

When outdoor air is cooled condensation normally occurs and the moisture content of the air is reduced. For condensation to take place, the dew point of the air must be higher than the temperature of the cold surfaces in the cooling coils. If the dew point of the air is lower than the temperature of the cold surfaces, the air will be cooled without it being dehumidified. This means that the change in temperature will take place at a constant moisture content x, i.e. along a vertical line in the Mollier diagram, similarly to when it is heated.

When condensation occurs on cooling it is not all that easy to determine the change in state of the air. This depends, among other things, on the size of the cooling battery and, primarily, the area of the cooling surfaces and their temperatures. A theoretical way of taking this into consideration is to assume that a certain proportion of the supply air (outdoor air) does not come into contact with the cold surfaces in the battery. It is then assumed that untreated outdoor air is mixed with dehumidified supply air after the battery. The theoretical proportion of mixed air is determined by a battery constant known as the bypass factor.

Whichever way the air is cooled, the require cooling power is derived from:

$$\dot{Q}_{cool} = \dot{V} \cdot \rho \cdot (h_{after} - h_{before}) \text{ kW} \qquad (1)$$

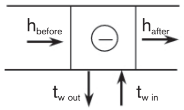
where:

$\dot{V}$ is the air flow in m^3/s

ρ is the density of the air in kg/m^3

h_{after} is the enthalpy of the air after the air cooler in kJ/kg

h_{before} is the enthalpy of the air before the air cooler in kJ/kg



As cooling batteries operate at smaller temperature differences between the surfaces of the battery and the air than in heating batteries, and also often dehumidify the air, they have much larger surface areas than heating batteries. A cooling battery normally comprises a number of rows of finned pipes.

As soon as the air has to be dehumidified, and not only cooled, the required cooling power increases significantly. The cooling batteries will then require larger surface areas, i.e. more rows of pipes.

Direct expansion coils with an evaporative refrigerant

When the cooling coils contain an evaporative refrigerant the surface temperature of the coils can be assumed to be the same as the vaporizing temperature of the refrigerant. Traditional refrigerants have a constant vaporization temperature at a given pressure. Many new refrigerants are mixtures of two or more different refrigerants and therefore display a variety of vaporization temperatures, a so-called temperature glide, at constant pressure. The vaporization temperature of mixed refrigerants can increase by 4 to 7 °C when they flow through the battery, while the temperature would be constant for traditional refrigerants.

To guarantee complete vaporization, the refrigerant is usually slightly heated, normally by 3 to 6 °C, before leaving the coils. This means that a small proportion of the surface area of the cooling coils will be slightly warmer than the vaporization temperature.

Figure 1 shows a typical change of state when cooling air in direct expansion coils from a design outdoor air state to a typical supply air temperature. The change in state of the air takes place towards the surface temperature of the cooling battery, which can be assumed to be the same as the vaporization temperature of the refrigerant. The surface tempera-

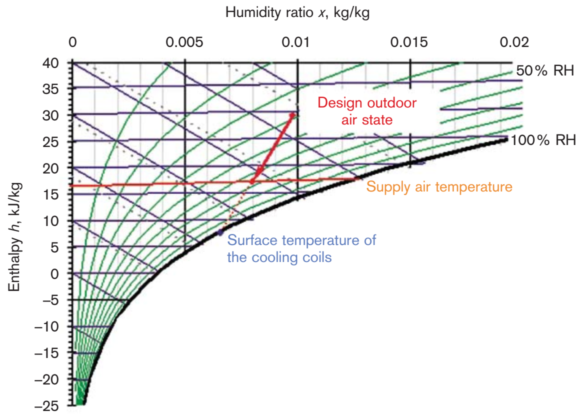


FIGURE 1. Example showing the cooling of air, in Berlin, from the design outdoor state to 17 °C using coils with a vaporizing refrigerant, with a vaporization temperature of + 8 °C. The 29.9 °C dry-bulb temperature and 18.9 °C wet-bulb temperature are only exceeded 0.4% of the time in a normal year.

ture of the cooling coils, i.e. the vaporization temperature of the refrigerant, is read off from the saturation curve in the Mollier diagram, which assumes that the moisture condenses and the surface is therefore wet.

The design outdoor air state is normally chosen according to the stipulated comfort requirements. Here, the most stringent has been chosen, one that is only exceeded for 0.4% of the time, i.e. 35 hours per year. Sometimes, less strict demands are made and the outdoor air state is exceeded for 1 to 2% of the hours in a normal year (88 to 175 hours per year). The demands made are often a question of tradition. If a less strict demand is made, the design cooling power will be lower and the size of the cooling plant will normally be smaller. The investment costs will therefore also be smaller. The choice of the design outdoor air state only affects the cooling energy to the extent that a large plant will be in operation for a longer period of time at reduced load.

Figure 2 illustrates the change in state of the supply air when the dew point of the outdoor air is lower than the surface temperature (vaporization temperature) of the cooling coils and the outdoor air is therefore not dehumidified. The state of the outdoor air shown in the example is not untypical for Berlin, 25 °C and 30% RH. The change of state of the air takes place as a dry change, i.e. along a constant value of *x*.

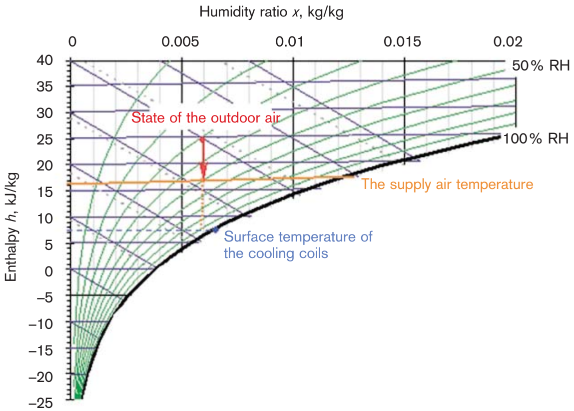


Figure 2. Example showing dry cooling of air when the dew point of the air is lower than of the surface temperature of the cooling coils, when using coils with a vaporizing refrigerant. The vaporization temperature is +8 °C.

Cooling coils with chilled water

When coils with chilled water are used the water is often conditioned in a liquid refrigeration process, normally operating according to the electrically driven vapour compression cycle. A refrigeration unit normally supplies chilled water at a constant temperature of around 5 °C, but more modern units can sometimes be regulated to raise the water temperature when less cooling is required. This will then increase the efficiency of the unit when partially loaded. Cooling coils are normally designed for a water temperature rise between 5 and 10 °C.

In a similar way to heating coils, cooling coils can be connected in a number of different ways, see Chapter 22/Heating supply air. The most common type of connection is the counter flow connection, which also provides the best dehumidification of the air. A counter flow connection offers the smallest surface area for a given cooling rating. If less dehumidification is required, the coils can be connected for parallel flow operation. In the following, the changes in state of the air are only illustrated in Mollier diagrams for counter flow coils. A Mollier diagram for cooling air using parallel connected coils is shown in Chapter 24/Humidification and dehumidification.

When illustrating the change in state of the air in a Mollier diagram, when it is cooled in coils with chilled water, it can be assumed that all the air that flowing through the coils passes over cold surfaces at the temperature of the outflowing water. In a similar way, all outflowing air meets the cold surfaces at the temperature of the incoming water. This is shown in Figure 3.

Figure 4 shows the change in state of the supply air from the design air state to a typical temperature when cooling using counter flow connected coils with chilled water. The water temperature varies between 6 °C, incoming, and 12 °C, outgoing.

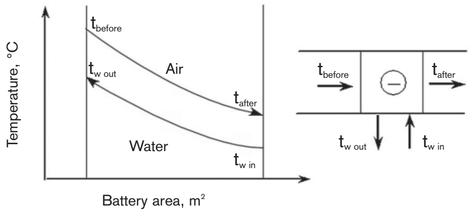


Figure 3. The temperature conditions in counter flow connected coils using chilled water.

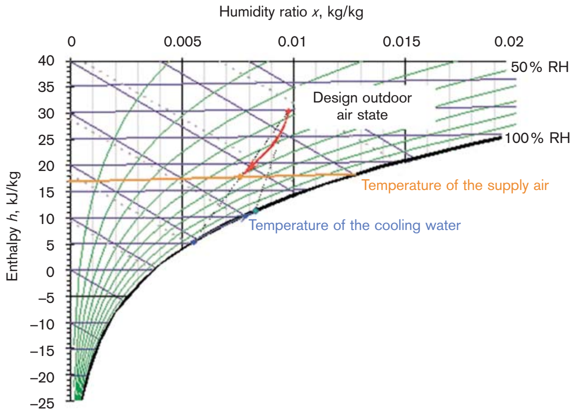


FIGURE 4. Example showing the cooling of air, in Berlin, from the design outdoor air state, exceeded only 0.4% of the time in a normal year, to 17 °C using counter flow connected coils with chilled water. The incoming and outgoing water temperatures are 6 °C and 12 °C respectively.

Sorptive cooling

Another way of cooling air is to combine the processes of dehumidification, heat recovery and humidification in a single process known as sorptive cooling. One of the advantages of this process is that the cooling and dehumidification processes are heat driven. A disadvantage is that a larger number of stages are needed in the air handling unit than when cooling coils are used. This causes a greater pressure drop and increased use of electrical power for the fans. A considerable amount of water is also needed for humidification.

An air handling unit with sorptive cooling can cool supply air in four different ways, namely by:

- Indirect evaporative cooling – cooling the supply air using a non-moisture transferring heat recovery unit using humid, and thereby cooled, extract air
- Direct evaporative cooling – cooling the supply air directly by humidification
- Combined direct and indirect evaporative cooling
- Dehumidification – to make the direct evaporative cooling process more efficient

The sorptive cooling process in an air handling unit, according to the Pennington cycle, comprises a number of air handling components with the following functions:

1. A hygroscopic drying wheel that both dries and heats the supply air (outdoor air) using extract air heated to its so-called regeneration temperature.
2. A heat recovery unit between the extract air and the supply air that does not transfer moisture and that cools the heated supply air with the help of the extract air. The heat recovery unit can also be used to cool the supply air using an indirect evaporative cooling process.
3. Direct evaporative cooling using a humidifier in the supply air.

These air handling functions are discussed in greater detail in Chapter 24/Humidification and dehumidification.

Figure 5 shows the principles behind air handling units that use sorptive cooling.

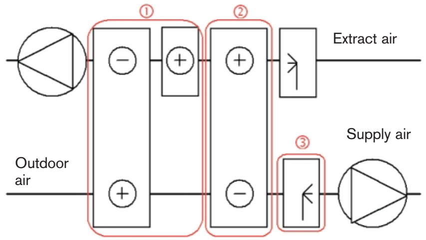


FIGURE 5. A schematic of an air handling unit operating according to the Pennington cycle, showing the three steps for treating supply air.

The change in state of the supply air in a sorptive cooling process, from the design outdoor air state to a typical supply air temperature, is shown in Figure 6. The drying wheel heats and dries the outdoor air along a line that is somewhat steeper than that for constant enthalpy.

The practicability of the sorptive cooling process depends on the state of the outdoor air, the desired supply air temperature and the regeneration temperature used in connection with the drying wheel. The more humid the outdoor air and the lower the supply air temperature, the higher the regeneration temperature must be.

In countries where district heating is available, it can be used in the summer to drive sorptive cooling processes. However, the flow tempera-

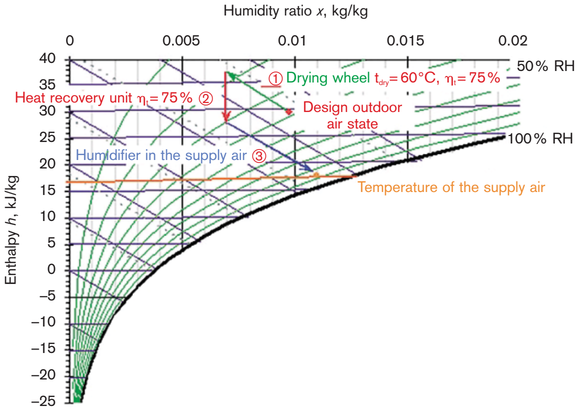


FIGURE 6. Example from Berlin, showing the cooling of supply air from the design outdoor air state, exceeded 0.4% of the year, to 17 °C using a sorptive cooling process.

ture from a district heating system in the summer is limited by the fact that heating is only produced for the production of domestic hot water. This means that the regeneration temperature in the extract air is limited to about 60 °C. An alternative would be to use solar heat energy to drive the sorptive cooling process. Even in this case, it would be desirable to limit the regeneration temperature, so that a reasonable heat supply could be maintained from the solar collectors. In other applications, regeneration temperatures up to 100 °C, and even higher, are used, which radically increases the practicability of sorptive cooling in warmer and more humid climates.

Figure 7 shows the limitations of the different techniques for evaporative and sorptive cooling. The diagram shows the air state limits for Stockholm and the limitation lines for indirect, direct, and indirect and direct evaporative cooling, and for sorptive cooling. In the air states to the left of and under each limit line the techniques shown can cool the supply air down to 17 °C. If the air states are above and to the right, cooling of the supply air can only be achieved to a temperature above 17 °C.

From Figure 7 it can be seen that, with the given equipment properties, only a small range of states can cool the supply air to 17 °C when using evaporative cooling, whatever the state of the outdoor air. Direct

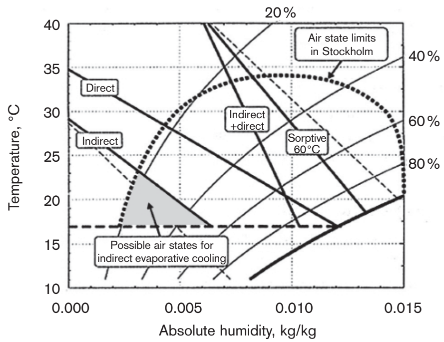


FIGURE 7. Curves showing the limits for evaporative and sorptive cooling based on the performance of components normally used in Sweden [Lindholm, 1998]. The supply air can be cooled to 17 °C and the heat supplied within the building corresponds to a further 7 °C, i.e. the extract air temperature will be 24 °C. The ratio of sensible heat supply to latent heat supply is 0.8. The air state limits for Stockholm are also shown in the diagram.

evaporative cooling can be used for a larger range of states but there could be a disadvantage, as the supply air might be too humid to meet good indoor climate comfort demands. If indirect and direct evaporative cooling methods are combined, the range of outdoor climate states increases dramatically, especially when it is warm outdoors. If the relatively low regeneration temperature of 60 °C were used, sorptive cooling would be viable in a number of different outdoor air states. However, there is still quite a large area in the Mollier diagram where the outdoor climate lies to the right and above the limiting line for sorptive cooling. On the other hand, the mumber of hours in Stockholm during a normal year, during which the outdoor climate lies in this area, is very limited – fewer than 30.

Figure 8 shows the number of hours that it is probable that the room temperature will exceed the desired temperature of 24 °C, when using sorptive cooling, on condition that the desired temperature of the supply air is lower than 17 °C and the regeneration temperature is 60 °C.

In Figure 8 it can be seen that in a Stockholm climate, if the supply air temperature is 17 °C, the desired room temperature of 24 °C is only exceeded for 23 hours during a normal year. If, on the other hand, the desired supply air temperature, to ensure a maximum allowable room

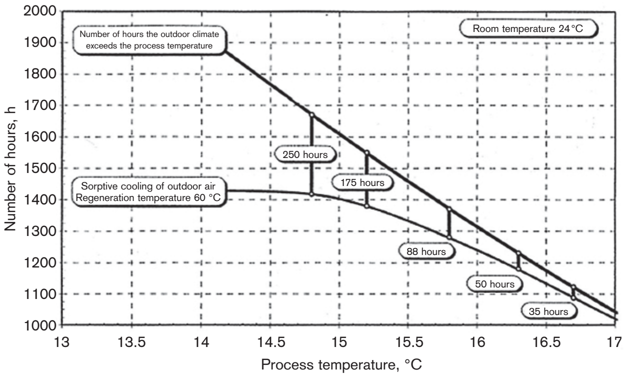


FIGURE 8. The number of hours that the room temperature in a building in Stockholm probably exceeds 24 °C as a function of the supply air temperature (the process temperature) when sorptive cooling is used with a regeneration temperature of 60 °C [Lindholm, 1998].

temperature of 24 °C, is just under 15 °C, i.e. 14.8 °C, the desired room temperature will most probably be exceeded for 250 hours during a normal year.

USING ENERGY FOR COOLING

The use of energy for cooling supply air is completely dependent on the outdoor climate and the desired state of the supply air. In northern European climates, the use of energy for cooling, and especially for dehumidification, is relatively small compared to the use of energy for heating air, particularly when heat recovery and return air are not used. In warm and humid climates, the amount of cooling energy used to cool the air is far greater than all other uses of energy for conditioning indoor air.

Figure 9 shows the duration curves for outdoor air temperatures representing three different European climates. The number of hours during which a typical supply air temperature of 17 °C is exceeded is shown for each climate. As shown in Chapter 21/Energy recovery, the required annual amount of energy for dry cooling can be calculated from the number of degree hours corresponding to these areas.

In Figure 9 it can be seen that a typical intake air temperature of 17 °C is exceeded for about 1210 hours per year in Stockholm, about 1790 hours in Berlin and about 3530 hours per year in Rome. These figures alone show that the use of energy for dry cooling in a Scandinavian climate differs greatly from the use in a Mediterranean climate.

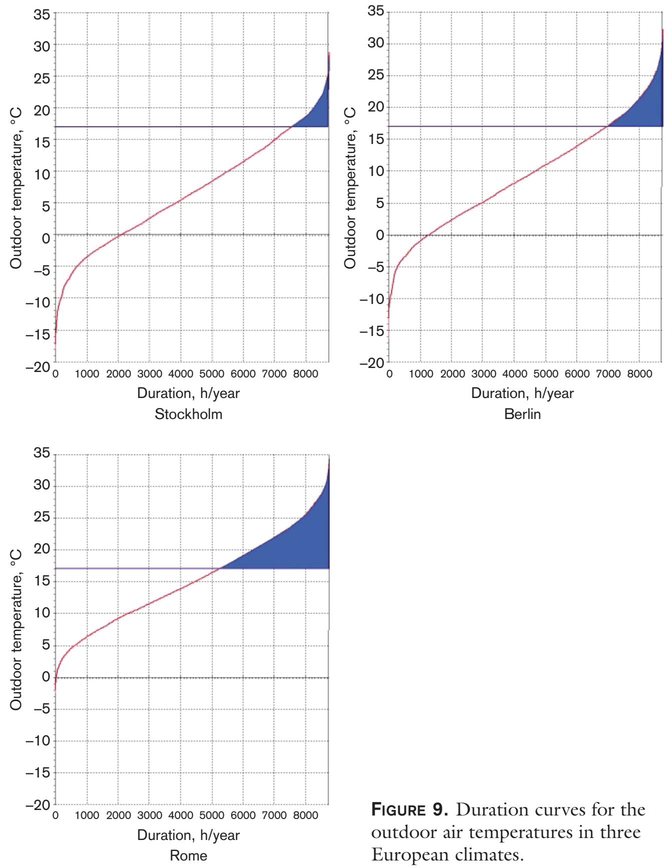


FIGURE 9. Duration curves for the outdoor air temperatures in three European climates.

The use of energy for dry cooling must be added to the annual use of energy for dehumidification of the supply air. This energy use depends on how cooling takes place and, primarily, on the temperature of the cooling coils. In Scandinavian climates, this means that the amount of

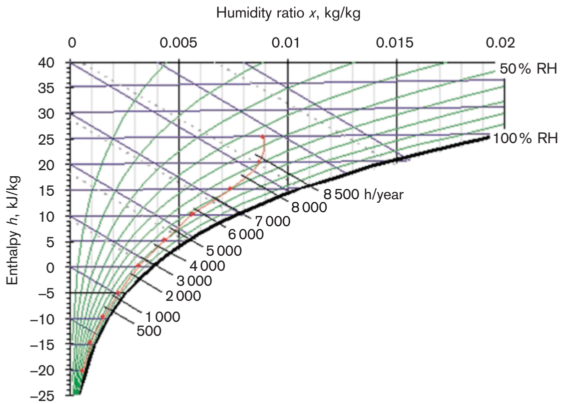


FIGURE 10. The climate curve for a normal year in Stockholm plotted on a Mollier diagram [Bigelius et al., 1983].

energy required for dehumidification in turn means that the dry cooling energy required, when cooling the supply air temperature to 15 °C, has to be increased by about 3 to 18%.

In order to determine the total cooling energy more correctly, the mean states of the outdoor air can be plotted on a Mollier diagram, for example, as shown in Figure 10 for Stockholm. Cooling is required when the states of the outdoor air correspond to a higher temperature than the desired supply air temperature. The cooling process can be plotted for a number of these points on the mean climate curve and the cooling required calculated point by point. Knowing the duration of each point on the mean curve, the cooling energy is then given by the product of the cooling power and the duration of the outdoor air state in question.

In some cases, an even more accurate calculation of the cooling energy might be required. The durations of small areas in the Mollier diagram, for example, squares of 1 °C by 1 g/kg, can then be used. The cooling power is calculated for the central point in each square and the cooling energy required will be the product of the cooling power and the duration of the conditions in the square.

SYSTEMS FOR GENERATING COOLING

This section contains a general overview of the different systems that are used to generate cooling in HVAC systems.

The vapour compression cycle (electrically driven)

An electric motor is normally used to drive the compressor in a vapour compression cycle. In countries where electricity is expensive and natural gas is cheap, the compressor is sometimes driven using a motor powered by natural gas. This is especially advantageous if there is a simultaneous demand for heating, as heat from the exhaust gases and the oil cooling system can be used. An example of this solution is tri-generation, also known as CCHP – Combined Cooling, Heating and Power generation – in which cooling, heating and electrical energy are produced at the same time. As an alternative to using the vapour compression cycle, a heat driven cooling process – the absorption process – discussed below can be used.

The vapour compression cycle for refrigeration is, today, the by far most commonly used cooling method. The method has a long history and dates back to an original patent from 1834, applied for by Jacob Perkins in the USA. The method has been developed over the years but its basic function has remained unchanged and builds on the fact that fluids at different pressure levels have different temperatures. The principle of the vapour compression refrigeration process is shown in Figure 11.

When the pressure of a gaseous refrigerant is raised in a compressor its temperature will increase. After it has reached a suitably high temperature, its heat is released in a condenser and the refrigerant returns to its liquid state. The pressure and, consequently, the temperature of the refrigerant are then lowered as it flows through an expansion valve. Heat is absorbed at a low temperature in an evaporator and the medium returns once more to its gaseous state.

The efficiency of the cooling process can be expressed using the term

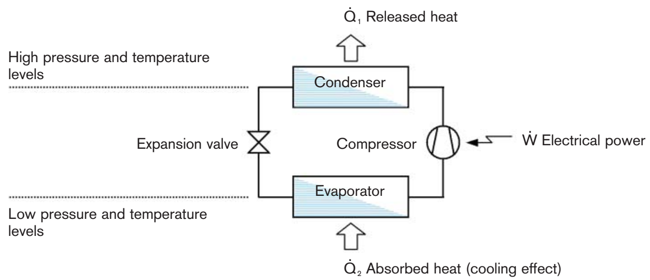


FIGURE 11. A schematic of the principle behind the vapour compression cycle.

COP, Coefficient of Performance, also known as the cooling efficiency factor, which is the ratio of the cooling power obtained to the electrical power supplied. The COP is only valid for a given set of operating conditions and is normally given for design temperature conditions. The efficiency of the cooling unit seen over a whole normal year can be expressed using the term SPF, Seasonal Performance Factor, which expresses the ratio of the annual cooling energy to the supplied electrical energy. This means that the SFP also takes into consideration the efficiency of the cooling unit when run at partial load.

The Coefficient of Performance of a vapour compression refrigeration process, abbreviated to COP_2, where the subscript 2 refers to the cooling side of the process, is defined as the absorbed cooling power divided by the electrical power supplied to the compressor:

$$COP_2 = \frac{\dot{Q}_2}{\dot{W}} \qquad (2)$$

When large refrigeration units are used, the powers of other pieces of electrical equipment, such as pumps and fans in refrigeration circuits and cooling towers, are also included in the equation.

The Coefficient of Performance of a heat pump, abbreviated to COP_1, where the subscript 1 refers to the heating side of the process, is defined in a corresponding way as the heating power released divided by the power supplied to the compressor. By examining the energy balance for a vapour compression process it can be shown that:

$$COP_1 = COP_2 + 1 \qquad (3)$$

In an ideal cyclic process, a so-called Carnot cycle, the COP_2 is only defined using the temperatures at which heat is absorbed and released, i.e. the evaporation and condensation temperatures expressed in Kelvin:

$$COP_2^{Carnot} = \frac{T_2}{T_1 - T_2} \qquad (4)$$

The COP_2 for a real vapour compression refrigeration process can be calculated in a similar way, if it is known to what degree the real process equals an ideal process. This efficiency is called the Carnot efficiency and is defined as:

$$\eta_{Carnot} = \frac{COP_2^{real}}{COP_2^{Carnot}} \qquad (5)$$

The Carnot efficiency of refrigeration units is normally between 40 and 60%. The larger the unit and the higher the evaporation temperature, the higher the Carnot efficiency will normally be.

Figure 12 shows how the COP_2 varies with evaporation and condensation temperatures in a vapour compression refrigeration process with a Carnot efficiency of 50%.

Both in Equation (4) and Figure 12 it can be seen that the magnitude of COP_2 is totally dependent on the temperature difference between the warm and cold side ($T_1 - T_2$). If the evaporation temperature is assumed to be 0 °C and the condensation temperature 20 °C, then the COP_2 will be just below 7.

If outdoor air is used to cool the condenser, then 20 °C is a reasonable condensing temperature in the spring and autumn, while the condensation temperature in the summer can become as high as 30 or 40 °C. In Figure 12 it can be seen that the COP_2 will then be roughly halved to between 4.5 and 3.4. If the refrigeration unit is run as a heat pump with a condensation temperature around 50 to 55 °C, then the COP_2 will be just above 2.5, i.e. the Coefficient of Performance, COP_1, will be just over 3.5.

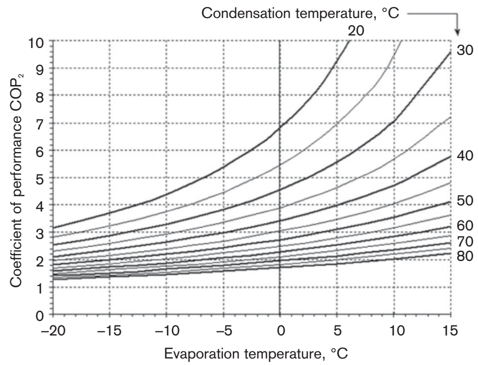


FIGURE 12. The COP_2 in a vapour compression refrigeration process as a function of the evaporation and condensation temperatures. The Carnot efficiency is assumed to be 50%

The absorption process (heat driven)

The first absorption process machine was developed in France in the mid-1800s. The ingeniousness of the process is that it functions similarly to a vapour compression refrigeration unit but, instead of using a compressor to raise the pressure, which requires a great deal of electrical energy, a pump is used, and this requires far less energy. To make the process work, the gas from the evaporator must be somehow converted into a liquid. This is achieved by absorbing the gas in a salt solution in an absorber and then pumping it to a generator. Before the gas can be sent on to the condenser, it has to be separated out in the generator. This is done by greatly increasing the temperature of the solution. The undiluted salt solution then flows back to the absorber to absorb more gas. Figure 13 shows the principle behind the absorption process.

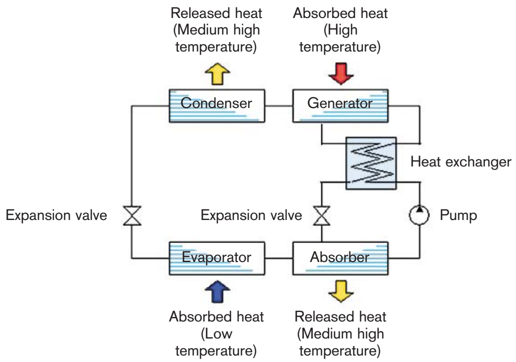


FIGURE 13. The absorption process.

Salt solutions commonly used in absorption processes are water-lithium bromide and water-ammonia. The former is particularly corrosive and requires the use of special materials.

The Carnot cycle COP_2 can also be calculated for an absorption system but, in this case, besides the temperatures in the evaporator and condenser, the temperatures in the absorber and the generator are also included, again expressed in Kelvin.

$$COP_{2\,Absorption\ process}^{Carnot} = \frac{T_{Generator}-T_{Absorber}}{T_{Generator}} \cdot \frac{T_{Evaporator}}{T_{Condenser}-T_{Evaporator}} \qquad (6)$$

The Carnot efficiency can be calculated in a similar way to that of a vapour compression refrigerating unit by comparing a real absorption process to an ideal Carnot process. Even in absorption processes the Carnot efficiency is around 50 %, reaching 60 % in larger and better systems. Figure 14 shows the absorption process COP_2 as a function of the temperatures in the evaporator and the generator. The temperatures in the absorber and condenser are assumed to be 35 °C.

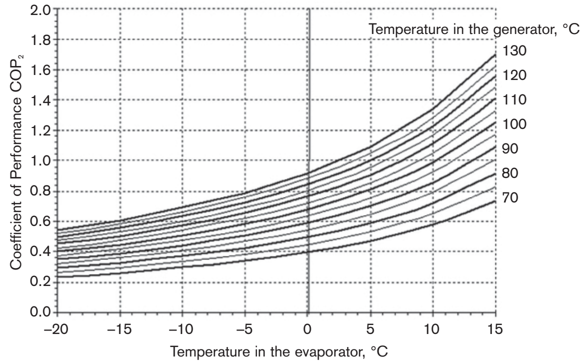


FIGURE 14. The COP_2 in an absorption process. The temperatures in the absorber and condenser are 35 °C and the Carnot efficiency is 50 %.

In Figure 14 it can be seen that at an evaporation temperature of 0 °C and a generator temperature of 70 °C, the COP_2 is not higher than 0.4. If the generator temperature is raised to 115 °C, the COP_2 increases to 0.8. Generator temperatures are normally between 115 °C and 130 °C, which means that the exhaust steam in back pressure turbines/non-condensing turbines can be used to drive the process.

In the same way as in the vapour compression refrigeration process, the COP_2 is greatly dependent on the evaporation temperature. If the cold side of the system can be designed so that the evaporation temperature can be raised using less cooling power, then this will have a positive effect.

The COP_2 shown in Figure 14 is for a so-called single-effect absorption chiller, shown in Figure 13. Double-effect chillers are quite common on the market and, slowly but surely, triple-effect chillers are beginning to leave the research laboratory stage. In a double-effect chiller, the generator is divided into two stages, which also leads to a greater number of heat exchanges than in a single-effect chiller. Triple-effect chillers are even more complicated. Triple-effect single-loop chillers comprise intricate connections between the three generators and the three absorbers while double-loop chillers comprise two cascade-connected single-effect chillers. The first of these requires such high temperatures that other salt solutions than water-lithium bromide have to be used. Multiple-effect chillers also require higher generator temperatures, usually about 190 °C in double-effect chillers and 200 to 230 °C in triple-effect chillers. The COP_2 for a double-effect chiller is 1.2 and for a triple-effect chiller about 1.8.

In Sweden, research is being carried out into the development of absorption process systems for use in district heating systems in which moderate temperatures are used. A trial installation in Gothenburg has attained a COP_2 between 0.7 and 0.8 at generator temperatures between 70 and 90 °C.

As absorption process systems are driven by heat, they are suitable for use in energy processes that produce waste heat at high temperatures or where district cooling is used, i.e. where high-temperature district heating is used to generate cooling energy. This solution is viable in Gothenburg, where adequate supplies of waste heat are available in the summer months.

District cooling

District cooling is available in a number of countries, including some in Europe. In North America, district cooling, as well as district heating, is available on university campuses, in hospitals and in a few towns. In Europe, district heating is commonly used in a number of countries, primarily in Eastern and Central Europe as well as in Denmark, Sweden and Finland.

Just like district heating, it is quite simple for a property owner to install district cooling – only two pipe connections are required. However, to enjoy the benefits of both district heating and district cooling systems, the heating and cooling systems used in buildings must be designed for variable water flows, so that the return temperatures can be low and high respectively.

District cooling is developing strongly in a number of European countries, for example, in Sweden and Finland. Both district heating and district cooling systems are being developed in Central Europe as environmental considerations now require reductions in the number of decentralized systems in single buildings and favour a smaller number of central

plants supplying whole districts or towns. The possibility of being able to generate electricity, district heating and district cooling at the same is also appealing and solutions embracing these three functions have been adopted in parts of Lisbon.

In a country like Sweden, district cooling is generated in a number of different ways. Since the 1980s, as it has been tradition to use giant heat pumps with 1 to 50 MW capacity to generate district heating, it is now possible to use the cold side of the pumps to generate district cooling. Originally, purified waste water was often used as a heat source and subsequent sources have included lake or seawater. Stockholm, the capital city, is said to have the most extensive district cooling network in the world, supplying in 2006 more than half of the country's total district cooling needs of just under 0.7 TWh/year. About 80% of the district cooling generated in Stockholm comes from the naturally cold water pumped up from the bottom of the sea, while the remaining 20% is generated by giant heat pumps that are also used to generate district heating.

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WHY HUMIDIFY – WHY DEHUMIDIFY?

Humidification or dehumidification of indoor air is sometimes necessary when the usage of the building requires the humidity of the air to be kept within narrow limits. This is often the case in museums, printing works and laboratories. In museums, a 40 to 60% relative humidity (RH) range is a typical requirement. An even stricter requirement, say between 45 and 55% RH, might be necessary in special laboratories and this would fall into the category 'very strict requirement'. Normally, no demands are made on the relative humidity of indoor air in residential buildings, offices and other premises and it is therefore quite unusual to install humidification or dehumidification equipment in buildings like these. However, dehumidification of supply air might be needed to avoid condensation on cold surfaces in connection with waterborne comfort cooling, for example, in chilled ceilings.

Sometimes the interval 30 to 70% RH is stipulated as desirable to ensure the comfort of the occupants. Higher relative humidities could contribute to microbial growth and much lower humidities to discomfort, causing dry mucous membranes and chapped skin. These particular limits correspond to about 5 to 12 g of water vapour per kilogram of dry air at 22 °C. The moisture content expressed in this way is called the humidity ratio.

In residential buildings, offices, schools etc, in northern European climates, the relative humidity of indoor air often falls to around 10 to 15% in winter. However, the air is still not humidified as the benefits are not sufficiently strong, i.e. the advantages from a comfort point of view are too small, and the consequences from operating and energy points of view are too excessive. In buildings located in humid climates, for example in Miami, USA, it is seldom dryer than about 30% RH indoors.

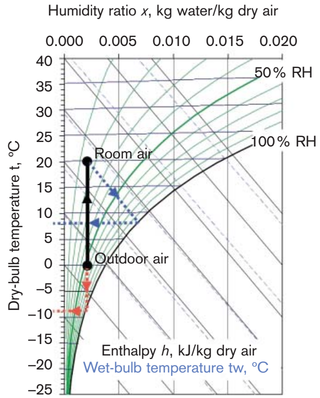
In Nordic climates in the summer, the relative humidity indoors is rarely more than about 60 to 70%. In a building in Miami, the corresponding level would be about 90%, if the air were not dehumidified. This is far above the recommended 70% RH and dehumidification will most probably be needed to prevent problems caused by moisture and mould growth within building structures.

The need for moisture control in buildings is primarily determined by the demands related to their usage. On the other hand, loads created by the outdoor climate can also be significant. In buildings where relative humidity requirements are stipulated, the operating capacities of the humidification and dehumidification equipment mainly depend on the state of the outdoor climate.

OUTDOOR CLIMATE

In northern Europe, the humidity ratio of the outdoor air varies between about 1 and 10 g/kg during the year. It only becomes dryer or more humid than this a few percent of the time. In southern Europe, the corresponding interval is about 2 to 20 g/kg. In Florida, USA, the humidity ratio is rarely below 4 g/kg but can quite often rise to about 20 g/kg. This means that widely different outdoor climate conditions can exist that affect the humidity of the indoor air and the size the equipment needed, if humidification or dehumidification is required.

If the humidity ratio of the outdoor air is, for example, 2 g/kg and its



The black line describes a change in the state of air when it is heated without the humidity ratio *x* being changed.
The blue lines show that the wet-bulb temperature of the room air is about 8 °C.
The red lines show that the dew point is about –9 °C.
Both states of the air have the same dew point as they have the same humidity ratio.

FIGURE 1. A Mollier diagram showing the change in state of air when heated at a constant humidity ratio. The air at the final state, 20 °C and 15% RH, has a dew point of about –9 °C. The energy content of the air, its enthalpy, is 25 kJ/kg and its wet-bulb temperature is + 8.3 °C.

temperature is ±0 °C, then its relative humidity will be about 60%. If the indoor air in a building under these climate conditions is 20 °C, its relative humidity will be about 15%, provided no moisture is supplied or removed within the building itself or by the air handling equipment. This indoor and outdoor data has been plotted on a Mollier diagram, named after its inventor Rikard Mollier (1863–1935), see Figure 1. This type of diagram is used when investigating the changes in state of moist air and to provide the values of important condition parameters, such as enthalpy and wet bulb temperature.

INDOOR MOISTURE GENERATION

A person who is sitting down and engaged in light work will release about 40 g of moisture per hour into the air and about 200 g per hour when carrying out more strenuous physical work. This latter figure corresponds to a work level of 3 met, see Chapter 10/Thermal Climate. If the outdoor air flow rate is 10 l/s per person, this means that the additional amounts of moisture in the air will be in the region of 1 g/kg dry air and 5 g/kg dry air respectively.

Depending on the moisture content of the surrounding air, up to 400 g of moisture per m^2 could evaporate every hour from an indoor swimming pool with a water temperature of 25 °C. Here, the temperature of the water is important: if the temperature of the air is 25 °C and its relative humidity 60%, the amount of evaporating water would be almost doubled, if the temperature of the water were raised from 25 to 30 °C. It is also important to note that evaporation from other wet surfaces, for example the floor surrounding the pool and the bathers themselves can significantly increase the moisture load.

In spaces where there are hygroscopic surfaces, the variations in the moisture content of the air will be smaller, as the material used will adsorb moisture when the humidity is high and release moisture when the humidity is low. A wood-wool building slab is a typical example of such a material.

RELATIVE HUMIDITY IN BUILDINGS WITHOUT HUMIDITY CONTROL

Table 1 shows examples of relative humidities of indoor air in buildings without moisture control or comfort cooling. The table shows, among other things, that in a cold and dry winter climate, such as in Stockholm, the relative humidity can fall to as low as 10%.

On humid summer days in Stockholm, the relative humidity can rise to about 60%. If the temperature is lowered by using comfort cooling, for instance, by using chilled beams, the relative humidity will rise to an even higher level. For example, if the temperature were lowered to 24 °C, without the air being dehumidified, the relative humidity would rise from 60 to about 70%. If cooling coils are used to cool the supply air, the relative humidity in the room will be lowered, as the supply air, besides being cooled, will also be dehumidified. This process is described in more detail below. In a climate like this, the humidity of the room air will almost always be lower than the recommended highest level, as mentioned above (70% RH). In this case, dehumidification, for comfort reasons, will not be required.

It can also be seen in the table that in a building in Rome or Miami, without any dehumidification, the relative humidity would rise to nearly 100%, if the room air were kept at 26 °C. If the room temperature were allowed to rise, for example, to 30 °C, the relative humidity would still be close to 80%. This suggests that there is quite a large need of dehumidification in buildings located in climates like these.

The humidity of the air in a building in Stockholm can be presumed to be lower than 30% for about 3200 hours per year, assuming that the room temperature is 22 °C in winter. In most cases this is acceptable, as the consequences of installing dehumidification equipment would not be reasonable from an operational and energy point of view. In buildings where usage dictates a minimum relative humidity, such as museums and special laboratories, the air must, however, be humidified for a large part of the year. If requirements state that the relative humidity must not fall below the above mentioned limit of about 30%, then humidification will be required for about 37% of the time. The corresponding figure for Rome would be about 2% of the time, i.e. about 200 hours per year.

TABLE 1. Examples of limit values for outdoor air humidity ratios in a number of locations, and for relative humidities in buildings without moisture control or comfort cooling. It has been assumed that the internal generation of moisture is 50 g/h per person and that the supply air rate is 10 l/s per person.

Location	**Humidity ratio of the outdoor air*** g/kg	**Relative humidity of the room air** Winter – at a room temperature of 22 °C	Summer – at a room temperature of 26 °C
Stockholm	0.8 to 11.5	12%	60%
Paris	1.9 to 13.2	19%	68%
Rome	2.3 to 19.1	21%	95%
Miami	4.5 to 20.3	34%	100%

*Climate data: 2005 ASHRAE Handbook-Fundamentals. The lower values of the humidity ratios of the outdoor air represent winter conditions and the high values summer conditions.

In Stockholm, the relative humidity of the air in a building will seldom exceed 70%, even without dehumidification, if the summertime temperature of the room air is assumed to be 26 °C. In a building in Rome, it might be necessary to dehumidify the air for about 600 hours per year, i.e. for about 7% of the time, to prevent the relative humidity from rising above 70%. The room temperature is still assumed to be 26 °C. In Miami's very humid climate, the corresponding figure is estimated to be about 4700 hours per year or 55% of the time.

BASIC HUMIDIFICATION METHODS

Humidification of room air can be carried out by either humidifying supply air in the air handling unit or direct humidification of indoor air. There are two basic methods that can be used: one is based on the introduction of moisture that has been pre-vaporized, i.e. humidification by the use of steam, and the other based on direct vaporization, so-called evaporative humidification. The last three solutions in the list below are examples of evaporative humidification methods.

- The introduction of steam
- Circulating water
- Spray humidification
- Ultrasonic humidification

When water is boiled steam is generated and microorganisms are killed. If the water used contains other pollutants, these can be transported by the steam and contaminate the humidified air. This is why it is important to make sure that the water is clean before the steam is produced.

A simple type of evaporative humidifier can be created by *wetting a large surface area* in contact with the air flow to be humidified. The excess water that is not absorbed by the air is collected in a tray at the bottom of the unit and recirculated by pumping it up to the top of the unit and using it again to keep the surface wet. As the water is used up, more water is supplied to the tray. As the circulating water is neither heated nor cooled but only circulated over the wet surface, the temperature of the water and that of the wet surface will assume the *wet-bulb temperature of the air*. The wet-bulb temperature of the air always assumes a value between the dew point and the dry-bulb temperature of the air.

When evaporative humidification is used the amount of moisture transferred per unit time increases as the temperature of the water increases. To humidify the air, the water temperature must be higher than the dew point, otherwise no water will evaporate. If the temperature of the water is the same as the dew point of the air, the humidification process will cease and the air will only be cooled and its humidity ratio remain unchanged. If the temperature of the water is lower than the dew point, the moisture in the air will condense onto the cold and wet surface.

When so-called *spray humidifiers* are used the water is vaporized with the help of compressed air and a fine aerosol is formed. The efficiency of this method can be increased by spraying the water onto a large surface and keeping it wet. Any excess water can also be recirculated.

If the humidifiers described above, with circulating water, are not very carefully maintained, there will be a great risk of microorganism growth. This could cause problems in the form of smells and even ill-health. Solutions like these should therefore be avoided.

A more efficient version of a spray humidifier can be created if the pressure of the water is raised using a high-pressure pump, which will mean that the aerosol will be more finely atomized. When extremely fine high-pressure nozzles are used the incoming water must be carefully filtered. High noise levels could limit the use of this type of solution when direct humidification is used indoors.

When *ultrasonic humidification* is used small droplets of moisture are formed with the help of ultra sound. Here, too, it is important to make sure that the water is very pure to avoid any damage to the equipment.

HEATING REQUIREMENTS FOR HUMIDIFICATION

Every solution used for humidifying air entails moisture being converted from its liquid phase to its vapour phase. The same amount of energy is required no matter whether the change of state is a result of humidification using steam or if it is achieved by any of the other methods described above, for example by introducing sprays of water droplets or vapour. In the first case, energy is used to boil water and convert it into steam and this is then introduced into the air. On the other hand, when vaporization takes place in the air, the heat required to vaporize the water will be removed from the air, causing its temperature to fall. If this fall in temperature is undesirable, the air will have to be heated. This means that more energy will be required and the amount will equal that used for humidification by steam.

As a consequence of the above, if humidification is required at the same time as the indoor air needs to be cooled, it could be advantageous to consider humidification by allowing cold water to be vaporized in the air, for example, by spraying it with an aerosol, as this will mean that the temperature of the air will also be reduced (this is also known as evaporative cooling).

Steam at 100 °C has an energy content (enthalpy) of about 2700 kJ per kg of steam. If this steam is used to increase the humidity air at 20 °C from 10% RH to 50% RH, the humidity ratio of the air will increase by about 6.5 g/kg and its temperature will increase to about 21.5 °C. If the air flow is 1 m^3/s, this will require a supply of 28 kg of water per hour, equivalent to 8 g/s. The power required to evaporate this amount of water is about 21 kW, which is derived from the product of the flow of water and the enthalpy of the steam. Basic relationships for this type of calculation can be found at the end of this chapter.

To reduce the considerable amount of energy required in connection with humidification, the unit should be fitted with a rotary heat exchanger, or thermal wheel, that is capable of transferring moisture. The rotor in this unit is hygroscopic, which means that moisture is adsorbed by the rotor on the extract air side and is released from the rotor on the supply air side, see Figure 2.

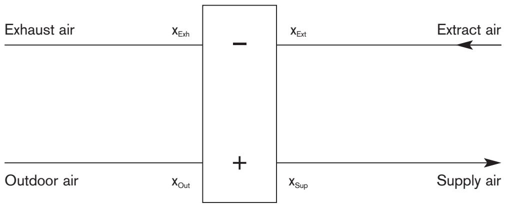


FIGURE 2. The principle behind a rotary heat recovery unit with a hygroscopic rotor. The humidity ratio of the air *x* is given using the indices in Equation (1).

The moisture transferring capacity of the unit is given its so-called humidification efficiency, see Equation 1.

$$\eta_x = \frac{x_{Sup} - x_{Out}}{x_{Ext} - x_{Out}} \qquad (1)$$

where:

η_x is the humidification efficiency of the hygroscopic rotor

x_{Sup} is the humidity ratio of the supply air in $g_{steam}/kg_{dry\ air}$

x_{Out} is the humidity ratio of the outdoor air in $g_{steam}/kg_{dry\ air}$

x_{Ext} is the humidity ratio of the extract air in $g_{steam}/kg_{dry\ air}$

The humidification efficiency is defined in a similar way to the so-called temperature efficiency, see Chapter 21/Energy recovery. At full speed, the humidification efficiency of a hygroscopic rotor is approximately the same as its temperature efficiency and a typical value at full speed is 80%. As the speed of the wheel is reduced, the humidification efficiency decreases at a greater rate than the temperature efficiency.

BASIC DEHUMIDIFICATION METHODS

Dehumidification of air can be carried out either in the air handling unit that supplies the whole building or by using dehumidification units in individual rooms, with the room air being circulated through the dehumidifier. There are two basic principles behind the processes for dehumidifying air:

- Cooling by condensation
- Hygroscopic transfer

Cooling by condensation

Heat energy is released when water in the vapour phase is condensed into the liquid phase. The amount of energy is the same as that bound in the steam when it was formed by the vaporization of water from the liquid phase. When air is dehumidified this so-called condensation energy must be removed by cooling, for instance by using cooling coils in a so-called wet cooler. In this process, the temperature of the cooling surface is so low that the air will condense on it. Dehumidification of supply air can therefore be achieved using different types of cooling coils: direct expansion coils and parallel flow or counter flow cold water coils. As shown below, the changes in state of the air are somewhat different depending on the type of coils used. Counter flow cold water coils provide somewhat poorer dehumidification than parallel flow coils over the same temperature drop in the air. The degree of dehumidification will be greater the colder the surfaces of the cooling coils.

As mentioned above, air can be humidified and cooled by spraying it with an aerosol or letting it flow over a large wet surface. In fact, this same technique can be used to dehumidify air, on condition that the temperature of the water is kept below the dew point of the air. The moisture in the air will then condense on the cold wet surface.

Hygroscopic transfer

A rotary heat exchanger, or thermal wheel, in the form of a so-called hygroscopic rotor, transfers moisture from the air to be dried, for example, from the supply air side to the exhaust air side. The moisture adsorbed by the hygroscopic rotor material is evaporated by an air heater, which greatly increases the temperature of the extract air before it reaches the wheel. The wheel becomes warm and the temperature of the dried air rises significantly as its moisture content decreases. The air downstream of the dehumidifying rotor must therefore be cooled. If this cooling effect is achieved by using evaporative cooling, the rotor is said to be a sorptive heat exchanger.

POWER AND ENERGY REQUIREMENTS FOR DEHUMIDIFICATION

When air is dehumidified by condensation onto a cold surface, as mentioned above, the heat of condensation will be released and must be removed by cooling. If an air flow of 1 m^3/s (about 1.2 kg/s) is to be dehumidified, so that the humidity ratio is reduced by 1 g/kg of air, the rate of condensation will be 1.2 g/s. For each kg of water that condenses the amount of energy released will be 2500 kJ and this will have to be removed by cooling. As 1.2 g of water condenses every second, the required amount of cooling power to achieve the desired reduction of humidity ratio by 1 g/kg will be 3 kJ/s, i.e. 3 kW.

As the process also entails the air being cooled, the cooling power required will be much greater than this, as illustrated in Figure 3. If the air to be dehumidified has a temperature of 20 °C and a relative humidity of 70% and if cooling is achieved across a surface at 5 °C, the temperature of the air will fall by about 4 °C for every g of water per kg of air that the humidity ratio of the air decreases. This reduction in temperature of an air flow of 1 m^3/s is equivalent to a cooling power of about 5 kW. These figures indicate that a total of about 8 kW of cooling power will be needed to dehumidify the air. If the air temperature is then raised to 20 °C, the air will have to be heated by 4 °C, which will require about another 5 kW.

Quite often, the need to dehumidify coincides with the need to reduce the temperature of the air, as the relative humidity of the outdoor air is high when its temperature is high, i.e. in the summertime. It is then possible to say that the dehumidification process is, in fact, included in the price of an airborne comfort cooling solution. The relationship between the degree of dehumidification and the reduction in temperature can be changed by connecting the cooling coils in different ways (parallel flow or counter flow) and by adapting the temperature of the chilled surfaces (changing the coolant temperature). As mentioned above, the counter flow coils provide a somewhat greater degree of dehumidification than parallel connected coils and the degree of dehumidification will be greater the lower the temperature of the coolant.

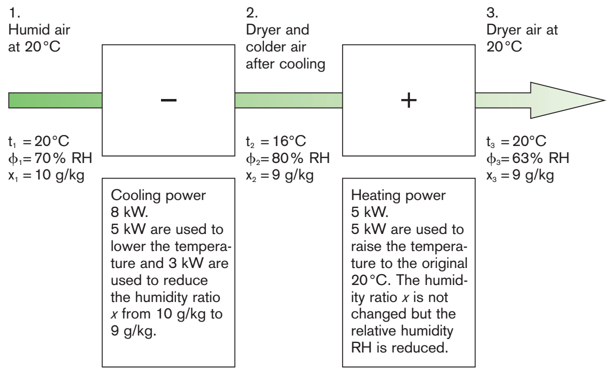


FIGURE 3. The principles behind dehumidification by cooling and heating air. The example assumes that the air is cooled by a surface at a constant temperature of +5 °C and that the air flow is 1 m^3/s. The method for plotting the change of state in a Mollier diagram, when cooling on a surface with constant temperature, is shown in Example 3 at the end of this chapter.

CALCULATION AIDS

A Mollier diagram is used to study how the thermodynamic state of the air changes during different processes, for example, humidification. This was illustrated in Figure 1. There is also an American/British version of this diagram, the so-called psychrometric chart, shown in Figure 4. Both diagrams can be used in exactly the same way – the only difference is that the axes have been transposed.

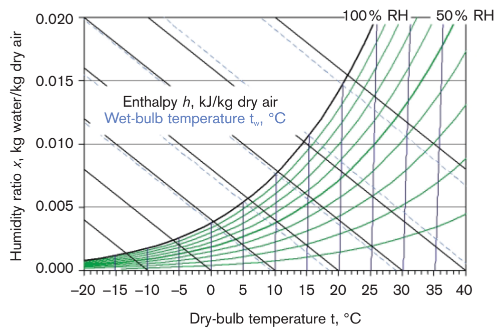


FIGURE 4. Psychrometric chart.

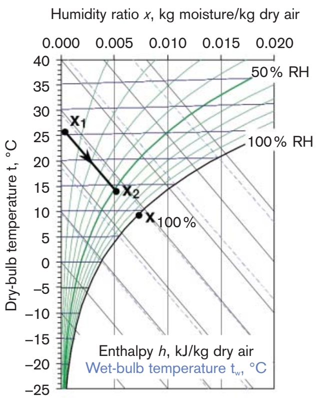
EXAMPLES OF HUMIDIFICATION AND DEHUMIDIFICATION PROCESSES

In the following, Mollier diagrams are used to provide an overview of the principles behind the different processes that take place when the humidity of air is changed. How to calculate the power required to change of the state of the air is also shown. In all examples, the state of the air before the change has a subscript ‘1’ and the state of the air after the change a subscript ‘2’.

The following symbols and units are used:

Symbol	Description	Unit
$\dot{V}$	air flow	m^3/s
$\dot{M}_v$	water flow in cold water circuit	kg/s
$\dot{M}$	water usage (evaporation)	kg/s
$h_ä$	enthalpy of steam	kJ/kg
h_1	enthalpy of the air before change of state	kJ/kg
h_2	enthalpy of the air after change of state	kJ/kg
x_1	humidity ratio of the air before change of state	kg_w/kg_a
x_2	humidity ratio after the change of state	kg_w/kg_a
t	temperature	°C
c_{pa}	specific thermal capacity of air	kJ/(kg · °C)
c_{pw}	specific thermal capacity of water	kJ/(kg · °C)
ρ	density of air	kg/m^3

Example 1. Evaporative humidification.

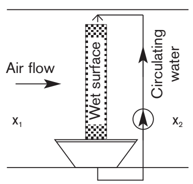


The efficiency of the evaporative humidifier is defined as:

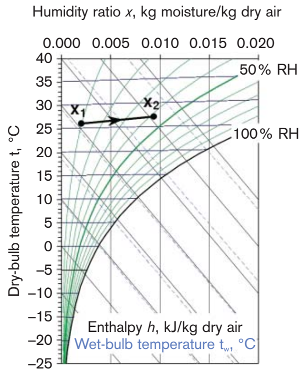
$$\eta = \frac{x_2 - x_1}{x_{100\%} - x_1}$$

The amount of water required is calculated using the following equation:

$$\dot{M} = \dot{V} \cdot \rho \cdot (x_2 - x_1)$$



Example 2. Humidification using steam.

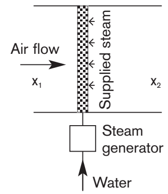


The amount of water required is calculated using the following equation:

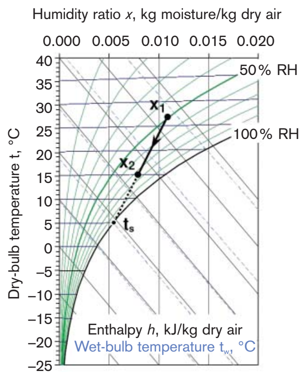
$$\dot{M} = \dot{V} \cdot \rho \cdot (x_2 - x_1)$$

And the power requirement for steam generation using:

$$\dot{Q} = \dot{M} \cdot h_ä = \dot{V} \cdot \rho \cdot (h_2 - h_1)$$

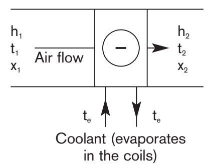


Example 3. Dehumidification using direct expansion cooling.



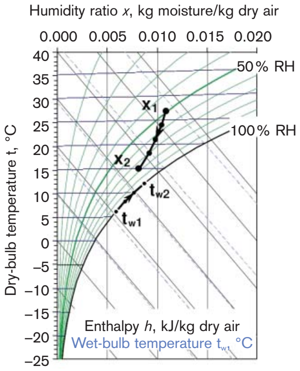
The cooling power is calculated using the following equation:

$$\dot{Q} = \dot{V} \cdot \rho \cdot (h_1 - h_2)$$



The temperature of the cooling surfaces in the direct expansion coils is t_s

Example 4. Dehumidification using condensation cooling (counter flow connected coils).



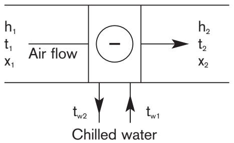
The cooling power is given by:

$\dot{Q} = \dot{V} \cdot \rho \cdot (h_1 – h_2)$ on the air side

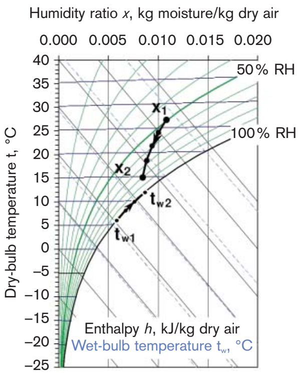
and

$\dot{Q} = \dot{M}_v \cdot c_{pw} \cdot (t_{w2} – t_{w1})$ on the water side.

The magnitudes of these two power requirements are the same.



Example 5. Dehumidification using condensation cooling (parallel flow connected coils).



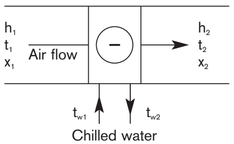
The cooling power is given by:

$\dot{Q} = \dot{V} \cdot \rho \cdot (h_1 – h_2)$ on the air side

and

$\dot{Q} = \dot{M}_v \cdot c_{pw} \cdot (t_{w2} – t_{w1})$ on the water side.

The magnitudes of these two power requirements are the same.



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25. FANS AND SFP, SPECIFIC FAN POWER

GUNNAR BERG Development Engineer, Swegon AB

INTRODUCTION

The purpose of a fan is to create a flow of air and it is this parameter that has the greatest influence on the power requirements and use of electrical energy in an air conditioning system. The net power requirement can be calculated by multiplying the air flow volume by the total pressure rise in the system. In order to determine the energy use, the operating time must also be taken into account. An effective way of reducing energy costs is to investigate the actual air flow requirements and optimize operational times for fans. Energy savings, however, must never be made at the cost of the quality of the indoor air. The most effective way of ensuring the quality of the indoor air and reducing the use of energy is to regulate air flows according to demand.

The Specific Fan Power, SFP, is a measure of how much power is required by a fan to transport a given volume of air and is expressed in kW/(m³/s). By stipulating a maximum SFP, it is possible to limit the power requirements for transporting air for a fan, for an air conditioning unit or for an entire building. The consequences of this limit value and other requirements, for example, concerning heat recovery, must be considered jointly to provide an overall picture of how energy is used. If the stipulated SFP is too low, this could cause problems, as the air conditioning system will then be more susceptible to external disturbances, for example, due to wind loads. An SFP_V rating (V stands for validation according to SS-EN 13779) for an air conditioning unit can and must always be checked after installation to verify that requirements have been met, as the design of the air conditioning unit and ducting system are based on them being correct.

THE FUNCTION OF A FAN

The primary function of a fan is to create a specified air flow, normally expressed as a volume rate of flow in m³/s, but also expressed as a mass flow in kg/s, for example, in process industries. In other instances, for example, where fans are used for pneumatic transport, the air speed through a particular cross-section is specified. Depending on the ventilation system in which it is installed, the fan will have to create either a high or a low total pressure rise. The total pressure rise is the sum of the static and dynamic pressure rises. The static pressure is created by the irregular omni-directional movements of air molecules, resulting, for example, in the same resistance being experienced when pressing any part of a balloon. The dynamic pressure, due to the speed of the air, is created by the molecules moving in a certain direction and, for example, causes a tree to bend in the direction of the wind. Static pressure can be converted into dynamic pressure and dynamic pressure can be reconverted to static pressure. The total pressure, however, is the same, if conversion losses are disregarded. The effective capacity of a fan can be calculated from the following expression:

$$\text{Effective capacity in W} = \text{Air flow in m}^3/\text{s} \times \text{Total pressure rise in Pa}$$

The power that is supplied to a fan can be expressed in two different ways:

- As the power to the impeller (i.e. the mechanical power transferred to the impeller), used when choosing a suitable motor
- As the electrical power supplied to the motor (including the electrical power to the frequency converter or – if not fitted – directly to the motor), used when calculating energy use. The electrical power includes the power to the whole drive system, from the mains supply via the frequency converter, motor, belt drive and bearings to the impeller. The subsequent use of electrical energy (power x operating time) is what has to be paid for when the fan is in operation.

The efficiency of the fan is defined as the effective capacity divided by the supplied power. The following relationships can therefore be derived:

$$\text{The impeller efficiency in \%} = \frac{\text{Air flow in m}^3/\text{s} \times \text{Total pressure rise in Pa}}{10 \times \text{Mechanical power supplied to the impeller in kW}}$$

$$\text{Total efficiency in \%} = \frac{\text{Air flow in m}^3/\text{s} \times \text{Total pressure rise in Pa}}{10 \times \text{Electrical power supplied to the fan in kW}}$$

If a complete fan, including its motor, is placed in an air stream, then the temperature of the air, after passing through the fan, will rise to a level corresponding to the *total* power supplied to the motor. This is a welcome side effect in cases when the air has to be heated anyway, for instance, after passing a supply air fan during the heating season.

Another side effect is that fans generate noise. Sound attenuators are often needed in ducting systems and the impeller, or the whole fan unit, might also have to be encased to achieve acceptable noise levels in occupied rooms. This will require extra investments and possibly more use of electrical energy to compensate for pressure drops across attenuators.

In some situations, it might be necessary to determine the new operating point when the speed of a fan is changed. This is where the affinity laws for fans can be used. These laws are valid when the total pressure rise created by the fan is less than 2000 Pa (incompressible flow, i.e. the compression of the air by the fan is negligible). Additionally, the laws

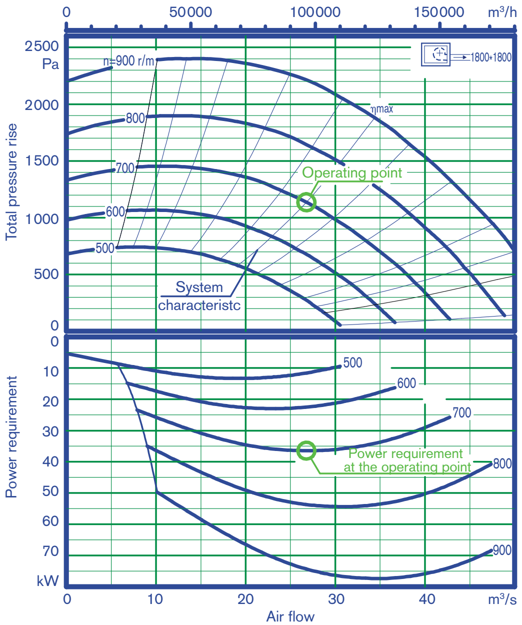

FIGURE 1. Fan characteristics showing a specific operating point.

must be applied along a system characteristic (the lines along which the pressure varies with the square of the air flow), see Figure 1. A fan will operate uniformly along a system characteristic and the fan efficiency will remain constant.

The affinity laws:

$$\text{Air flow} = \text{Original air flow} \times \frac{\text{fan speed}}{\text{original fan speed}}$$

$$\text{Total pressure rise} = \text{Original total pressure rise} \times \frac{(\text{fan speed})^2}{(\text{original fan speed})^2}$$

$$\text{Power} = \text{Original power} \times \frac{(\text{fan speed})^3}{(\text{original fan speed})^3}$$

FANS IN AIR CONDITIONING UNITS

To satisfy the requirements specified in different applications, a number of different types of fans have been developed and these, in turn, can be classified using a number of sub-groups and versions. In this chapter, the discussion is limited to fans used in air conditioning units.

The most common type of fans used today are centrifugal fans with double inlets, backward curved blades and spiral casings, as shown in Figure 2, and plug fans, i.e. centrifugal fans with backward curved blades without spiral casings, as shown in Figure 3.

FIGURE 2. A double-inlet radial fan.

FIGURE 3. A plug fan.

Traditionally, centrifugal fans with double inlets and forward curved blades and, to some extent, axial fans have also been used. Both of these offer compact solutions, as they can deliver high air flows despite their small size. This means high outlet velocities, a high proportion of dynamic pressure and a great risk of pressure losses, so-called system losses. The use of fans with forward curved blades has fallen, as these are less energy-efficient.

When a fan is encased in a housing there will be pressure losses and increased noise. These effects must be taken into account when looking at performance but, unfortunately, this is not always the done.

Many ventilation equipment manufacturers only state performance data for fans when they are not subject to a load, i.e. data that was supplied by the fan manufacturer. Figure 4 shows how the intake air flow pattern for an enclosed fan with double inlets differs from that of a similar unenclosed fan.

FIGURE 4. Air flow patterns around an enclosed and unenclosed centrifugal fan with double intakes.

Centrifugal fans with double intakes are the most suitable choice when relatively high outlet velocities (>8 m/s) are required and the ducting is straight, without any restrictions caused by changes in cross-sectional area, or by dampers or attenuators, on the output side of the fan. Conditions like these are most common in large ventilation plants. This type of fan, with backward curved blades, offers the highest fan efficiency. Centrifugal fans with double inlets are normally belt driven but this also means that they will require maintenance and will create dirt and belt dust.

Plug fans are used where the ducting is designed for low velocities (<6 m/s) and small pressure drops. The ducting after the fan can be designed without limitations – the fan is said to be of "wide-tolerance" type. However, flows must always be considered from a practical point of view, so that low pressure drops can be avoided. The impeller is normally mounted directly on the motor shaft and the motor requires a frequency converter to adjust the operating point. The frequency converter makes it possible to maintain a constant flow of air or to meet different demands, for example for day/night operation or for VAV (Variable Air Volume) systems. Plug fans are very easy to access for cleaning.

AIR FLOWS, OPERATING TIMES AND PRESSURE DROPS

The electrical power requirement of a fan and its use of electrical energy in an air conditioning system depend on the air flows, total pressure rise and operating time. Consequently, the functional requirements specified for the fan will affect the air flows, the indoor temperature and noise levels. These factors will, in turn, affect the pressure drops in air heaters and air coolers, and the pressure drops in the attenuators. Furthermore, the size of the fan, the type of drive and motor, the system components (dampers, heat recovery systems, etc), ducting dimensions and available space for the plant will all have an affect on the use of electrical energy. The air flow and operating time have, however, the greatest effect. The air flow affects the power required both directly and indirectly, as the pressure drop depends on the air flow for a given size of plant and given dimensions of the ducting. If the air flow increases by 10%, the power required will increase by about 30% (in a VAV system at constant pressure the power required will increase by about 20%). Very low air flows are undesirable, both for a building itself and its occupants, while high air flows will require a greater use of energy. It is therefore essential that both the design and measured air flows are correct. Today, most fans used in air conditioning plant can be fitted with reliable flow meters.

Operating times have a direct affect on the use of energy. When operating times are reduced, there is a corresponding reduction in the variable costs. Energy savings, however, must never be allowed to impair the quality of the indoor air. The best way to ensure the quality of the indoor air and to reduce energy costs is to regulate the air flows according to demand. The demand is governed by a number of factors: the number of people in the building, the use of the building, whether it is summer or winter, day or night, and the orientation of the building. A change in the air flow also affects the energy costs for heating and cooling the supply air and indoor air.

The pressure drop across an air conditioning unit depends, to a large extent, on the size of the space available for the equipment and the choice of components. If plenty of space is available, a larger unit can be installed and there will be fewer problems when installing the ducting. If a heat recovery system is chosen, the pressure drop will increase as will the amount of energy required to transport the air. In most cases, however, this will mean a substantial reduction in the power requirements and use of energy for heating and cooling. Low velocities in the ducting also result in lower pressure drops.

LOSSES IN ROTARY HEAT EXCHANGERS

Rotary, regenerative heat exchangers are the most temperature-efficient heat recovery units. However, they are not completely airtight, which means that steps must be taken to ensure that contaminated extract air is not allowed to leak into the supply air. This can be accomplished by ensuring that the pressure balance in the air conditioning unit is such that any leakage takes place from the supply air side to the extract air side, i.e. by ensuring that the pressure on the extract air side is lower than on the supply air side. If the pressure balance is incorrect, then an extra pressure drop must be created on the extract air side. Another potential source of contamination of the supply air by the extract air is the small amount of air that is trapped in the impeller and carried over from the warm extract air side to the cold supply air side. This trapped air mass must be replaced by clean outdoor air – the so-called venting or purging flow – in a venting or purging section placed between the supply air and the extract air. As shown in Figure 5, the venting flow is created on the outdoor air side and is evacuated with the extract air. When specifying the extract air fan the extra pressure drop required to ensure the correct leakage direction must be taken into account. The flow through the extract air fan is also increased due to the leakage and venting flows (the magnitude of this increase depends on the pressure conditions and seals, and is stated by the

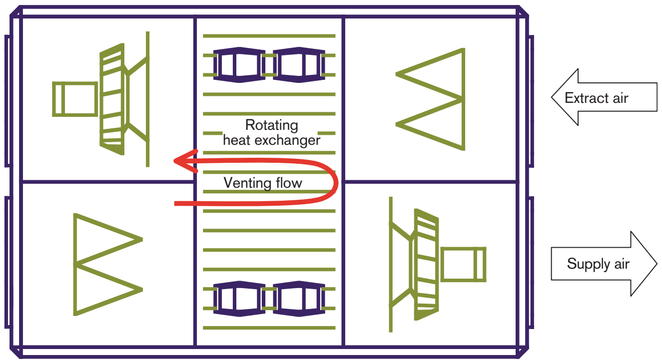

FIGURE 5. Air conditioning plant with a rotary heat exchanger.

supplier. It is often more than 10% of the extract air flow). These compensatory flows are described in EN 13053 and SS-EN 13779.

POWER LOSSES

The energy used by a fan depends on the efficiency of the motor and what is included in the drive system, i.e. if direct drives or belts are used, and whether a frequency converter has been installed.

A fan with direct drive, in which the impeller is mounted on the motor shaft, has no transmission losses. If the fan is belt-driven, the transmission losses can be calculated from the drive efficiency, which depends on the type of belt used (V-belt, flat belt, etc), and the size of motor. V-belts are the most common type of belts. A motor rated at 1.5 kW will have an efficiency of about 93% while a 30 kW motor will have an efficiency of about 96%. The efficiency of the motor will depend on its type, size and class. The most common type of motor on the market is the asynchronous motor but, thanks to technical advances, permanent magnet motors have become more popular. Permanent magnet motors are always controlled electronically. They have high efficiencies over the whole of their operating ranges, i.e. from low to high loads and from low to high speeds. Asynchronous motors are robust and can be connected directly to the power network but they have somewhat lower efficiencies, especially when operating at low speeds. In the 1990s, the EU and the European manufacturing organization CEMEP produced a joint classification system for motor efficiency ratings. Standard motors are classed as EFF2 and high efficiency motors are classed as EFF1. The efficiency of a four-pole 1.5 kW motor is at least 78.5% in class EFF2 and at least 85% in class EFF1. Corresponding figures for 30 kW motors are 91.4% and 93.2% respectively.

Today, it is quite common for fans to be fitted with frequency converters to regulate their speeds, for example, when demand-controlled air flows are required. When calculating the efficiency of a fan with a frequency converter, the effect of the frequency converter on the motor losses as well as the losses in the electronic filters, needed to minimize electromagnetic interference, must also be included. The nominal efficiency of a single-phase frequency converter, with an active PFC (Power Factor Correction), is about 95 % and about 97 % for a three-phase converter. To a reasonable degree of accuracy, the losses in a frequency converter, in absolute figures, can be regarded as being constant no matter what the operating speed, which means that the following formula can be used for calculations along a system characteristic:

$$\text{Electrical power} = \text{Reference power} \times \frac{\text{shaft speed}}{\text{reference shaft speed}} + \text{power losses}$$

where:

Reference power = Measured (or calculated) electrical power at the reference shaft speed – power losses

Figure 6 shows the efficiency of a frequency converter at different relative shaft speeds when the motor, operating at full load at a relative shaft speed of 100%, has an efficiency of 80%.

The efficiency of the frequency converter, which falls as the relative shaft speeds fall, does not cause any problems as such, as the power required falls at an even faster rate. However, this does put a limit on how low the SFP can be for an air conditioning unit. A frequency converter of reasonably good quality will have a high power factor (small proportion of harmonics) and a phase shift close to 1, i.e. the undesirable reactive effect caused by electric motors will be minimized.

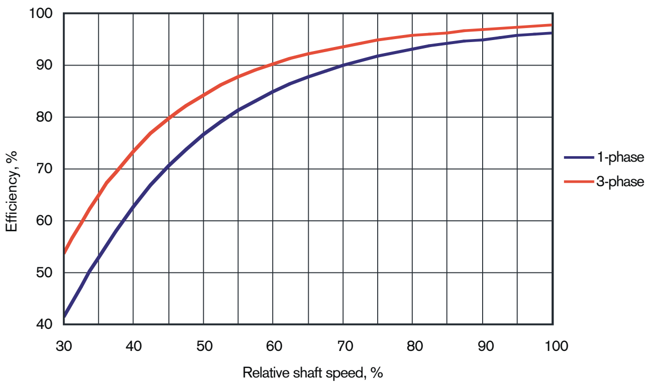

FIGURE 6. The efficiency of a frequency converter as a function of the relative shaft speed.

SPECIFIC FAN POWER

The Specific Fan Power, SFP, is the power that has to be supplied to a fan so that it can create a flow of 1 m^3 of air per second. The SFP rating is given in kW/(m^3/s). By specifying a maximum rating, it is possible to place an upper limit on the power made available to a fan, an air conditioning unit or even a whole building. An important consequence of this is that both the equipment chosen and the flow system solutions will be included in the evaluation of the performance of the fan and that the specified rating can be checked.

The disadvantages of specifying an SFP rating can be seen when evaluating alternative system solutions (constant flow as opposed to variable flow) and when heat recovery is used. In this latter case, the use of energy for heating and/or cooling can be reduced by choosing a recovery system with a higher temperature efficiency and, in turn, a higher pressure drop. Consequently, this type of solution would require a *higher* SFP rating. It should be noted that if cooling recovery is required, the total use of electrical energy might be reduced when the SFP rating is increased, as less power has to be supplied to the refrigeration compressor.

Unfortunately, the use of electrical energy for building services is not always considered as a whole. To ensure that the grounds on which decisions are made are correct, and to avoid sub-optimisation, it is recommended that a thorough analysis, in which all available solutions are included and evaluated, be carried out.

The SFP rating is primarily defined as follows:

$$\text{SFP rating in kW/(m}^3\text{/s)} = \frac{\text{Power supplied from the mains in kW}}{\text{Air flow in m}^3\text{/s}}$$

It is easy to convert the formula so that the rating can be calculated from the total pressure rise across the fan divided by the total efficiency of the fan. It is then quite clear that the SFP rating primarily depends on the location of the air conditioning unit, as most of the pressure drops occur in the unit itself (at least if energy recovery is used) and, if a lower SFP rating is required, the size of the unit will have to be increased.

Very low SFP ratings and pressure drops in equipment and ducting can also lead to problems, as the system will become more sensitive to external disturbances such as wind, or to doors and windows being left open or closed. Furthermore, filters will have to be changed more frequently and it might also be necessary to clean the extract air ducts more often, if pressure drops are to be kept as low as possible. The air in an extraction duct is not filtered and, if velocities are low, the air will not be able to remove particles and dust, allowing them to precipitate in the ducting system.

DIFFERENT DEFINITIONS OF SFP RATINGS

As mentioned previously, SFP ratings can be defined within different system boundaries. The Swedish Indoor Climate Institute proposed the following: *An SFP rating is the combined total fan power of the supply and extract air system fans when operating at a design air flow equivalent to the total air flow through a building.* The industrial association Swedish Ventilation reduced the scope of the definition to apply to individual fans and air conditioning units by using the notation SFP_V. The subscript V was originally derived from the name of the association but in EN 13779 it stands for 'validation'. Contracts with suppliers on the Swedish market are normally based on this definition. Furthermore, agreement was reached regarding the prevailing conditions under which calculations are to be made, e.g. with regard to air flows (the larger of the supply and extract air flows), filters (clean), cooling batteries (dry) etc, and to rotary heat exchangers (including the extra pressure drop to ensure the correct leakage direction and the increase of the exhaust air flow due to the leakage and venting flows). To facilitate control and validation it is assumed that the filters are clean and the working conditions dry.

A working group within Eurovent, the European industrial organization, has created another rating, SFP_E (the subscript E originally referred to Eurovent, but the word 'evaluated' is also appropriate) based on the design pressure drops for filters and a mean value of the drops in cooling coils (both dry and wet) etc. The SPF_E rating corresponds more precisely to the SFP rating for a whole building than the SFP_V rating, but is more difficult to check. All these ratings have been deemed relevant and are included in Appendix D in EN 13779.

VERIFICATION

Unlike the fan efficiency, which can only be measured and checked in advanced air testing laboratories, the SFP_V rating can be easily verified. Verification of the SFP_V rating should be carried out to a far greater extent than is the case today, as the rating is used when selecting the size of the air conditioning units. A competitive edge can be gained if lower ratings are specified (smaller and cheaper fans and units can be chosen) and every client should demand that checks be made to ensure that requirements are met. The filters in the unit must be clean when the checks are carried out.

Temperatures, flow volumes, external pressure drops and active power must be measured in the prevailing air flows. Measurements made during periods of very high or very low pressure should be avoided, unless the barometric pressure can be accurately determined. Otherwise it will be necessary to allow for a larger uncertainty when calculating the density of the air.

Nowadays, it is nearly always possible to carry out flow measurements for fans after they have been installed in air conditioning units. Systematic errors as low as 5% are common. Air flow measurement methods for fans are normally more accurate than the methods used for measurements in ducting systems. When measuring air flows in ducting systems, the temperature must be measured not only at the same cross-sectional point as for the air flow measurement but also at the fan inlet, to ensure that the temperature difference does not exceed 15 K. Otherwise, the results will be directly affected, if the fan speed has not been temperature compensated.

Active power must be measured directly using a wattmeter and not indirectly by measuring voltage and current. If a frequency converter is used, the measurements must be carried out on the power input side using a TRMS, true root mean square, instrument with a bandwidth of at least 1 kHz.

WARNING!
NEVER CARRY OUT MEASUREMENTS BETWEEN
THE FREQUENCY CONVERTER AND MOTOR.
THIS COULD BE LETHALLY DANGEROUS!

The operating conditions for the calculated SFP_V rating are often different to those when the rating is checked. It is therefore recommended to recalculate the rating according to the prevailing conditions. The measured electrical power should then be recalculated for an air density of $\rho = 1.2$ kg/m^3.

Manufacturers, whose products are certified by Eurovent, must guarantee certain performance data, for example, regarding air flows and power, i.e. the data used when calculating the SFP_V rating. Measurement errors need only be taken into account when comparing measured and declared ratings.

REFERENCES

EN 13053:2006. Ventilation for buildings. Air handling units. Ratings and performance for units, components and sections.

EN 13779:2007. Ventilation for non-residential buildings – Performance requirements for ventilation and room-conditioning systems.

26. SOUND AND SOUND ATTENUATION

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This chapter is a supplement to Chapter 11/Building acoustics in which the theoretical background, quantities and units used in acoustics are explained. This chapter can also be seen as a supplement to Chapter 19/Ducting systems, which discusses the importance of designing and installing functional ducting systems, how they are integrated into buildings, the quality demands made and how airtightness is essential to function and operating costs.

This chapter takes a look at system components, primarily fans, that can create noise and how different attenuation measures can be taken, how ductwork and components create and dampen noise and how ducting can convey sound from one room to another, so-called crosstalk.

Noise is usually regarded as one of the factors threatening our living environments. In our mechanized society, we are surrounded by numerous sources of noise and, unfortunately, they seem to be ever increasing in number and strength – noise from traffic outdoors and from installations indoors. Silence is becoming an increasingly scarce commodity. When designing and building HVAC systems, to provide us with better thermal indoor climates and better air quality, it is important to realize that this must not be done at the cost of polluting the indoor environment with unwanted noise.

Many of us experience a feeling of relief when the ventilation system shuts down at the end of the working day and silence reigns once more. This type of dissatisfaction with noisy ventilation systems must be avoided and it is important that as much care is taken when planning the acoustics in a system as when tackling factors affecting our well-being and comfort.

Silence – the absence of noise – is now often a rarity and this can lead to stress and discomfort. It is also important to remember that prevention is better than cure – reengineering is a more difficult, more expensive, more time-consuming and more troublesome way of reaching acceptable solutions. In addition, it is more difficult to persuade those who have already been disturbed that they should now be happy with a new solution.

HVAC equipment, and especially fans, pumps and compressors, is the dominating source of noise in a building and it can even disturb people in the nearby surroundings. It is therefore important that all equipment is selected and located in such a manner that noise emitted disturbs neither building occupants nor neighbours.

Noise created by a component in an installation, for example, a fan, is transmitted through a building in different ways: via walls, floors and leakage points to adjacent rooms and via the supply and extract ductwork to the rooms connected to the ducting.

Vibrations from equipment can also cause structure-borne sound to be propagated through a building. These vibrations can cause walls, floors and other installations, such as piping, to vibrate and thereby create airborne noise. This, in turn, can cause disturbing noise in rooms a long way from the plant room.

INSTALLATIONS AS SOURCES OF NOISE

Building services installations for ventilation, heating, cooling and sanitary purposes have a common feature: the noises that they create come from flowing media – air, water or coolants. This applies to fans, air terminal devices, ducts, pumps, pipes, valves, compressors and flushing toilets. In every case, the amount of noise created is not only determined by the speed of the media and the pressure drop across the components but also by how well the components were originally designed from a noise point of view.

FANS AS SOURCES OF NOISE

Ventilation systems in buildings are often regarded as noisy and Figure 1 illustrates how noise can be spread.

a) Vibrations can cause structure-borne sound.
b) Airborne sound can be transmitted via the inlet to the fan and via the outlet ducts into the building.
c) Airborne sound can spread from the fan room to adjacent rooms.
d) Noise can be created in ductwork, dampers and terminal devices.

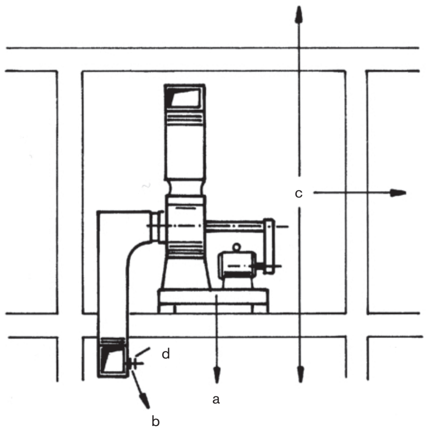

FIGURE 1. Fan noise can spread in different ways and in different directions.

Airborne sound, however, is not the only type of noise arising from a fan. The fan can also cause vibrations in the building structure, 'a' in Figure 1, if it is not statically and dynamically balanced, mounted on a correctly dimensioned and vibration-free foundation and connected to the ducting system via anti-vibration mounts. Power cables to the fan and any drainage pipe connections must also be flexible. This means that there must be a total avoidance of rigid vibration bridging between the equipment and the structure of the building. Otherwise, the other sound attenuation measures will be hardly meaningful.

If vibrations are transmitted to the structure of a building, these will create structure-borne sound that will be able to create new airborne sound in other rooms. Once structure-borne sound has occurred there is nothing that can be done, other than to remedy the source of vibration and inhibit any vibration bridging. It is not sufficient to count on the effects of other measures, as the system contains too many variables and is therefore completely indeterminate.

If the fan room has hard surfaces and, consequently, long reverberation times, see Chapter 11/ Building acoustics, the noise from the fan will result in the sound level in the room being too high and the airborne sound in the fan room, 'c' in Figure 1, will disturb surrounding rooms – above, to the side and below – as the walls and floors between them will start to vibrate and thus create new airborne sound in the adjacent rooms.

To prevent the fan noise from disturbing adjacent rooms, the walls, floors and doors must have high sound reduction indices, see Chapter 11/Building acoustics. And, as the airborne sound easily passes through small gaps and cracks, the penetration points for pipes, cables and ducts through the walls must be well sealed. The fan room doors must also be fitted with rubber sealing strips.

The most efficient and easiest way to avoid the problems described above is to locate the fan room as far as possible from sound-sensitive rooms and plan the building so that the fan room is adjacent to store rooms, corridors and other similar spaces where there are no permanent workplaces. If the fan room is located in the basement of a building, this is usually easy to arrange. It is considerably more difficult and demands much more care, if the fan room is located in a loft above the highest floor level. Top floor offices, for example, are usually the most attractive and this is where demands regarding low noise levels are the most stringent.

It is also important to have a sufficiently spacious fan room, so that the ductwork can be connected in the most suitable way possible from a flow point of view, i.e. without sudden bends or high flow speeds, and so that sound insulation material and silencers can be fitted correctly.

Back to Figure 1 – the noise from the fan will also propagate to the ducting system, 'b' in Figure 1, and this can cause high sound levels in ventilated rooms closest to the fan, as the noise will not have had a chance to attenuate in the ducting. At greater distances from the fan, the fan noise is reduced by the ducting system and this is where secondary sources of sound, from dampers and terminal devices, 'd' in Figure 1, will dominate.

Choice of fan

A low self-noise level is an important criterion when specifying and choosing equipment. Fans should be chosen so that they can operate at high levels of efficiency within their normal operating ranges. Fans that are made to run at unsuitable operating points, with subsequent poorer efficiencies, are often noisier than those that have been chosen correctly.

In CAV (constant flow) systems, fans should be chosen so that their maximum efficiencies are at the design air flows. In VAV (variable flow) systems, fans should be chosen so that they can operate with optimal efficiencies and stability in the most frequently used working ranges. A correctly chosen and installed fan reduces the need for noise attenuation in the ducting system. The following points should be kept in mind:

- Design the systems – the ducts, terminal devices and components – for low pressure drops.
- Compare sound data for different types of fans and from different manufacturers and choose the quietist.
- Choose variable speed control for air flows rather than damper control.

SOUND CREATED BY FANS

Fans generate two main types of sound:

- Rotation sound
- Turbulence and vortex sound

Rotation sound

Rotation sound in a fan occurs when the rotating field of flow passes fixed parts in the fan casing, for example, the narrowest passages in a centrifugal fan, the bars in an axial fan or the vanes in the inlet to the fan.

The speed profile at the periphery of the impeller will have a minimum at the edges of the blades and a maximum between them. The blade passing through the narrowest section in the fan casing, past the so-called tongue in a centrifugal fan, will give rise to pressure changes – and thereby sound – the sizes of which depend on how much the pressure drop is affected, i.e. the distance between the fan blade and the tongue. The frequency of the sound will depend on the speed of the fan and the number of blades on the impeller. The natural frequency of the fan, also known as the blade frequency, can be expressed as:

$$f_s = n \cdot s \qquad (1)$$

where:
f_s is the natural frequency or blade frequency in Hz
s is the number of impeller blades
n is the fan speed in rps

The blade frequency is the most characteristic and noticeable frequency but it is also often possible to discern the next two overtones. Higher overtones are generally drowned by other sounds from the fan.

The sound power radiating from the fan is therefore dependent on the air speed profile at the narrowest section of the fan casing. This means that a fan with a large number of blades will create a lower sound power level than a fan with fewer blades, on condition that the distance between the impeller blades and the casing is the same in both cases. This is because a large number of blades will even out the variations in air speeds.

For the same reason, the blade frequency will not be noticeable in a fan with a large number of blades, for example, so-called squirrel cage impellers or sirocco impellers. The natural tone and overtones, dependent on fan speed and number of blades, will drown in the sound from other sound sources.

In fans with double inlets, the rotation sound will be reduced if the impellers are evenly offset to each other. For example, two six-blade impellers would be offset by 30 degrees to each other. In this way the amplitude of the rotation sound will be displaced half a wavelength out of phase, which means that the total amount of radiated sound will greatly decrease.

Turbulence and vortex sound

Turbulence and vortex sound occurs in fans for a number of reasons, including:

1. Turbulence in the air in the fan.
2. Turbulent boundary layers next to the blade surfaces.
3. Shedding of vortices at the edges of the blades.

These types of sound dominate in fans as soon as the frequencies, at which the blade frequency and its first overtones dominate, are exceeded. Shedding of vortices is especially noticeable in centrifugal fans that are heavily throttled. The air leaving the blades causes loud vortex noises on release.

CALCULATING THE SOUND POWER OF A FAN

When sound power level data for fans is not available or when a given value has to be checked, it is often possible to calculate the sound power level to a reasonable degree of accuracy. A number of formulae are available for providing rough estimates and common to them all is that the sound power is given as a function of the flow speed to the power of five, i.e. v^5. The same speed dependency is also applicable to other components in a ventilation system with fixed damping and turbulent flow.

As pointed out above, the operating point of a fan has a great effect on the generation of sound. A fan that is chosen for its high efficiency also generates, in general, the least sound for a given flow and pressure rise.

When the air flow and, consequently, the air speed in the system are increased or decreased, the change in sound power level can be written as:

$$\Delta L = 10 \cdot \log \left(\frac{v_2}{v_1}\right)^5 = 50 \cdot \log \left(\frac{v_2}{v_1}\right) \qquad (2)$$

Example:
When doubling the flow through a fan or ventilation system the sound power level in dB will increase by:

$$\Delta L = 10 \cdot \log \left(\frac{2}{1}\right)^5 = 50 \cdot \log\,(2) = 50 \cdot 0.3 = 15 \text{ dB}$$

If the flow is halved, the sound power level will decrease in a similar way, by 15 dB.

Using SI units, the formula for calculating the sound power level will be as follows:

$$L_{wtot} = 40 + 10 \cdot \log\, q + 20 \cdot \log\, p_r \qquad (3)$$

where:
L_{wtot} is the total sound power level for the fan in dB(relative to 1 pW)
q is the air flow through the fan in m^3/s
p_r is the pressure rise across the fan in Pa

'40' in Equation (3) represents the so-called specific sound power level. This value assumes a normal fan efficiency (53% for axial fans and 63% for centrifugal fans) and that the constants connected to the units used are taken into account. If q and p_r are replaced by older units (m^3/h and mm water column, i.e. kp/m^2) the specific sound power level will have a value of 25 for similar fan efficiencies.

In general, the equation has an accuracy of about ±4 dB, on condition that the fan has been correctly chosen and operates within the range of maximum efficiency. The values obtained by the equation denote the sound power levels at the inlet to and outlet from the fan when the fan has been installed, i.e. when both the inlet and outlet are connected to the ducting.

The extent to which the sound power level in the fan room will be

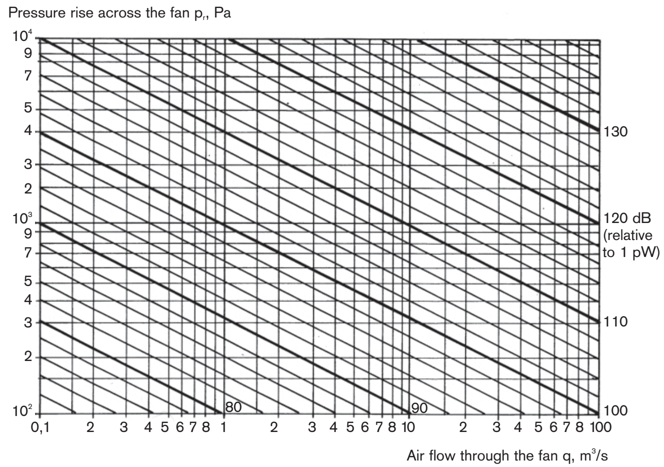

FIGURE 2. Graphical representation of Equation (3) when using SI units.

reduced depends on the type of fan, the thickness of the fan casing and the number of duct connections. The following values can normally be used as approximations:

- The total sound power level at the inlet or outlet after connecting the fan to the ducting: L_{wtot}
- The total sound power level in the fan room after connecting the fan to the ducting: L_{wtot} – approx. 10 dB
- The total sound power level in the fan room for an unconnected fan: L_{wtot} – approx. 6 dB

It is more difficult to predict the sound power level for fans installed in air conditioning units, as they will be affected by the acoustic surroundings in the unit in such a way that the level cannot be calculated but has to be measured.

OCTAVE BAND DISTRIBUTION OF THE SOUND POWER LEVEL OF A FAN

As the acoustic properties of sound sources and absorbents etc are strongly frequency-dependent, see Chapter 11/Building acoustics, sound calculations regarding the total sound power level for a fan are normally carried out for specific octave bands. If supplier's figures are not available, the following calculation method can be applied:

The total sound power level of a fan is spread over the different octave bands, with most of the sound within the octave band corresponding to the natural frequency. The natural frequency of the fan is calculated first, see above, and placed in the correct octave band.

Within the natural frequency octave band and the octave bands that have a higher frequency than the blade tone, i.e. higher up in the frequency scale, the sound power level will decrease by about 4 dB per octave band, and lower down, below the octave band of the fundamental tone, the octave levels will fall by about 3 dB per band.

Example:
Fan data:
Air flow $q = 4$ m^3/s; pressure rise $p_r = 1$ kPa; number of blades $s = 6$; speed $n = 2700$ rpm.

What is the sound power level in the 1000 Hz band?

Using the fan data in Equation (3)

$$L_{wtot} = 40 + 10 \cdot \log 4 + 20 \cdot \log 1000$$
$$= 40 + 10 \cdot 0.6 + 20 \cdot 3$$
$$= 106 \text{ dB(relative to 1 pW)}$$

Using the fan data in Equation (1)

$$f_s = \frac{2700}{60} \cdot 6 = 270 \text{ Hz}$$

This means that the blade tone belongs to the 250 Hz band, see Figure 3 in Chapter 11/Building acoustics.

Octave band, Hz	125	**250**	500	1000	2000	4000
Total sound power level L_{wtot}, dB	106					
ΔL/octave band, dB	–7	–4	–8	–12	–16	–20
Sound power level in octave band L_{woct}	99	102	98	**94**	90	86

Sound power level in the 1000 Hz band = 94 dB(relative to 1 pW).

SOUND CREATED IN VENTILATION DUCTING

Sound can be both created and dampened in the ducting system. Close to the fan, the sound from the fan will dominate but at a distance from the fan this sound will have been dampened in different ways and secondary sound sources in the ducting system will then dominate, with bends, junctions, dampers and terminal devices becoming sources of sound that will disturb room occupants more than remaining fan sound.

As the sound created by these components is strongly affected by the speed of the air in the ducting, it is important to keep speeds as low as possible and especially near occupied rooms. This can also have beneficial effects with respect to energy use. For the same reasons, ducting systems should be designed so that throttling dampers and other components that cause pressure losses, such as 90° bends and expansions and contractions to new cross-sectional areas, can be avoided.

The starting and finishing points for the air in a ducting system – the supply air and extract air terminal devices – must be chosen carefully.

The sound data supplied by the manufacturer must be checked to see whether it is applicable to the type of duct connection chosen. If there are a number of terminal devices in a room, the sound from these will be added logarithmically. As shown in Chapter 11/Building acoustics, the resultant sound level at a particular point in a room depends on the distance to the terminal devices, directivity factors and the equivalent sound absorption area of the room. Sound data for terminal devices and other components in rooms is normally given as the sound level in dB(A) in a room with an equivalent absorption area of 10 m^2 when they are placed in the reverberant field of the room.

To avoid disturbing flow-generated noise, one should:
- Design ducting and duct fittings for low air speeds.
- Avoid unnecessary turbulence by providing adequate distances between components (at least three duct diameters, but preferably more).
- Choose components that allow smooth flows through the ducting, bends, junctions and terminal devices.
- Avoid sudden cross-sectional changes or sudden changes of flow direction in the ducting system.

ATTENUATING SOUND IN THE DUCTING SYSTEM

Fan noise is reduced in a number of different ways when it is transmitted through the ducting system:
- As a result of sound power distribution.
- By attenuation in suction and pressure chambers.
- By leakage or so-called break out through duct walls.
- By attenuation in internally insulated ducts and bends.
- By using silencers in the ducting system.

Sound power distribution
The sound from a fan, like the air from a fan, is normally distributed between the branch ducts leading to the rooms served by the ventilation system. The proportion of the total sound power entering a given branch duct can be calculated from the ratio of the partial air flow to the total air flow, by using the following equation:

$$\Delta L = 10 \cdot \log \frac{q_{partial}}{q_{total}} \qquad (4)$$

where:
ΔL is the reduction of sound due to the sound power distribution in dB(relative to 1 pW)
$q_{partial}$ is the air flow in the branch duct in m^3/s
q_{total} is the total air flow from the fan in m^3/s

Example:
The air flow through the fan is 10 m^3/s and through the branch duct 100 l/s.

The attenuated sound power entering the branch duct is given by:

$$\Delta L = 10 \cdot \log \frac{q_{partial}}{q_{total}} = 10 \cdot \log \frac{0.1}{10} = 10 \cdot \log 10^{-2}$$
$$= 10 \cdot (-2) = -20 \text{ dB(relative 1 pW)}$$

If the flow conditions are expressed in percent, $[(q_{partial}/q_{total}) \cdot 100]$, the following values will be obtained:

Flow ratios, %	50	33	25	20	10	5	2	1	0.5
Attenuation, ΔL, dB	3	5	6	7	10	13	17	20	23

As in all sound calculations, sound level values have to be treated logarithmically.

Attenuation in suction and pressure chambers
A ducting system for supply air can be designed with a pressure chamber immediately downstream of the fan. Air is then distributed from this chamber via circular ducts to the different ventilated rooms. This is an excellent solution, as it prevents the transmission of disturbing fan noise through the ducting.

The sound, when it passes through the absorbent lined chamber, will be attenuated in proportion to the difficulty it has in finding its way out of the chamber. The smaller the outlet opening in relation to the total lined area, the more the sound will be forced to reverberate between the lined surfaces and the more it will be reduced. It is, of course, important that the inlet and outlet openings are not located opposite each other in the chamber, as there is a risk of the sound radiating straight across the chamber. If an outlet opening has to be located in this way, the chamber should be fitted with an internal baffle, lined on both sides with absorbent material and placed between the openings, so that the sound is forced around it.

The approximate attenuation in the chamber can be calculated as follows:

$$\Delta L = 10 \cdot \log \frac{S_0 \cdot \alpha}{S_1} \text{ dB} \qquad (5)$$

where:
S_0 is the lined surface area of the chamber including the openings in m^2
α is the sound absorption factor of the lining, see Chapter 11/Building acoustics
S_1 is the size of the outlet opening for the branch duct in question in m^2

The equation can also be solved graphically, see Figure 3. The diagram presented here is for a chamber lined with 10 cm mineral wool.

Unlined ducts
Unlined sheet steel ducts can attenuate low frequency sound, as the sound energy causes the thin walls of the duct to vibrate and thus function as a membrane absorbent, see Chapter 11/Building acoustics. The duct has a relatively high attenuating effect around the natural frequency of the sheet steel but is selective and only provides low attenuation above or below this frequency. This attenuation cannot be calculated as it is determined by too many variables, such as the thickness of the sheet steel and how it is stiffened, the ratio of the free area to the circumference of the duct, and how the ductwork is attached to the structure of the building.

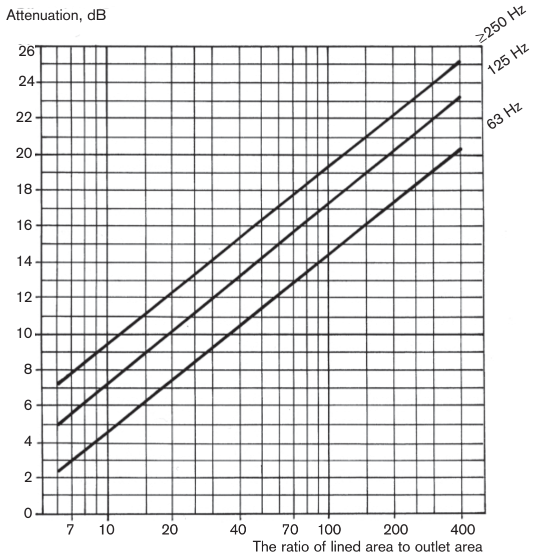

FIGURE 3. Equation (5) expressed graphically. Attenuation in a suction or pressure chamber internally lined with 10 cm of mineral wool.

When sound inside a duct causes it to vibrate, it will also vibrate externally and generate new airborne sound in a room: the inside of the duct acts as a microphone membrane and the outside of the duct as a loudspeaker membrane. Sound will leak, or break out, of the duct. This type of attenuation can, therefore, often have negative consequences, as the sound level in the surrounding space will be raised.

When rectangular ducting is installed in sound-sensitive rooms the ducts should be stiffened by cross creasing, scoring or grooving or by using external stays or stiffeners, and possibly by lining the ductwork with mineral wool or lagging it with gypsum board – see below in the section on crosstalk between ducts and rooms.

Unlined rectangular ducting

Straight unlined rectangular ducts that are not stiffened or strengthened – just like other membrane absorbents, see Chapter 11/Building acoustics – provide relatively good low frequency damping, see Table 1.

Unlined circular ducting

Circular spiral ducts, so-called spiro ducts, are considerably stiffer than rectangular ducts and therefore provide much less low-frequency attenuation and, consequently, less sound break-out, see Table 1.

TABLE 1. Attenuation in straight sections of 1 mm thick sheet steel ducts.

Duct size	**Attenuation in dB/m for different octave bands, Hz**				
	63	**125**	**250**	**500**	**≥1000**
Rectangular ducts					
75–200 mm	0.60	0.60	0.45	0.30	0.30
200–400 mm	0.60	0.60	0.45	0.30	0.20
400–800 mm	0.60	0.60	0.30	0.15	0.15
800–1000 mm	0.45	0.30	0.15	0.10	0.06
Circular ducts					
∅ 75–200 mm	0.10	0.10	0.15	0.15	0.30
∅ 200–400 mm	0.06	0.10	0.10	0.15	0.20
∅ 400–800 mm	0.03	0.06	0.06	0.10	0.15
∅ 800–1600 mm	0.03	0.03	0.03	0.06	0.06

Unlined bends

Depending on the dimensions of the ducts in relation to the particular wavelengths of the sound passing through them, a proportion of the sound will be reflected back into the duct at bends in the system. In round bends, reflection will be naturally quite small but will increase as the frequency of the sound increases, having a maximum value of about 3 dB, see Table 2.

TABLE 2. Approximate attenuation in round bends.

Duct diameter mm	**Attenuation in dB for different octave bands, Hz**					
	125	**250**	**500**	**1k**	**2k**	**≥4k**
125–250	0	0	0	1	2	3
280–500	0	0	1	2	3	3
530–1000	0	1	2	3	3	3
1050–2000	1	2	3	3	3	3

In rectangular lined bends, the damping will be considerably greater, especially at frequencies with the same approximate wavelengths as the width of the ducting. If a bend is lined with an absorbent, the damping will increase significantly.

Lined ducts and bends

Attenuation in lined ducts can be calculated using the following equation:

$$\Delta L = 1.05 \cdot \frac{P}{A} \cdot \alpha^{1.4} \qquad (6)$$

where:
ΔL is the attenuation in dB/m
P is the circumference of the lining in m
A is the cross-section area, the open area, in m^2
α is the absorption factor of the lining material in the relevant octave band

There is no point in lining a longer length of ducting than about five times its width. After this, the sound wave will have become flat and will be transmitted, primarily at its higher frequencies, relatively unattenuated in the middle of the duct (a certain amount of attenuation will, however, occur here by so-called diffraction).

When sound is attenuated using internal linings in the ducting the correct quality of material is essential – it must be able to withstand the air flow through the duct, it must not erode or emit particles, and it must be possible to clean the ducting using standard methods. This is also discussed in Chapter 19/Ducting systems. To avoid erosion of the lining material, the absorptive surface can be protected by using perforated metal sheeting, sometimes with the addition of a thin underlying layer of textile fabric. The metal sheeting will not reduce the absorption capability of the lining, compared to when the lining is completely unprotected, as long as the sheeting is chosen with an open area of at least 20%, i.e. with a maximum of 80% of the surface area of the lining covered by the sheeting.

Lining ducts with absorbents is an effective way of reducing noise, on condition that the material is placed so that the sound waves actually strike it. The absorption materials must therefore be attached to the ducts after changes in size and after bends, points where the air flow is turbulent. For example, lining the inside of the duct connecting to the fan outlet with absorption material is very effective. Here, the sound waves are very turbulent after leaving the fan impeller before they are realigned by numerous reflections against the duct walls. The placing of absorbents like this is called cross-wave attenuation.

When an absorbent lines a bend in a duct system it will attenuate directly incident sound as well as reflected sound both upstream and downstream of the bend. The extent to which the sound is attenuated in the bend will depend on the relationship between the duct size and the wavelength of the sound in the duct.

Table 3 shows the attenuation data for a bend in which the lengths of the lined duct sections before and after the internal bend are at least twice as long as the internal duct width and the lining material has a thickness t corresponding to at least 10% of the duct width, as shown in Figure 4.

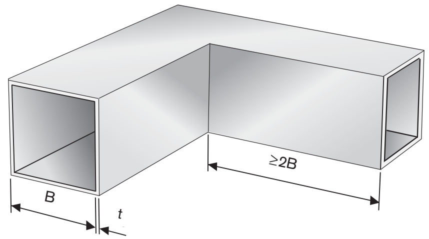

FIGURE 4. A lined bend.

TABLE 3. Attenuation data for rectangular bends with and without absorbent linings.

Internal duct width in mm	Attenuation in dB for difference octave bands, Hz						
	125	250	500	1k	2k	4k	8k
Bend without absorbent lining							
125				6	8	4	3
250			6	8	4	3	3
500		6	8	4	3	3	3
1000	6	8	4	3	3	3	3
Bend with absorbent lining before the bend							
125				6	8	6	8
250			6	8	6	8	11
500		6	8	6	8	11	11
1000	6	8	6	11	11	11	11
Bend with absorbent lining after the bend							
125				7	11	10	10
250			7	11	10	10	10
500		7	11	10	10	10	10
1000	7	11	10	10	10	10	10
Bend with absorbent lining before and after the bend							
125				7	12	14	16
250			7	12	14	16	18
500		7	12	14	16	18	18
1000	7	12	14	16	18	18	18

Flexible ducts
A flexible duct connection between a sheet steel duct and a supply air terminal device can be a convenient way of ensuring that the device can be positioned to fit the frame pattern of the suspended ceiling. However, care must be taken – if the flexible duct is pulled out of shape this could lead to a significant rise in the sound emitted in the terminal device, compared to that emitted when a straight connection is used.

SOUND TRANSMISSION BETWEEN DUCTS AND ROOMS

Sound leaking from ducting – break-out noise – can be regarded as a sound source in the room through which the duct passes. The sound power level created by a duct, as a source of sound in a room, can be calculated as shown below.

When a duct passes through a room, without either supplying or extracting air, the leakage of sound into or out of the room can cause problems, requiring the duct to be lined or other remedial measures to be taken.

The calculations are similar to those carried out for airborne sound insulation between two rooms, see Chapter 11/Building acoustics. The symbols and the influencing factors are, however, somewhat different:

$$L_{w(outside)} = L_{w(inside)} + 10 \cdot \log\left(\frac{S}{A}\right) - R_{duct} \qquad (7)$$

where:

$L_{w(outside)}$ is the sound power level emitted from the outside of the duct into the room in dB(relative to 1 pW)

$L_{w(inside)}$ is the sound power level of the sound in the duct in dB(relative to 1 pW)

S is the surface area of the duct emitting sound to the room in m^2

A is the cross-section area of the duct in m^2

R_{duct} is the sound reduction index of the duct wall in dB

The sound reduction index R_{duct} varies, depending on:

- Duct shape – whether rectangular or circular. A spiral circular duct is stiffer than a rectangular duct with the same free cross-sectional area and has, therefore, a higher sound reduction index.
- Duct size – the greater the width of a rectangular duct the more it will vibrate and emit noise.
- If the duct sheeting is stiffened, for example by cross creasing, it will have a higher sound reduction index than an unstiffened duct of the same dimensions.
- The frequency of the sound. The sound reduction index varies with frequency and rectangular ducts have a higher index at higher frequencies than at lower frequencies. In spiral ducts the opposite is true.

END REFLECTION

At the point where a duct is connected to a terminal device there will be a low-frequency damping effect due to reflection back into the duct. The damping will depend on the size of the opening and its location in relation to the walls and ceiling, i.e. its directivity factor Q, see Chapter 11/Building acoustics, and Figure 5.

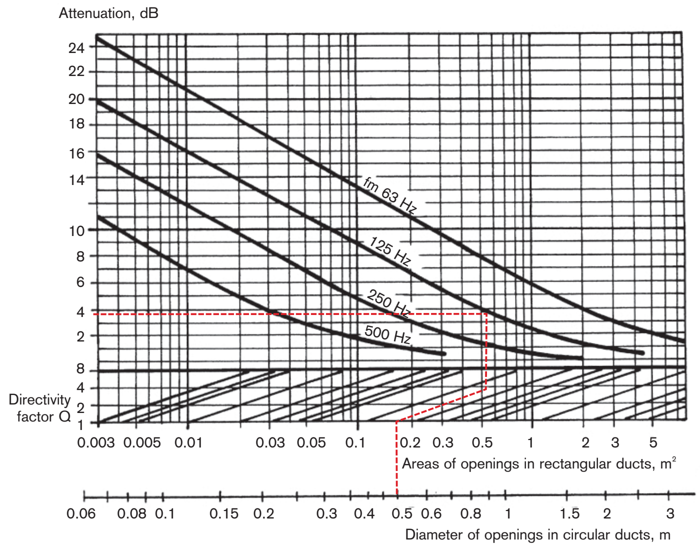

FIGURE 5. Attenuation in terminal devices and duct openings. The dashed line illustrates the following case: Directivity factor, Q = 4 and the duct diameter = 0.5 m. Attenuation at a frequency of 125 Hz will be 4 dB.

SILENCERS

Silencers, or attenuators, should normally be placed as close to the sound source as possible but at a sufficiently large distance to ensure that the air flow in the duct has a reasonably stable speed when it reaches the silencer. As in other natural phenomena, sound also flows from a higher, louder, energy level to a lower, quieter, energy level. This is why a silencer, fitted to reduce the fan sound transmitted by the ducting, should be positioned next to the fan room wall connection. This prevents the sound emitted in the fan room, which would be at a higher level than the sound in a duct section after the silencer, from being transmitted into the duct through the duct walls. If this silencer location is not possible for space reasons, this section of the duct should be lined with mineral wool or lagged using gypsum board.

Equation (6) above shows how attenuation in a lined duct depends on the sound absorption factor α and the relationship between the circumference of the lined duct area and the open area, P/A. If the P/A ratio is increased for a given open area, then the attenuation will become more effective. Advantage of this fact is taken in prefabricated silencers in which the surface area of the lining is increased by placing walls lined on both sides, baffles, parallel to the air flow. The attenuation depends on the distance between the surfaces and the shorter the distance the more effective the silencer will be. This solution will, however, cause an increase in the pressure drop across the silencer.

To improve the attenuation properties at low frequencies, where porous absorbents have low attenuation, see Table 6 in Chapter 11/Building acoustics, porous material is sometimes combined with the excellent low frequency properties offered by membrane absorbents. This can be achieved by lining part of the porous absorbent surface with sheet steel.

The silencer not only attenuates sound but also generates sound, just like any other duct-mounted components, and, like these, its self noise increases as the air speed increases. When choosing a silencer it is important to check the manufacturers data for both of these properties, i.e. attenuation and self noise.

To facilitate operations when cleaning ducts, it should be possible to remove the silencers, especially those with baffles and those installed inside the supply air ducting. Cleaning ducts is discussed in Chapter 19/Ducting systems.

Crosstalk attenuators

It is often important to prevent speech from being transmitted between workplaces and other rooms in a building, not only for reasons of secrecy but also to reduce disturbing noise. How this affects the choice of partition walls, and their sound reduction indices and airtightness, is discussed in Chapter 11/Building acoustics.

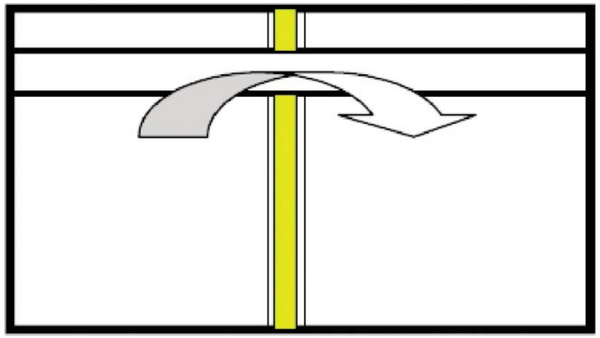

Figure 6. Crosstalk between two rooms via ducting.

When rooms are connected to a common supply air duct and/or extract air duct it is important that speech cannot be conveyed between the rooms, so-called crosstalk. In the daytime, when the ventilation system is in operation, this is normally not a problem – sounds from the system create a background level that is sufficiently high to drown speech. When the system is turned off at the end of the working day the conditions change and the ventilation system no longer has an attenuating effect. Supply and extract air terminal devices should therefore also provide sufficient attenuation, to prevent crosstalk, when air flows in the ducting cease. As an alternative, duct connections to the terminal devices can be fitted with attenuators to reduce crosstalk.

TERMINAL DEVICES

The location of the supply and extract air terminal devices will affect the sound distribution in a room as the walls and ceiling will reflect sound from the terminal devices. This means that terminal devices have directional characteristics, directivity factors, see Chapter 19/Ducting systems. If a device is placed in the corner of a room it will be surrounded by three reflective surfaces. A higher sound level will then result, at a given distance from the device, compared to that from a device placed at the junction of a wall and the ceiling (two reflective surfaces) or in the middle of the ceiling (one reflective surface). For each additional surface the sound at a given distance from the device will increase by 3 dB.

Sound data for a terminal device is normally expressed as the sound level in dB(A) in a room with a 10 m² equivalent absorption area measured in the reverberant field of the room. It is therefore important to check the following conditions, which can affect performance, in the installation in question:

- The setting of the terminal device to provide the design air flow and correct distribution pattern.
- The connection of the terminal device to the duct – is this via a straight section or a bend?
- The directivity factors.
- The effects of turbulent flow through the device (disturbances, for example, caused by balancing damper in the duct).

Adjust the sound level value if actual conditions do not agree with the product data, for example, for distances and the equivalent absorption area of the room, and add the increase due to multiple parallel devices in the room.

ACTIVE NOISE ATTENUATION

Although active silencers are unusual today, they might become more common in the future. They are especially suited to attenuating low frequency noise, for which standard attenuators are ineffective. An active attenuator produces sound waves that are out of phase with those in the ducting.

An active silencer functions as follows: A reference microphone is placed upstream on the wall of the duct and measures the sound. After it has been analysed, a mirror image of the measured noise with the same amplitude is fed into the duct via a loudspeaker placed on the duct downstream of the reference microphone. This anti-phase (180 degrees out of phase) sound will effectively interfere with the noise in the duct and attenuate it. The system is also fitted with an error microphone that measures the resulting sound levels after the attenuator and adjusts the analyser to refine the signal to the loudspeaker. As all parts are mounted remote from the air flow the silencer does not create a pressure drop or any self noise.

The air speed and the turbulence of the air flow must not be too high, if the attenuator is to work properly. The microphones used to measure the sound cannot differentiate between sound, i.e. pressure propagation in an elastic medium, and pressure changes caused by air movements in the duct. This is why TV reporters, when interviewing people outdoors, use special windshields to protect their microphones.

Active silencers should only be used in ducts where the air speed does not exceed about 8 m/s and at a sufficient distance from components creating turbulence, i.e. more than about 5 duct diameters upstream and about 3 diameters downstream from them.

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27. AIRBORNE INDOOR CLIMATE SYSTEMS

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WHAT IS AN AIRBORNE INDOOR CLIMATE SYSTEM?

What is the first thing you think about when you hear the expression 'climate system'? Most people usually think of the sun, wind, sea, air and the atmosphere, now being destroyed by mankind. This is not at all strange, as we are constantly reminded in the media about the dangers to which our global outdoor environment is being subjected. Far fewer people think about how *indoor* climates affect us. It's perhaps sufficient to point out that living in modern society means that we spend more and more time indoors, in fact, as much as 90% of our time. We can note that our daily intake of food is 0.75 kg and we drink about 1.5 l of liquids. This can be compared to the 15 kg or so of air that pass through our lungs every day. These intakes affect our health and the more pollutants and poisons we consume, the more our purifying systems will have to do. If we consider the possible contents of the air we breathe and what happens in our body cells when blood is oxygenated, it is clear that it would be a good idea to make sure that the air we breathe indoors maintains a sufficiently high quality. Ensuring this quality is one of the most important tasks of an airborne indoor climate system and this and other functions of such systems are discussed in this chapter.

An airborne indoor climate system is a system in which conditioned ventilation air in a building is the carrier of the cooling and heating energy required to provide a correct indoor climate and to satisfy the occupants' needs for a good indoor climate. It must ensure that all the different rooms in a building are provided with an adequate climate with respect to air quality and the thermal environment. And, of course, it must also provide an acceptable acoustic environment.

Practical applications

Airborne indoor climate systems are often used in industrial buildings and shops, especially outside Scandinavia, but they are much less common in other types of commercial buildings and residential buildings. Here, heating is often achieved by using radiators under windows or underfloor heating systems. However, the opportunities for using air as an energy carrier are increasing, as buildings are being developed to meet requirements for less heating. An argument against heating using ventilation air is that it is not sufficiently efficient. A critical factor is the ability of a system to meet temperature demands along the insides of external walls. Today, there are many good examples of office buildings where this has been achieved. An office building often has a sufficiently large heat surplus during the whole of the year, while the building is in use, which means that heating is only required while the building is empty. An important condition, however, is that critical building components, such as external walls and windows, have adequately large thermal insulation properties. Another important condition is that the HVAC system has effective and efficient supply air terminal devices, ATDs, that can discharge air at an over temperature in an efficient manner. Finally, a good control system is required. Both full-scale tests and field tests have shown that airborne indoor climate systems work extremely well, if all the requirements are properly analysed and the plant installed can work in conjunction with the structure of the building and the materials used.

In the following, we will concentrate on how airborne indoor climate systems work in individual rooms.

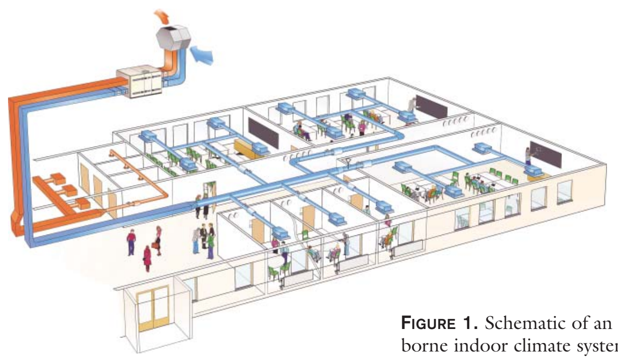


FIGURE 1. Schematic of an airborne indoor climate system.

THE CLIMATE IN A ROOM

A building is a structural envelope in which people can live and work. If rooms are to be used for their intended purposes, a number of technical climate factors must be kept within suitable limits. The first three factors in the following list are directly dependent on the air handling system:

– Air quality
– Thermal climate
– Acoustic environment
– Visual environment

These factors and the effects they have on people's health, comfort and work performance are described in Chapter 1/Health and well-being in indoor environments, and the chapters in Part D/The indoor environment – in a wider sense.

THE DISTRIBUTION OF AIR IN A ROOM

In the parts of a room that are occupied by people, it is important that the speed of the air is not too high and its temperature not too cold, so that no uncomfortable draughts are experienced. To be able to provide the correct temperature and ventilate a room at all, zones are required into which air can be supplied. This is why the concept of a so-called occupied zone has to be defined. There are a number of definitions and the following is taken from the Swedish Building Regulations:

An occupied zone in a room is limited by two horizontal planes, one 0.1 m above the floor and another 2.0 m above the floor, and by a vertical plane 0.6 m from the external wall or other external limit, or 1.0 m from a window or door.

In practice, the size of the occupied zone is greatly dependent on the choice of supply air system and, above all, by the type of ATDs used.

Air speeds within the occupied zone must not, as a rule, be allowed to exceed 0.15 to 0.25 m/s. The speed of the air, at which it can be regarded as accepable, will depend on its temperature. The lower the temperature, the more its speed has to be limited. Speed limits can, therefore, be different in summer and winter.

ATD manufacturers supply data for so-called throws, sizes of near zones, functional distances etc, so that air speeds in the occupied zone can be taken into account during the planning phase of a project. These details describe the spread of the air jet from the ATD into a room or space. The spread is often defined as the boundary distance at which the air speed has fallen to 0.2 m/s. Figure 2 shows the throw for a mixed flow ceiling mounted device, where the air speed beyond the boundaries is less than 0.2 m/s.

Figure 3 shows an example of the spread in the near zone across a floor from an ATD used in a thermally controlled displacement ventilation system.

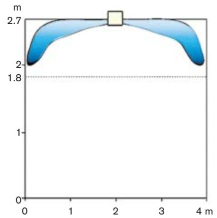


FIGURE 2. The throw of a ceiling mounted ATD.

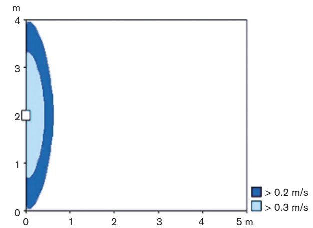


FIGURE 3. Horizontal near zone spread pattern.

Air jets

Three main types of air jets are normally used in comfort ventilation systems. In free radial and axial jets, the supply air jet will follow the surface of the ceiling or a wall. In jets like these, the supply air is discharged at high speed and the room air is inducted into the air jet. At the other end of the scale are jets with low speeds and low momentum, into which the room air is not inducted.

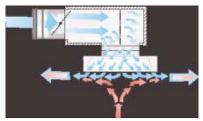


FIGURE 4. Radial free air jet.

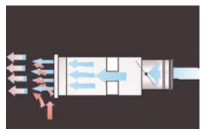


FIGURE 5. Axial free air jet.

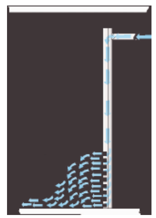


FIGURE 6. Low momentum air jet.

As cold air is denser than warm air, it is important when designing supply air systems to choose terminal devices with correct throws, to prevent draughts from being created in the occupied zone. Here, it is a question of being able to control the direction and spread of the air flow in a room. If there are a number of ATDs in the same room, this feature will be even more important. And, just as in acoustics, there are additive effects of air jets placed close to each other. Jets colliding with each other or with walls, pillars, fittings and ceiling beams etc must, of course, also be avoided. Collisions like these normally lead to the air jets being deflected into the occupied zone, resulting in annoying draughts.

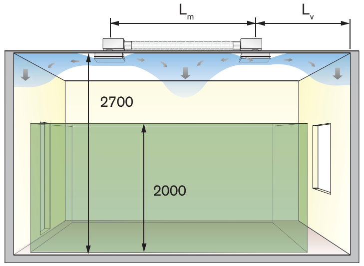


FIGURE 7. Collision risks between air jets. The green volume represents the occupied zone.

The safest way to introduce an air jet at high speed is to direct it parallel to the ceiling, or the walls, in the room. This will create an under pressure above, or behind, the air stream, helping it to cling to the adjacent surface. This effect is often called the ceiling effect or Coanda effect, after the Rumanian scientist Henri Coanda (1886–1972).

VENTILATION PRINCIPLES

There are three different ways in which an airborne indoor climate system can ventilate a room. The two most common are based on the properties of mixed flows and thermally controlled flows. In very special instances the air flows can be created by so-called piston flow, which is used in clean room applications. To get a picture of how good these different flow techniques work, a number of efficiency concepts can be used, such as ventilation efficiency, air change efficiency and temperature efficiency. These different efficiencies and how they can be determined are discussed in Chapter 16/Air change and air flow.

To achieve suitable air flows with high degrees of efficiency, the design engineer must pay great attention to the following parameters:

– Where the terminal device is mounted
– The type of supply or extract air terminal device
– The speed of the supply air
– The temperature difference between the supply air and the extract air
– Obstacles in the room, heat sources, type of activities, sizes of windows etc.

Figures 8 and 9 illustrate the air flow principles for mixed flow ventilation and thermally controlled ventilation. Thermally controlled ventilation can be divided into two types: displacement flow ventilation and so-called equalizing ventilation.

Mixed flow ventilation is characterized by the room air being inducted into a high-speed air jet. When displacement ventilation is used supply air at low speeds is discharged into a room close to floor level. The supply air must be cool so that it can spread across the surface of the floor. When the air is subsequently heated by heat sources in the room it will rise towards the ceiling, carrying with it pollutants from the occupied zone. In equalizing ventilation systems the room air is mixed with the supply air. This means that the supply air gains a temperature level close to that of the room air. The different air flow principles are discussed later on in this chapter.

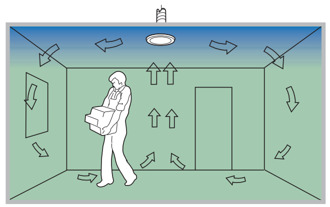


FIGURE 8. Mixed flow ventilation.

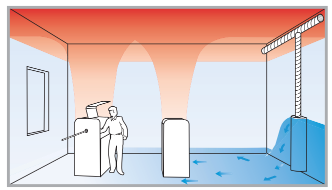


FIGURE 9. Thermally controlled displacement ventilation.

COOLING WITH AIR

When can the air in a ventilation system be used as a cooling medium?
To achieve a cooling effect using the ventilation air, the supply air temperature must be lower than the room air temperature. As mentioned above, there are limits regarding how cold the air can be without it causing discomfort in the form of draughts.

The risk of draughts increases the higher the air flow. A typical temperature of the supply air in a mixed flow ventilation system is about 8 °C below the room temperature. In displacement ventilation systems, the supply air temperature might have to be limited to being 3 or 4 °C below.

An alternative to airborne cooling is to remove the heat surplus in a room using cold water in ceiling mounted cooling coil units, so-called chilled beams. Systems like these are discussed in Chapter 28/Waterborne indoor climate systems. Both methods have advantages and limitations.

Advantages of using air as a cooling medium:
- In large parts of the northern hemisphere, it is possible to make use of so-called free cooling, as the outdoor air is sufficiently cold for most of the year
- No extra water systems are required, i.e. no extra piping, pumps or control systems.
- The same system is used to meet the requirements for good air quality and good thermal comfort.

Disadvantages of using air as a cooling medium:
- Larger air flows than otherwise are needed, if the ventilation is only designed with respect to the quality of the air. This, in turn, requires larger diameter ducting.
- Great care is required when choosing the correct size and type of air terminal device, so that draughts can be avoided.

Different ventilation principles have different surplus heat removal properties.

Table 1 shows a number of recommended values for maximum cooling powers when using different types of ventilation. The table is for a room with a ceiling height of 2.8 m and an occupied zone as in Figure 10.

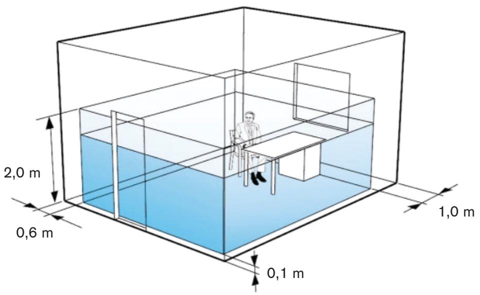


FIGURE 10. The occupied zone.

TABLE 1. Recommended values for cooling powers for different ventilation principles, expressed in W/m² floor area.

Ventilation principle	Max. cooling power W/m²
Mixed flow ventilation	
Ceiling mounted with nozzles	90 to 120
Other ceiling mounted types	60 to 80
Rear wall mounted	50 to 60
Front wall mounted	60 to 70
Window sill	50 to 70
Displacement ventilation	
Floor mounted	30 to 35
Wall mounted, base 0.6 m above floor level	35 to 40
Equalizing ventilation	
With induction unit, 1.2 m above floor level	40 to 45

SYSTEMS AND COMPONENTS

A distinction is made between CAV, constant air volume, systems and DCV, demand controlled ventilation, systems. The principles governing these systems are described in detail in Chapter 17/Demand-controlled ventilation.

To ensure a comfortable indoor climate and good air quality without wasting energy, the ventilation flows must be utilized as efficiently as possible. A good way of doing this is to install a DCV system. This type of system ensures that proper ventilation is achieved where it is required and that air flows are minimized, if temporarily not required. This, combined with the fact that it is now possible to make these systems self-regulating and self-diagnosing, means that it is now possible to design very energy-efficient indoor climate systems.

To create a well-functioning ventilation system, user-friendly, purpose built and reliable components will be required. The components in the central air handling unit, for example, heat exchangers, fans and filters, can, in principle, be the same, irrespective of whether they are used in a CAV or a DCV system. Components like these are discussed in other parts of this book. When it comes to components for flow control these can often be simpler in CAV systems than in DCV systems. These components are discussed briefly below.

Components for flow control

To attain the correct size of air flow, it must be possible to measure, balance and control it. This is done using different types of components:
- Sensors for measuring pressure, flow, temperature and air quality.

- Regulators and room thermostats etc.
- Dampers for balancing and flow control.
- Air terminal devices with built-in measuring and control functions.

Components used in ducting systems must be fitted with dampers so that excess pressures can be reduced while keeping the noise level to a minimum. It must be possible to lock the damper in the required position and clearly see the setting. The functions for measuring pressure and air flow must be reliable even at low flow rates. To achieve acceptable measuring accuracy, straight and uninterrupted sections of ducting are required before the measuring point. It is, of course, best, if these straight sections are not required to be too long. The components should also be easy to install.

The room components should be user-friendly with respect to those authorized to use them and it should not be possible for unauthorized persons to change their settings. It must also be easy to reset desired values. Even these components should be easy to install.

Examples of components for flow control in CAV and DCV systems are shown in Figures 11 and 12.

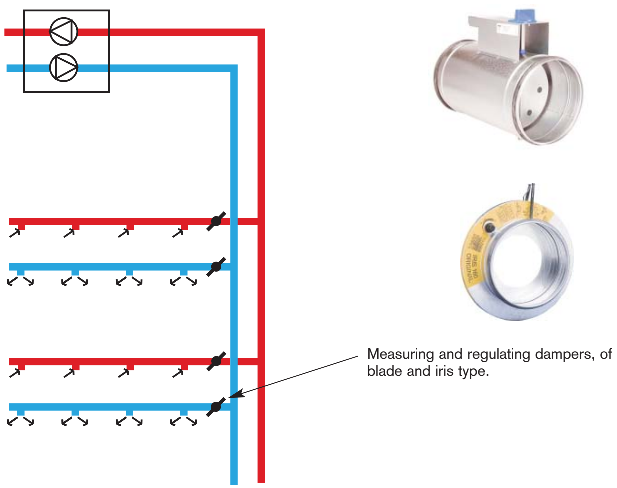


FIGURE 11. A CAV system.

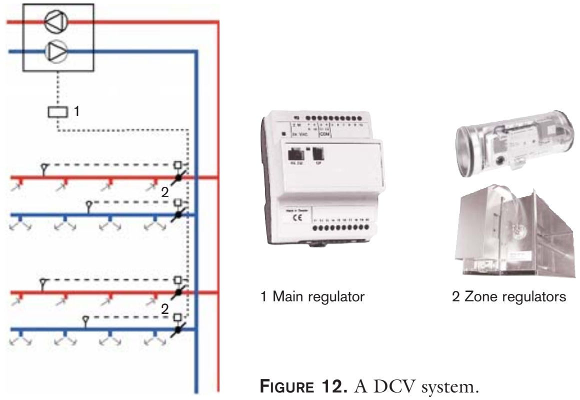


FIGURE 12. A DCV system.

Examples of system solutions using room products in a DCV system are shown in Figures 13 and 14 below.

FIGURE 13. An active ATD with an external room thermostat and sequentially controlled radiator valves.

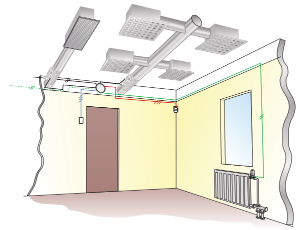


FIGURE 14. Passive nozzle diffusers with centrally controlled flows governed by duct dampers, air quality sensors and sequentially controlled radiator valves.

In addition to the solutions shown in Figures 13 and 14, there are a large number of other solutions that can be used when there is a central communications system.

No matter what the conditions, it is very important that the systems are simple to install, to use and to maintain.

LOCATING TERMINAL DEVICES

Locating supply ATDs in mixed flow ventilation systems

Room air can be well-mixed in a number of ways. Most important to remember, however, is that it must be done in a controlled way. This is why this section concentrates on supply and extract mechanical ventilation systems. Natural ventilation and extract air ventilation systems, in which the outdoor air is admitted via trickle ventilators/slot air valves above or below windows, is not discussed.

Centrally placed, ceiling mounted, built-in supply ATDs
The ceiling is the best place to mount a mixing or swirl diffuser, as the air can be discharged over 360° with an excellent Coanda effect. This means that air at an under temperature can be used without any great risk of causing draughts. The jets must be prevented from colliding with obstacles such as surface-mounted light fittings, ceiling beams etc. Otherwise there is a risk that the flows will be directed into the occupied zone, with draughts as a result. If there are a number of diffusers on the ceiling in the same room, care must be taken so that the jets cannot collide with each other. This is avoided in most modern ceiling diffusers by using deflecting vanes. Convective flows from radiators, office equipment etc must also be taken into account.

Another great advantage of ceiling mounted diffusers is that there is sufficient space for the connecting ducts. And the straighter the ducting leading into the diffuser, the quieter and more precisely adjusted the air flow will be.

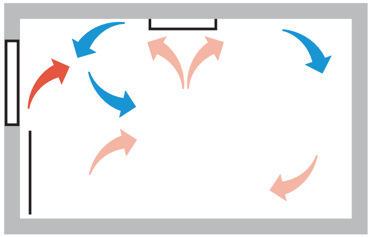


FIGURE 15. A centrally placed, ceiling mounted, built-in diffuser.

Front wall, ceiling mounted, built-in supply ATDs.
In this case the device is mounted close to an external wall, i.e. far from the corridor wall. Even this is one of the best locations. Air cannot be discharged over 360°, only over 180°.

As the diffuser is far from the back wall, the throw has to be somewhat longer than for a centrally placed, ceiling mounted device.

When air at an under temperature is used a design throw that is slightly greater than the depth of the room is recommended, so that the jet will not deflect down into the occupied zone too soon.

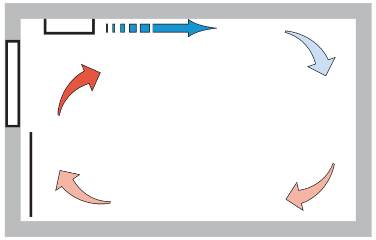


FIGURE 16. A front wall, ceiling mounted, built-in diffuser.

Back wall mounted, built-in supply ATDs
Sometimes it is not possible to mount the diffuser on the ceiling. An economical way of solving this problem is to mount it on the back, or corridor, wall immediately next to the ventilation ducts in the corridor.

As wall diffusers do not have the same performance characteristics as ceiling devices, it is essential to choose the correct type of device for this application. The throw here is very important. When air at an under temperature is discharged into a room it is recommended that the throw be at least 75% of the depth of the room. The device should preferably be placed 100 to 200 mm below the ceiling and its air jet directed at about 45° towards the ceiling, to attain the best possible Coanda effect. In this application, there is hardly any space at all for connecting the ducting to the device in a satisfactory way and this must be taken into consideration

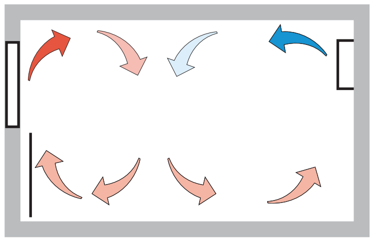


FIGURE 17. A back wall mounted, built-in diffuser.

at the planning stage, so that no unnecessary noise or poorly balanced terminal devices are created.

Built-in supply ATDs for windowsills and floors
Office buildings are sometimes fitted with so-called perimeter systems, which are, briefly, systems in which cooled or heated air is discharged vertically towards the ceiling via a windowsill. In most solutions, the air is distributed via an induction unit placed under the windowsill. In this application, it is important to keep an eye on the temperatures of the supply air and the surface of the window. If both these temperatures are lower than the room temperature, there is a risk that cold down-draughts will be created in the occupied zone closest to the window, i.e. next to the workplace. This means that perimeter systems must have sufficient pressure to ensure that the air jet has a suitably long throw, so that the risk of draughts can be eliminated.

Locating supply ATDs in thermally controlled ventilation systems
As the air in these systems is often discharged directly into the occupied zone, it is important that the supply air diffuser is positioned correctly. The position is, in many cases, dependent on the type and intensity of the activities taking place in the room as well as the type of room. People who work sitting down should not be placed too close to the diffusers.

In premises where pollutants are generated to a relatively high degree, it is even more important that the displacement diffusers discharge air at low speeds and with a minimum of induction of the room air. It is also important that premises like these have high ceilings. The higher the ceiling, the more space for the polluted air to collect above the occupied zone.

Floor mounted ATDs for thermally controlled displacement ventilation systems
The supply air in these systems is discharged at low speed from the whole surface area of the diffuser.

As the air is at a slight under temperature, it will fall onto the floor relatively quickly. How quickly this occurs depends on the flow rate and the temperature of the air. In normal cases, at a distance of 0.5 to 1 m from the diffuser, the air jet will have spread out over the surface of the floor to a depth of 50 to 70 mm above the floor. If the diffuser is fitted with a function for adjusting the spread pattern after installation, this is, of course, an advantage, as the diffuser is actually in the occupied zone.

The under temperature of the supply air is normally between 3 and 6 °C, depending on the type and intensity of the activities in the room.

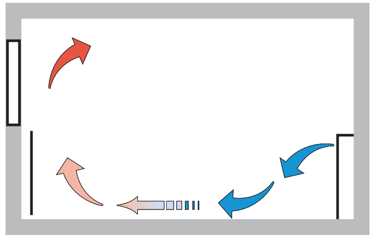


FIGURE 18. Floor mounted supply air diffuser in a thermally controlled displacement ventilation system.

Floor and wall mounted ATDs for thermally controlled equalizing ventilation systems
In these systems, a certain amount of return air is used. This is achieved either by integrating an induction unit into the diffuser or by placing the diffuser high up in the room, or by combining both these solutions.

Sometimes both mixing and displacement supply ATDs are used in the same system, for example in conference rooms. If the cooling power has been designed based on the mixing function, then there must be some sort of compensatory function for the displacement diffusers, as these cannot manage as low supply air temperatures as the mixing diffusers. This is achieved by fitting the displacement supply ATDs with reheaters or an induction unit, which is most probably the most cost-effective solution.

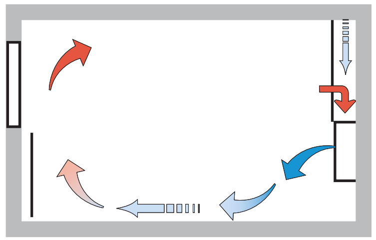


FIGURE 19. A wall mounted supply ATD in a thermally controlled equalizing ventilation system.

If an induction unit is used, the primary air can be allowed to have an under temperature of 6 to 9 °C.

When the primary supply air flows through the unit, the room air is inducted into the diffuser. In this way, the temperature of the air that is blown into the device is evened out. The air is then discharged into the room at a low speed from over the whole surface of the diffuser. The diffuser then functions as a normal, floor mounted, displacement supply ATD.

SUPPLY ATDS FOR MIXED VENTILATION SYSTEMS

FIGURE 20. A ceiling mounted supply ATD in an office environment.

A wide range of supply and extract ATDs are available on the market today. Supply ATDs should be able to:
- Discharge large volumes of air without causing draughts.
- Measure and regulate air flows.
- Operate at low noise levels.
- Change their spread patterns.
- Offer aesthetically pleasing designs for greater acceptance by architects.
- Offer simple and cost-efficient installation, commissioning and maintenance.

In order to fulfil these requirements, the ATD must have:
- A high induction capacity for room air to reduce draught risks at low supply air temperatures.
- Adjustable deflector vanes or nozzles that can be set even while in operation, to offer flexible spread patterns.
- Air flow measuring points for adjustment and control.
- An integrated air flow adjustment damper, to reduce the risk of having a poorly balanced system.
- A connection box with effective noise attenuation, to reduce the risk of crosstalk via the ducting system.
- A design that complies to building standards to reduce building costs.

To be able to fulfil all these functional requirements, the ATDs often comprise a diffuser and a ducting connection unit, the latter in the form of a plenum box with built-in functions for air flow regulation, noise attenuation, air distribution and air flow measuring. If only the diffuser element is used, it will not be possible to control the discharged air.

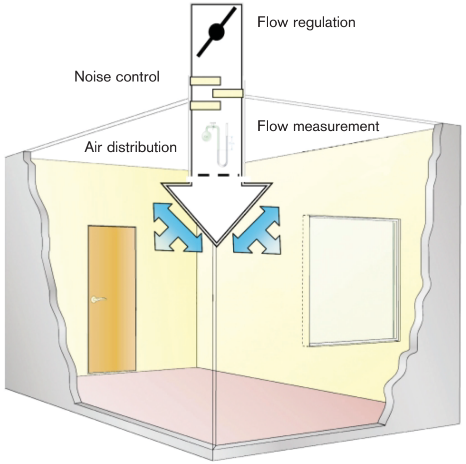


FIGURE 21. Supply ATD with adjustable functions.

Passive supply ATDs

The simplest types of supply air diffusers have constant discharge areas. This means that if the air flow is reduced, it will be difficult for the air jet to support the air above the occupied zone over a sufficiently long distance. There are, however, passive devices that can accommodate quite large variations in air flow.

Active supply ATDs
In recent years, two types of variable or active supply air terminal devices have come onto the market. Their different functions are described below.

Active ATDs with automatic direction change of the air jet. These are supply air diffusers for premises with high ceilings. They are used to supply air either at an under or over temperature, using the same device. The diffuser is fitted with an electric or thermal motor that drives a damper arrangement to change the direction of the air jet, depending on the temperature of the supply air. The main areas of use include arenas, large shops, industrial buildings etc.

Active ATDs with automatic continuous regulation of the discharge area. These are supply air diffusers for premises with normal ceiling heights and are used exclusively for demand-controlled ventilation systems. The continuously variable regulation of the discharge area has two functions. Firstly, the discharge area is always optimized to suit the flow at any given time, which eliminates the risk of draughts. Secondly, the variable discharge area works as an air volume regulator. Active devices are most often used in office buildings and schools.

There are many different types of ATDs and this is due to a number of reasons, though all of them are related to room parameters, such as:
- Use – Residential, office, industrial etc.
- Ceiling height – High ceilings require special terminal devices.
- Thermal climate – Air at under or over temperatures requires special terminal devices.
- Air quality – Demand-controlled air flows require terminal devices that can manage variable air flows.
- Design – Specially designed terminal devices might be required, choice of colours etc.
- Type of ventilation system – Perimeter and floor mounted systems require special terminal devices.

ATDs are normally categorized depending on where they are placed in a room:
1. Ceiling diffusers
2. Wall diffusers
3. Window sill diffusers
4. Floor diffusers

1a. Passive ceiling diffusers

Nozzle diffusers
Nozzle diffusers referred to here are ATDs with a number of individually adjustable, aerodynamically designed plastic nozzles that interact to provide an optimal and flexible air jet.

FIGURE 22. Nozzle diffusers.

Important features:
- The design of the nozzle creates an evenly distributed air flow across the whole surface area of the device, ensuring a low noise level and even spread pattern in the room.
- A high degree of induction. The nozzles help to quickly and effectively mix the room air with the supply air jet, allowing the temperature difference between the room air and supply air to be greater than for other types of devices, without causing draughts in the room. This makes the device very suitable for handling cooled air.
- The flexible spread pattern can be easily changed, without the set flow, pressure drop or noise level being changed in the room.

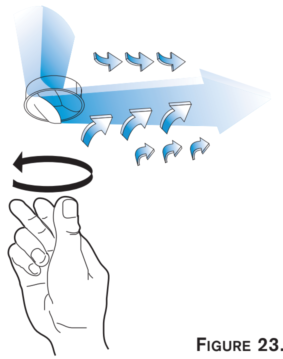


FIGURE 23. Flexible spread patterns.

- Can manage supply air at large under temperatures.
- Can manage variable flows, as low as 20% of normal flows rates, thanks to excellent induction properties.
- Both short and long throws can be achieved, both horizontally and vertically. Combined horizontal and vertical spread patterns are also possible. This type of diffuser can be used in rooms with relatively high ceilings.

Disadvantages:
- The air flow capacities are not as great as those in perforated ATDs.

Swirl diffusers
These devices are characterized by discharge openings shaped so that the air stream swirls like a horizontal vortex. This gives the diffuser an extremely good induction capacity.

FIGURE 24. A swirl diffuser.

Important features:
- The design of the discharge openings creates of an effective circular and tangential spread pattern.
- High induction capacity. The spread pattern facilitates effective mixing of the room air into the supply air jet. This makes it very suitable for handling cooled air.
- Can manage large under temperatures.
- The high induction capacity means that the device can easily manage variable flows, down to 30% of normal air flows.

- Some versions of swirl diffusers have adjustable discharge gaps, making them also suitable for use in rooms with high ceilings.

Disadvantages:
- Limited flexibility with regard to spread pattern.
- Do not have as large air flow capacities as perforated devices (with the exception of some of the devices adapted to industrial ventilation applications).

Perforated diffusers
The advantage of a perforated diffuser is that it can distribute large air flows over relatively small areas without causing draughts.

Despite its large air flow capacity, this type of diffuser has a comparatively small throw. Perforated diffusers are mostly used in rooms with ceiling heights lower than 3 m.

FIGURE 25. A perforated diffuser.

Important features:
- Can supply large air flows to rooms with ceiling heights lower than 3 m.
- Short throws.
- Can manage supply air at quite large under temperatures.
- Can provide flexible spread patterns.

Disadvantages
- The discharge opening is not flush with the ceiling.
- Limited flexibility with regard to spread pattern when compared to the nozzle diffuser.

Linear slot diffusers
Above all, linear slot diffusers provide an excellent solution where the appearance of the diffuser and the ability to match the design of the room are the most important properties. This type of diffuser is often used in countries where architects have high status.

FIGURE 26. A linear slot diffuser, also shown installed in a suspended ceiling.

Important features:
- Ability to match the interior design.
- High degree of induction of room air.
- Ability, to a certain extent, to manage supply air at under temperatures.
- Can be joined together to form long continuous slots.
- Ability, to a certain extent, to provide flexible spread patterns.

Disadvantages:
- Limited air flow capacity.
- Difficult to supply an even flow along the whole of the diffuser.
- Long throws despite high degree of induction.

Circular and square slot diffusers

FIGURE 27. Circular and square slot diffusers.

This type of diffuser can be divided into two groups: single-slot diffusers and multi-slot diffusers. The narrower the slots, the better the induction of the room air. Generally speaking, this type of diffuser has the highest capacity with respect to air flow rates.

Important features:
- Can manage large air flows.
- High degree of induction when narrow slots are used.
- Relatively large under temperatures can be allowed.

Disadvantages:
- Long throws.
- Limited opportunities for flexible spread patterns.

Jet diffusers
Jet diffusers are used in premises with very high ceilings. They are often used in industrial buildings, airports and shopping centres etc.

FIGURE 28. A jet diffuser, also shown installed in visible ducting.

Important features:
- Can manage large air flows.
- High degree of induction.
- Supply air at relatively large under temperatures can be used.
- Extremely long throws (required in large premises).

Disadvantages:
- Most often have to be motorized to attain directional flexibility (often mounted high up).

Duct diffusers
When supply ATDs are required in rooms without suspended ceilings, duct diffusers often provide a suitable solution. These combine ventilation ducting and diffuser features so that they can be assembled in long

uninterrupted lengths. The most common type is the circular duct diffuser with different types of openings for discharging the air. Flexible duct diffusers made of textiles or plastic are also available.

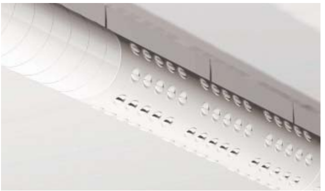


FIGURE 29. Duct diffuser fitted with adjustable nozzles.

FIGURE 30. Duct diffuser installed in a restaurant.

Important features:
- The design of the duct diffuser allows an evenly distributed air flow across the whole of the face area of the diffuser and ensures a low noise level and even spread pattern.
- Duct diffusers that are fitted with nozzles offer a high degree of induction. The nozzles facilitate quick and effective mixing of the room air in the supply air jet. This makes it possible to have a temperature difference between the room air and supply air that is larger than for most other types of diffusers, without creating draughts. The diffuser is therefore well suited for use with cooled supply air flows.
- The induction properties of duct diffusers fitted with nozzles means that they can manage variable flows as low as 20 % of normal flows.
- The spread pattern for a duct diffuser with rotatable nozzles can be easily changed while in operation without the set flow, pressure drop or noise level being changed. In addition, short or long throws can be set. Horizontal or vertical, as well as combined horizontal and vertical, spread patterns can be created. This type of diffuser is suitable for relatively high rooms.
- Available in models that can be mounted in corners of ceilings.
- Flexible textile or plastic ducts are light and can be easily repositioned to suit different furnishing alternatives.

Disadvantages:
- The cross-sectional area of the main duct connection can be a limiting factor with respect to the full flow capacity of the diffuser.

1b. Active ceiling diffusers

Active diffusers have openings that are continuously variable between fully open and fully closed. The openings are controlled by signals from room thermostats, presence sensors or CO_2 sensors. This type of diffuser is specially designed for use in demand-controlled ventilation systems.

FIGURE 31. Different types of active ceiling diffusers.

Important features:
- Always correct throw, irrespective of air flow.
- Manages very large under temperatures.
- Very high degree of induction.
- Very quiet in operation.
- Air flows can be preset in the factory.

Disadvantages:
- Require electric power.
- In certain cases the throws can be regarded as being excessively long but, as the flow rates vary, experience has shown that this, as a rule, is not a problem.

2., 3. and 4. Wall, windowsill and floor mounted diffusers

Back wall diffusers

Back wall diffusers are always mounted on the rear walls of a room and discharge towards the external wall, hence their name. They often offer an economical solution from an installation point of view, as the supply ducts are located in the corridor ceiling behind the back wall. Long throws are required when back wall units are used and a rule of thumb says that they should be about 75% of the depth of the room. The air jet should also be directed slightly upwards to make full use of the Coanda effect. This will also require a space greater than 100 mm between the top of the diffuser and the ceiling. It is also important that the air jet is not directed at 90° towards the ceiling.

Figure 32. Back wall diffusers.

Important features:
- Nozzle diffusers offer very flexible spread patterns and can be mounted eccentrically on a wall and still achieve an acceptable spread pattern by adjusting the nozzles.
- The throw can also be adapted to the required distance by adjusting the nozzles.

Grille diffusers for wall, floor and windowsill mounting

Grille diffusers have very low pressure drops and poor induction properties and are therefore not suitable when supply air is required at under temperatures. The grilles are available with fixed or adjustable vanes. Windowsill grilles are often used together with built-in perimeter induction units to cover the outlet opening and to control the direction of the air.

Important features:
- Can manage large air flows.

Disadvantages:
- Very poor induction capacity.
- Long throws.

Figure 33. Grille diffusers. Wall, window sill and floor mounted.

SUPPLY ATDS FOR THERMALLY CONTROLLED VENTILATION SYSTEMS

Supply ATDs for thermally controlled ventilation systems – known as displacement diffusers, low speed ATDs or low momentum ATDs – are also available in a number of different types. Again, we have to ask ourselves the question, “What are the requirements that good displacement diffusers must fulfil?” They should be able to:
- Discharge large volumes of air without creating near zones that are too large, which reduces the risk of draughts.
- Provide an even distribution of the air at low speeds across the whole of the discharge area, to reduce the risk of draughts and to reduce induction and noise levels to a minimum.
- Measure and regulate the air flow.
- Regulate the air flow in DCV systems.
- Change their spread patterns.

Figure 34. A thermally controlled, wall mounted displacement diffuser.

- Offer aesthetically pleasing designs to increase acceptance by architects.
- Withstand mechanical loads when place in the occupied zone.
- Offer simple and cost-efficient installation, commissioning and maintenance.

This means that the diffusers must have the following features:
- Air deflectors or adjustable nozzles for flexible spread patterns.
- A measuring outlet for measuring air flows for adjustment and control.
- A damper for adjusting the air flow.
- Effective internal noise absorption, to reduce the risk of cross-talk via the ducting system.
- A design that complies with building standards to reduce building costs.
- Flexible and adjustable spread patterns to facilitate different furnishing arrangements.
- Sufficiently robust construction, for positioning within the occupied zone.
- Aesthetically pleasing design for increased acceptance by architects.
- Robust mechanical construction.

When it comes to measuring and regulating the air flow, most displacement diffusers have a built-in measuring outlet and an adjustment damper in the connecting ducting is used to regulate the air flow.

The air should be discharged into a room at low speed through the whole of the face area of the diffuser, in order to eliminate unnecessary noise and undesirable induction of the room air, as well as to achieve the best possible distribution of the supply air.

As the air jets from displacement diffusers do not have to carry cold air, as they do in mixed ventilation diffusers, they are insensitive to variations of air flow. These diffusers are therefore well-suited for use in variable flow displacement ventilation systems. The variations in the air flow can be controlled by a motor driven damper installed in the duct connecting to the diffuser.

In the case of supply ATDs used in equalizing ventilation systems, the same applies as above, except that an induction unit is also included. This makes it possible to create a certain amount of mixing of the room air. Here, it is important to position the induction unit as low as possible

in the ATD, to keep the "clean" zone in the occupied zone as high as possible.

Displacement ventilatipon works best in premises with high ceilings. Typical types of premises include industrial buildings, shops, atriums, lobbies, classrooms, assembly halls, lecture theatres, cinemas, theatres, conference halls, arenas, sports centres etc.

In premises with lower ceiling heights, such as conference rooms, open areas in office landscapes, lounges etc, the diffusers can be used as a complement to other ventilation systems.

A wide variety of displacement diffusers is available, with the different designs depending on, among other things, the use of the premises, ceiling heights and room layouts, as well as the structural design of the building in question.

Flexible spread patterns

As displacement diffusers are normally placed in the occupied zone, it can be difficult to find suitable positions because of the locations of workstations or other pieces of furniture, the allocation of the floor area, etc. One way of making it easier to position displacement diffusers is to fit them with devices so that the spread pattern can be adjusted after the unit has been installed.

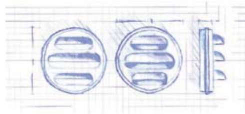


Figure 35. Rotatable air vanes in a displacement diffuser.

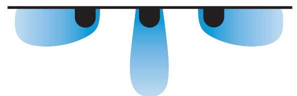


Figure 36. Adjustable spread patterns.

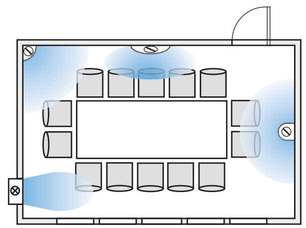


Figure 37. Standard spread patterns in a conference room.

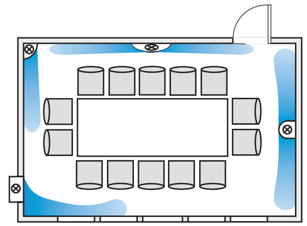


Figure 38. Adjusted spread patterns in a conference room.

Diffusers for use in displacement ventilation systems

FIGURE 39. Circular diffuser.

FIGURE 40. Semicircular diffuser.

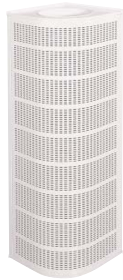


FIGURE 41. Quadrant diffuser.

Diffusers used in thermally controlled displacement ventilation systems can be designed in a number of different ways, for example, they can be completely circular, semi-circular or quadrant-shaped, as shown above. Diffusers like these can be designed so that the near zone is kept shallow. For example, there are semi-circular diffusers that discharge sideways rather than straight into a room. The diffusers can also be fitted with rotatable air deflectors behind the front panels. This means that completely circular diffusers do not have to discharge air radially over 360°, but can direct the flows precisely where needed.

Other types of displacement diffusers include bow-shaped and integrated wall units. Diffusers like these cannot discharge air sideways to any great degree, which means that they have a somewhat deeper horizontal spread pattern. On the other hand, they do not require as much floor space.

FIGURE 42. Bow-shaped diffusers.

FIGURE 43. A diffuser built into a wall.

Displacement diffusers are also available in a range of models to suit different needs in buildings with large open spaces, such as sports centres and industrial plants. These diffusers are more robustly built than those presented above but otherwise function in similar ways.

Diffusers are also available for ventilation systems in which the supply air can be discharged into the space under raised floor structures. These diffusers are used to supply air from a number of points equally distributed over the whole of the floor surface. This is a commonly used solution in cinemas, theatres, assembly rooms and congress halls etc.

FIGURE 44. Different types of displacement diffusers installed in raised floors.

Diffusers for use in equalizing ventilation systems

Equalizing ventilation systems were briefly discussed at the beginning of this chapter. These systems are used when the supply air temperature is too low to be used in a pure displacement ventilation system but where it would still be viable to use some of the advantages of displacement diffusers. The simplest way of doing this is to place the diffuser high up in the ventilated space and let the cold air entrain the room air as it falls into the occupied zone. A disadvantage here is that polluted air, due to the high position of the diffuser, is also drawn down into the room. In other words, it is very important to consider what type of activities can be allowed in the premises.

To ensure that the inducted room air is brought from the clean zone of the room, it might be necessary to fit the diffuser with a mechanical induction unit.

Previously in this chapter, we could see that it is not possible to supply air at an over temperature when displacement diffusers are used. There is, however, a special type of diffuser that is designed to do just this. The application requires the diffuser to be mounted high up and to be fitted with a supply air control function for variable temperature. The diffuser has two discharge settings, for horizontal or vertical flows, and these are chosen depending on the temperature of the supply air. The

FIGURE 45. Displacement diffusers with both horizontally and vertically oriented spread patterns.

change in setting from vertical to horizontal discharge is regulated by a thermostat in the ducting and a damper in the diffuser or by a thermally controlled damper in the diffuser. This type of diffuser is mostly used in industrial applications. Further information about thermally controlled ventilation systems can be found in the REHVA handbook, Displacement Ventilation in Non-residential Premises.

SUMMARY People spend up to 90% of their working and recreational time indoors. Together, we must ensure that indoor climates are both comfortable and healthy for all occupants in all types of buildings.

The type of space to be treated is always the starting point when designing an indoor climate system. All equipment installed in an indoor climate system is for the benefit of its users and their activities. This is why every individual product manufacturer must regard a building, to a greater extent than today, as a system. It is no longer sufficient to be an expert on the technology used in one's own products. We must become better at understanding how our own products affect, and are affected, by the other systems and services in a building and how we should develop our products so that the overall functioning of the building will be better. In other words, it's time to look beyond the traditional interfaces of engineering contracts. Every party involved in the building process must expand their views. It's not enough if each tends to their own. We must work together and from an earlier stage in the building process. In order to create satisfactory and cost-effective buildings with low energy requirements and comfortable indoor climates, all players must assume joint responsibility and analyse the consequences of the effects that their products and systems have on each other – not only from a technical point of view but also from a cost and energy use point of view.

Only by working together can we find the key to creating perfect indoor climates.

28. WATERBORNE INDOOR CLIMATE SYSTEMS

GUNNAR SVENSSON Regional Manager Middle East, Swegon AB

INTRODUCTION To create a good indoor climate, it must be possible to control the quality of the air, its temperature and its speed as well as any noise created in the process. When its temperature is not noticeable, its quality is acceptable and no one complains about draughts or annoying noise – only then can the indoor climate said to be satisfactory. The media used to supply or remove heat are air and water. Air has an advantage, as it can be used to change both the quality of the air and its temperature. Water can only be used to change the temperature of the indoor air and has to be used in conjunction with fresh air to change the quality of the air. Water, on the other hand, is a far better energy carrier than air, thanks to the physical differences between the two media. The density of water is 1000 kg/m^3 while the corresponding figure for air is only 1.2 kg/m^3. In addition, the specific heat capacity of water is 4.18 kJ/(kg · °C) compared to 1.0 kJ/(kg · °C) for air.

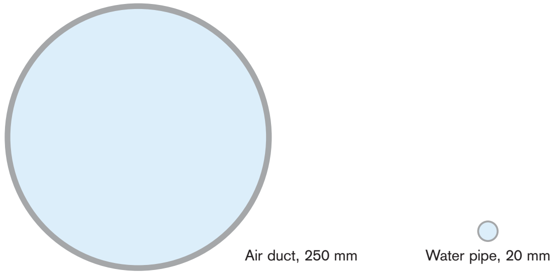


FIGURE 1. A 250 mm air duct and a 20 mm water pipe can carry the same amount of energy at standard air and water flow rates.

Waterborne indoor climate systems use a combination of air and water, the air being used to regulate the quality of the indoor air and water to maintain the correct temperature. The products and equipment chosen must fulfil stipulated requirements with respect to draughts and noise levels.

Historical background

At the end of the 1950s, the Norwegian engineer Gunnar Frenger patented the so-called Frenger System, based on horizontal aluminium panels attached to a piping system, in which hot or cold water was circulated. Via radiation and convection, water can either absorb heat from or emit heat to the surroundings. The system is based on water, in small volumes, being able to carry large amounts of energy. The system is often known as a radiant ceiling.

In order to increase the capacity of the system, the next step was to develop a product with vertical components, in which the convection capacity was increased while the radiation capacity remained constant – a so-called radiant heating panel or beam. The first units were installed at the beginning of the 1970s.

The next step was to integrate a converter into the beam, creating the so-called passive beam.

To increase the capacity even further, the supply air system was also integrated into the beam and most of the cooling capacity was then provided by induction. The so-called active beam had been created.

Radiant ceiling beams for heating and cooling

Most radiant ceiling beams are made up of steel or aluminium panels. Units made of aluminium are more efficient at transferring heat than those made of steel.

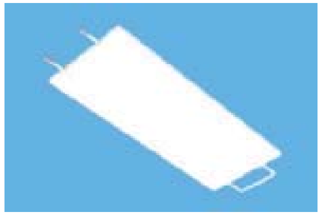


FIGURE 2. An aluminium panel with integrated copper piping.

Function: When used for heating, hot water is circulated in the piping. The heat carried by the water is transferred to the piping and then to the panel via conduction. The panel becomes warm and transfers its energy via radiation to the surrounding surfaces and via convection to the air at lower temperatures than the panel.

The proportions of heat transferred by radiation and convection are 70% and 30% respectively.

When used for cooling, cold water is circulated in the piping and the panels become cooler. Heat is then absorbed from the surrounding surfaces and from air at higher temperatures than the panel.

CLIMATE BEAMS Beams for cooling, heating and ventilation, so-called climate beams, are suitable for use in both new and renovated buildings and in a wide variety of environments, including:
– Individual offices
– Open plan offices
– Hotels
– Shopping centres
– Banks
– Schools

Climate beams offer high levels of thermal comfort by combining the advantages of water as an energy carrier with fresh air to ensure the quality of the indoor air. When compared to all-air systems for cooling and heating, the air flows can be kept much lower, which means that fan rooms, duct shafts and suspended ceiling space can be minimized.

As the system is practically static, i.e. with only a few moving parts, the maintenance required is minimal.

Systems for cooling using climate beams must always be designed as dry systems. If dehumidification of the outdoor air is required, this must be carried out in the air handling unit and the flow temperature in the piping must then be above the dew point of the air. This means that no condensation will occur in the room where the unit is installed and that there will be no need for a condensation system or condensation pumps, which also means less maintenance.

When used for cooling the climate beams are dependent on a supply of cold water, which means that the chiller must be started as soon as cooling is required. Unlike in all-air systems, it is not possible to use cool outdoor air for cooling purposes to the same extent, i.e. by using so-called free or summer night cooling.

Climate beams can be divided into three main categories:
– Passive beams
– Active beams
– Fully integrated beams, so-called comfort modules

To be able to comply with standard comfort requirements when cooling is used, the power rating ranges given below can be regarded as maximum cooling powers per m^2 floor area. If the cooling power is higher than theses limits, there will be an increased risk of discomfort due to so-called cold radiation or draughts.

- Passive beams, 60 to 80 W/m^2
- Active beams, 100 to 110 W/m^2
- Comfort modules, 120 to 140 W/m^2

General descriptions of the different types of climate beams
Passive chilled beams. In principle, a passive beam comprises a casing of sheet metal with a built-in finned pipe arrangement and a perforated base plate. This type of beam, known as a chilled beam, is only used for cooling.

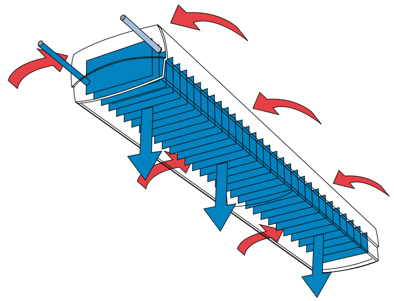


FIGURE 3. Air flows created by a passive chilled beam.

Function: The cooling capacity of a chilled beam is primarily provided by natural convection and a small proportion of radiation. Cold water circulates in the piping and comes into contact with the surrounding warm room air, which is cooled and becomes heavier than the surrounding air. The air flows past the piping from above and descends into the room.

The process continues as long as there is a temperature difference between the cooling pipes and the room air. The only way in which the capacity of the beam can be changed is by changing the temperature difference between the air and the cooling surfaces in the beam – either by changing the water flow rate or by changing its flow temperature.

Passive chilled beams are not connected to the ventilation system. Supply air is introduced via separate supply ATDs, air terminal devices, of either mixing or displacement type. The supply air system must be designed so that it does not affect the function of the beams. If ceiling diffusers are used, they must be positioned so that they cannot interrupt the convection currents around the beam. Otherwise, there is a risk that the function of the beam will be affected: if the convection currents increase, this will cause draughts and if they decrease, cooling capacity will be lost.

Passive beams can also be used in conjunction with systems in which displacement ventilation is used. This arrangement allows the down-flowing convection currents and air supplied to the occupied zone to be well mixed.

The beams can be installed either fully visible or integrated into suspended ceilings.

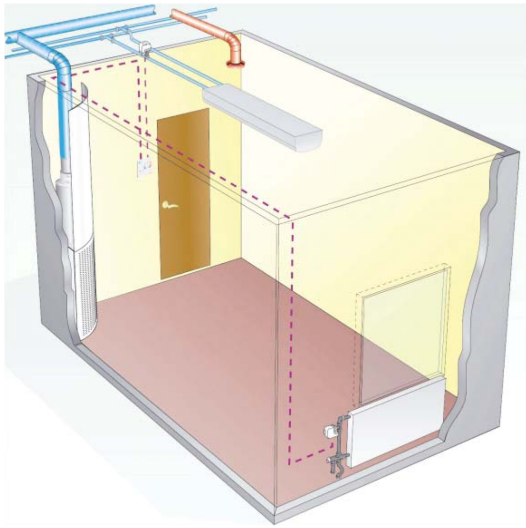


FIGURE 4. An individual office with a passive chilled beam.

When the beams are integrated into suspended ceilings a return air grille will also be needed, so that the room air can circulate.

To avoid draughts from a passive chilled beam, it is recommended to position it to one side of a workplace.

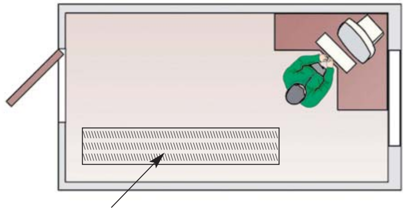


FIGURE 5. Correct positioning of a chilled beam.

Active climate beams. Active climate beams comprise supply air chambers, finned pipes, supply air nozzles, supply air openings and perforated base plates for air circulation, as shown in Figure 6. This type of beam can be used for both heating and cooling.

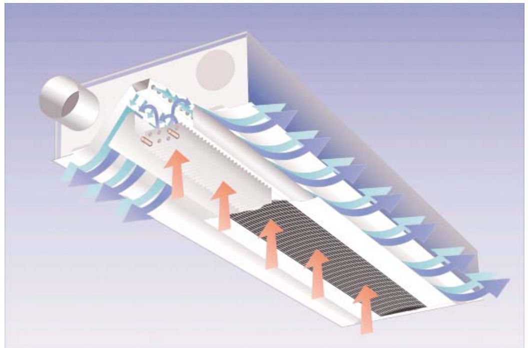


FIGURE 6. Air flows to and from an active climate beam.

Function: Primary air from the supply air system enters the unit via one or a number of connections in the end plates or side plates. Strategically placed nozzles force the air to flow past a heat exchanger in the form of cooling/heating coils. The high exit speed from the nozzles creates an under-pressure just above the coils. This under-pressure is used to induct warm room air. The primary air is mixed with the cooled inducted air and is ejected into the room parallel to the ceiling. The so-called Coanda effect helps the cooled air to hug the ceiling, maximizing the size of the mixing zone, an important feature if draughts are to be avoided.

The cooling capacity of an active beam is determined by the temperature difference between the room and the water, as well as by the air flow and nozzle pressure.

Active climate beams are also available with integrated heating functions. In this case a 4-pipe system is used, in which there are two separate coil arrangements, one for circulating the cold water and one for the hot water. The temperature is controlled by valves regulated by a room regulator, which operates sequentially. There is normally a dead zone of 1 or 2°C between the cooling and heating functions to prevent their simultaneous operation.

Active climate beams can be positioned in a number of different ways in a room:

A. At right angles to an external wall

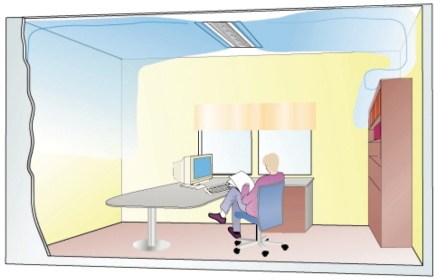


Figure 7. An active climate beam positioned at right angles to an external wall.

In a normal office module, the beam is usually placed centrally at right angles to the external wall. This provides a good solution, as large areas can be used as mixing zones and the under temperature air will have plenty of time to mix with the warm room air. Excellent opportunities exist here to vary the air flow without creating problems with draughts. A bookcase, see Figure 7, will not normally cause any problems, as the air speed will already be low when the air current reaches it and also because the temperature of the air jet has had time to increase thanks to the induction of the room air. The installation is simple as there is only a small distance between the end plate of the cooling beam and the corridor.

A centrally placed beam offers great freedom when it comes to altering the position of the walls.

B. Parallel and close to an external wall

Figure 8. An active climate beam placed parallel to an external wall.

Another variant is to place the beam close to an external wall. This solution offers advantages from a comfort point of view when compared to a beam placed close to a corridor wall. By distributing the air asymmetrically, with the largest proportion being directed towards the corridor wall, maximum use can be made of the heat loads created at the desk and window to create a long mixing zone. The distance between the connecting points and the corridor is, in this case, increased and this will require more installation work.

C. Parallel and close to a corridor wall

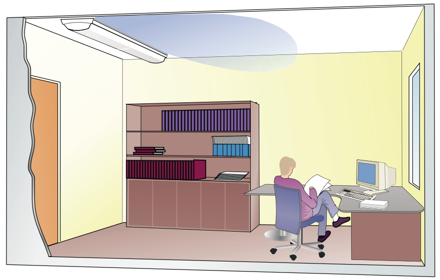


Figure 9. An active climate beam placed close to a corridor wall.

As a third alternative, an active beam can be placed close a corridor wall. If this solution is to work satisfactorily, it must be well planned with regard to the distribution of the mixed air. Tests have shown that the best effects can be achieved if 50% of the air flow is distributed along the ceiling and 50% along the corridor wall. However, opposing air flows from heat sources, such as desk equipment or the windows, could still cause problems. These flows could cause the air jet to deflect down into the occupied area and increase the risk of draughts.

To ensure a good climate, beams with adjustable deflectors should be used.

From an installation point of view, the corridor wall solution is just as advantageous as the centrally placed solution, as the connection points are close to the corridor.

D. Open plan offices

FIGURE 10. Active climate beams in an open plan office.

The type of room that requires the greatest care in the planning stage is the open plan office. Climate beams with air flows directed towards each other, and with short mixing zones, require special attention. Although wide ranges of air flow rates are acceptable when centrally placed beams are used in individual offices, the situation here is quite the opposite. To avoid undesirable down draughts of cold air, the nozzle pressure must minimized. This is achieved by using different nozzle configurations. To further ensure a good degree of comfort, air flow deflectors can be used. Using these, different air flow patterns can be created and these will then counterbalance the negative effects of opposing air jets. In open plan offices, there are often demands for high flexibility with respect to movable walls etc. This must be taken into consideration when planning the ventilation system and if the above-mentioned recommendations are followed, then there are good chances of creating a draught and noise free environment.

Comfort modules. The basic functions of comfort modules are closely related to those of active chilled beams.

The main difference is that comfort modules distribute air in four directions instead of two. This maximizes the space available for mixing the supply air with the room air. This, in turn, means that more cooling or heating power can be supplied to a limited ceiling area.

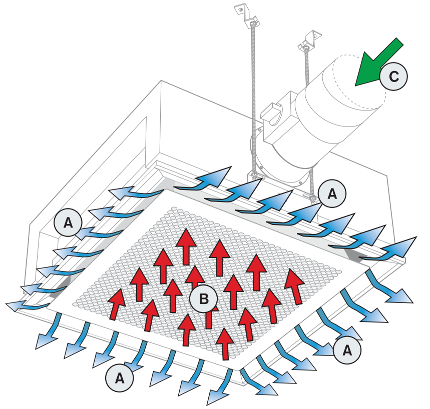


FIGURE 11. Air flows to and from a comfort module. A. Tempered air B. Inducted room air C. Primary air from the air handling unit.

Function: The comfort module is connected to the supply air system. Primary air from the supply air system is supplied to a plenum fitted with nozzles. When air is blown into a room through the nozzles the room air is inducted into the cooling/heating coils. This means that a mixture of supply air and cooled, or heated, circulation air is ejected into the room to provide the correct temperature.

The air opening in the comfort module is designed to provide a high degree of turbulence for quick mixing of the room air, to provide maximum comfort. The unit has a four-way spread pattern and each side can be individually adjusted. Even the amount of air flowing out from each side can be adjusted individually.

Recommended design values: Pressure drop across the nozzles from 50 to 150 Pa (in heating mode not less than 70 Pa).

Comfort modules are available for integrated or visible installation, see Figures 12a and 12 b.

FIGURE 12A. Comfort module, integrated installation.

FIGURE 12B. Comfort module, visible installation.

As mentioned above, the comfort module is also suitable for heating rooms. The design of the outlet opening means that the supply air will be well mixed with the room air, even when used for heating. Figure 13 shows results from laboratory measurements. It can be seen that the temperature gradient in the occupied zone is less than 2 K.

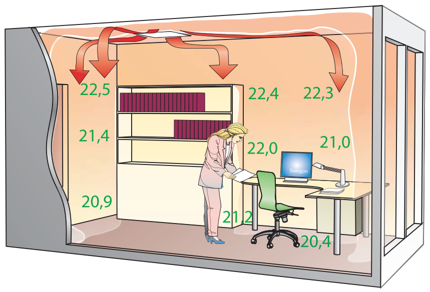


FIGURE 13. Temperature distribution in a room when a comfort module is used for heating.

Induction units for external wall or ceiling installation

OTHER UNITS FOR WATERBORNE INDOOR CLIMATE SYSTEMS

Induction units for external wall or ceiling installation are based on the same principles as active climate beams or comfort modules. The difference lies in where they are placed. External wall units are normally installed under windows, while ceiling mounted induction units are normally installed horizontally, above suspended ceilings.

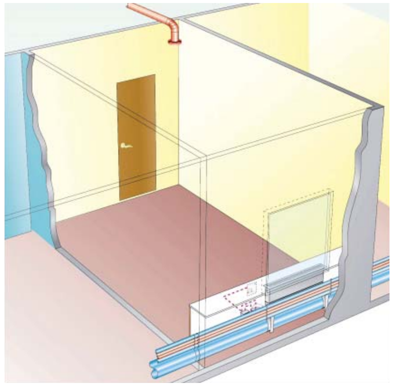


FIGURE 14. An induction unit for cooling, heating and ventilation placed next to an external wall in an office module.

Function: Primary air is distributed via the ducting system to the units. The duct pressure forces the primary air through a number of nozzles after which it passes over a heat exchanger. This creates an underpressure, which causes the room air to be sucked into the heat exchanger by induction. There are two separate coils in the heat exchanger, one for circulating cold water and the other for circulating hot water. Depending on how the module is used, the room air is either cooled or heated in the heat exchanger before it is mixed with the primary air in the outflow spigot. In a wall unit, the mixed air is distributed upwards along the external wall to the ceiling, where the Coanda effect helps the supplied air to mix with the room air. When neither cooling nor heating is required no water at all is circulated in the coils. The only cooling or

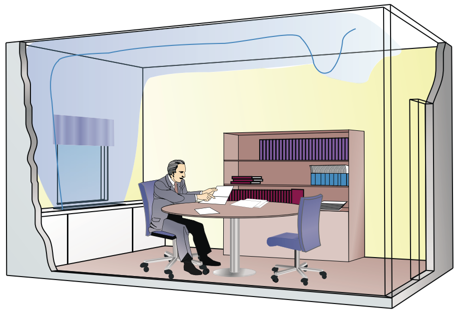


FIGURE 15. The air stream from an induction unit.

FIGURE 16. An induction unit with factory installed room control equipment.

heating effect will then depend on the temperature of the primary air. The units can also be ceiling mounted.

Fan convectors

A fan convector is quite simply a cooling, or heating, unit comprising a finned water element and a fan enclosed in a casing. The lower part can be designed as an ATD. Alternatively, one or a number of ATDs can be connected via ducting. The fan, which can be regarded as the generator in the system, draws in air from the room and forces it past the element. Fan convectors are not connected to the supply air system in a building and, for hygienic reasons, the air required has to be supplied separately.

Normally, cold water is supplied at a temperature below the dew point, which means that condensation takes place in the unit and this, in turn, has to be drained off. Fan convectors normally have higher power ratings than climate beams or comfort modules but, on the other hand, require considerable maintenance, see the section on maintenance below, and also generate a lot of noise.

Fan convectors are available for installation on ceilings and walls, as well as above suspended ceilings.

FIGURE 17. A fan convector designed as a ceiling cassette. The circulating air from the room is sucked in from directly beneath the unit and is discharged into the room, either cooled or heated, via the four supply air openings.

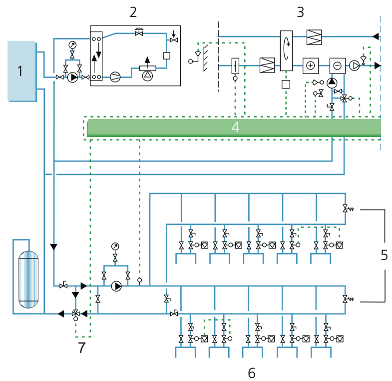
FIGURE 18. A ceiling mounted 4-pipe fan convector unit. Two or more supply ATDs can be connected to the unit. The supply air is blown out above the suspended ceiling and is then sucked into the fan convector together with the air inducted from the space below.

SYSTEM DESIGN TO PREVENT CONDENSATION

A waterborne indoor climate system with climate beams or comfort modules should always be designed so that the moisture in the indoor air cannot condense on the cooling surfaces. This means that the temperature of the cooling water must always be kept above the dew point of the indoor air.

One of the following solutions should therefore be considered, to prevent condensation. It may suffice if the solution is based on one of the three alternatives below but it can sometimes be advisable to combine two of them. The size of the plant will depend on the solutions chosen.

Alternative A. When the supply air has to be dehumidified on demand. This is best achieved by cooling in the central air handling unit.



1. Storage tank
2. Chiller
3. Air handling unit
4. Control system
5. Overflow valve (can be replaced by a pressure regulated pump)
6. Climate beams
7. Main shunt

FIGURE 19. Schematic of the system design when dehumidification is required.

Alternative B. The flow temperature of the cold water is regulated by using a shunt and the water is supplied to the cooling units at a temperature above the dew point of the room air. A dew point sensor is placed in the extract air at a representative location to measure the humidity of the air. If the dew point of the air is higher than the supply water temperature, a signal will be sent to the shunt that then raises the flow temperature above the dew point.

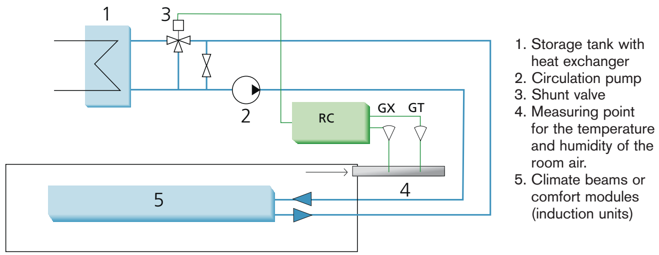


FIGURE 20. Schematic of the solution using a centrally placed humidity sensor to regulate the main shunt, so that the flow temperature is higher than the dew point of the room air.

Alternative C. When local increases of humidity can be expected, for example, in a room with high occupancy rate or where windows can be expected to be opened. In this case, a dew point sensor can be installed in the room itself. With the help of this sensor, and if there is a risk of condensation, the control system can close the valve so that the temperature of the cooling surfaces increases. The valve is reopened when the humidity in the room has fallen to a level where there is no longer a risk of condensation. Controlling a system like this means that there is no cooling capacity when the valve is closed.

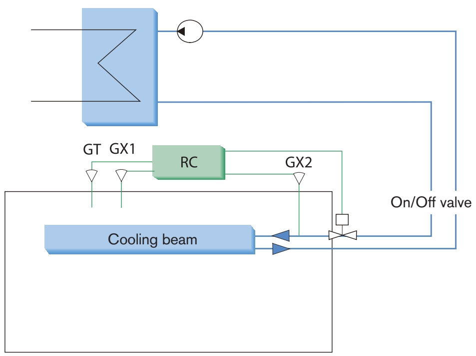


FIGURE 21. Schematic of the solution when a dew point sensor is placed in a room.

When fan convectors are used cold water is normally supplied at a temperature lower than the dew point of the room air. This means that condensation will take place inside the fan convector and it will have to be connected to a run-off system for the condensate. If piping cannot be installed to provide natural run off, a condensate pump will have to be installed.

ROOM CONTROLLERS

The purpose of room control equipment is to adapt the capacity of the air conditioning system in a building to the required heating and cooling needs. The most common solution is to let the supply air temperature remain constant and then regulate the room air temperature via the units located in the different rooms, for example, climate beams and comfort modules.

It is advisable to adjust the supply air temperature according to the outdoor air temperature, which means that the supply air temperature will be lower during the summer than in the winter.

Two basic methods can be applied:

- Zone regulation
- Individual room regulation

Zone regulation means that the temperature is regulated in each separate zone, for example, a group of rooms, whereby a temperature will be reached that can be regarded as a compromise. If loads vary between the different rooms, this can mean that some rooms will be too cold while others will be too hot. The temperature is then often adjusted according to the coldest room, which means, in turn, that a number of rooms will be too hot.

When individual room controls are used this means that the occupants can regulate temperatures themselves. Irrespective of whether there are different cooling or heating needs in the different rooms, the occupants can always choose their own desired temperatures – on condition that the system has been correctly designed and commissioned.

Comfort modules are often supplied with factory installed control equipment, which considerably simplifies the installation work.

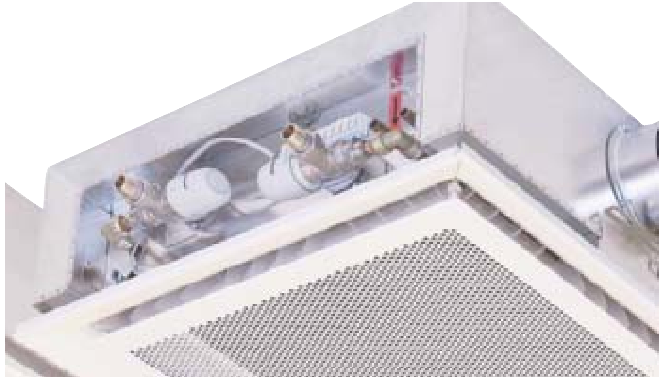


Figure 22. Control equipment for heating and cooling, factory installed in a comfort module.

Climate beams, comfort modules and induction units

MAINTENANCE Climate beams, comfort modules and induction units have few moving parts and therefore require a minimum of maintenance. The only moving parts are the control valves and actuators.

Depending on the type of installation, a cooling/heating coil unit can require vacuum cleaning every 2 to 5 years. There are no filters in the units as the fin separation is often as much as 3 to 5 mm. The particles that are sucked in with the circulating air are so small that they will pass through the coils.

This means that filter changes and fan servicing will only be required in the central air handling unit. These systems are also dry systems, which means that neither condensate drainage systems nor condensate pumps are required.

Fan convectors

Fan convectors normally comprise a filter, a fan, cooling coils and automatic control devices. A condensate pump is also normally included. Filters must be changed at suitable intervals and, depending on the use of the premises, this usually means once or twice per year. In order to provide the desired capacity, the fans will also have to be serviced, preferably when the filters are replaced – but not every time. A suitable service interval is about two years. The functions of the fans and condensate pumps should also be checked and drainage connections checked for blockages.

SUMMARY Active climate beams, comfort modules and induction units described in this chapter have the common advantage of being able to provide a quiet supply of tempered air to a room, while only requiring a minimum of maintenance. By combining the advantages of water as an energy carrier with air, to ensure the quality of the indoor air, the best of both worlds can be enjoyed in these air-conditioning systems.

Fan convectors also belong to the category waterborne indoor climate systems but, just as in the case of passive beams, require a separate system for supplying air to ensure good air quality. As fan convectors have greater service and maintenance needs, service personnel must be able to access the rooms where the units are installed. On the other hand, fan convectors have a greater power rating than correspondingly large climate beams or comfort modules, which means that they can more quickly compensate for rapid increases in heat loads.

REFERENCE For further information about climate beams, see REHVA Guidebook no. 5, Chilled Beam Application Guidebook [www.rehva.eu].

29. BALANCING VENTILATION SYSTEMS

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INTRODUCTION

The process of balancing a ventilation system and the planning of this work during the design phase are key to creating conditions so that the design air flows can be maintained. Commitment and competence are two qualities that both the design engineer and the commissioning engineer must possess, if successful results are to be obtained.

Of course, it would be very convenient if we could skip the balancing work. It takes time and costs money. Experience data from the Nordic countries [NVG, 1981] shows that costs for balancing a traditional ventilation system, with both supply and extract air fans, can account for 2 to 4% of the total cost of the installation. The costs, however, are greatly dependent on how well the system has been adapted to the requirements of the commissioning method to be used.

The commissioning work itself is not only a question of balancing air flows. The distribution patterns from the different supply ATDs, air terminal devices, must also be adjusted so that draughts can be avoided. The costs for this adjustment work are not included in the figures above.

The engineers who carry out the commissioning work make up the last link in the chain of specialists involved in a ventilation contract. These are the engineers who have to make sure that the functional requirements, originally specified for the system by the client in consultation with the project engineer and according to which the design engineer designed the system, will now be met.

The commissioning engineer must obviously have sound knowledge of measuring and balancing techniques, as these are essential ingredients of the main assignment. Furthermore, the engineer must also have a good working knowledge of other aspects of ventilation technology, for example, air flow dynamics and acoustics.

An understanding is also required of how the flow patterns in the ducting affect the properties of components, such as ATDs and dampers, with regard to pressure drops and noise creation. The products used and their properties must also be familiar.

The design designer must also embrace all this knowledge, as he is responsible for designing the system and must, therefore, ensure that the correct products are chosen with respect to the specified functions. Different products, for example, have to be connected to different lengths of straight ducting in order to function properly. The commissioning engineer must, therefore, confer with the design engineer, if he discovers something that could cause incorrect flow patterns in the ducting and which would subsequently cause difficulties in the commissioning work.

Experiences from a large number of training courses in balancing techniques have, unfortunately, provided a rather negative picture of the level of competence displayed by commissioning engineers. In order to tackle these and other problems, the National Swedish Organisation for Ventilation Balancing, RSVI, was formed in 1998. One of the main aims of the association is to provide information about commissioning techniques. A handbook was published in 2005, to provide members with easily accessible information about current developments in this field [RSVI, 2005]. Another aim was, and still is, to encourage certification of active commissioning engineers and to act as advisers to consultants and clients.

A general problem to be dealt with concerns the recommended methods for measuring, referred to in the Swedish building regulations [Johansson and Svensson, 1998] and on the agenda of the TC156/WG4 working group of the European Committee for Standardization CEN, which are rarely used correctly in practice.

The recommended methods have often been simplified, so that the work can be carried out in a shorter time. Carrying out measurements accurately is a difficult task and, to meet the normal tolerances required in Sweden (±15% including measuring errors), reliable and approved methods must always be used.

Bearing all this in mind, it is clear that new measuring methods must be developed. However, it is easy to find oneself at a dead end, as it is not possible to simplify the methods much further, if traditional system design concepts are retained. The development of balancing methods must therefore proceed hand in hand with work towards improved system design. Developments should aim at the elimination of manual balancing, as used today, and the introduction of automatically distributed air flows by using active components in the system. Checking to ensure that the flow distributions are as specified could then be easily limited to simple pressure measurements in the different parts of the system.

In Sweden, we have long experience of how different system solutions and their practical applications have performed over long periods of time. And, since Sweden is unique when it comes to compulsory ventilation inspections, our experience will most probably have great significance for other countries.

System imbalances and noise have always been common problems and research has shown that we must pay much more attention to the design of ventilation systems [Engdahl, 2002].

Measurements show that air flows, after a few years of operation, seldom match the specified design rates. This is often due to fouling and contamination but also to the fact that it might never have been possible to balance the systems correctly, due to faults in the original system designs.

As a result, a prioritised requirement for many years has been that it must be possible to carry out simple checks to confirm that the air flows in the system are balanced and that these checks must be carried out regularly.

It is important that we learn from these findings, mainly from the Nordic markets, if we want to steer development towards more stable and functional systems.

When contracting balancing work, the areas of responsibility for the engineers must be clearly stated and all the necessary documentation must be presented. This means, for example, that pressure drop calculations, as well as nominal flows and tolerances, must also be included. Noise levels and maximum allowable velocities in the occupied zones must also be specified.

It should be pointed out that the scope of the balancing work, i.e. planning, preparations and execution, as well as the time required and costs, are all highly dependent on the specified allowable tolerances.

The functions of each sub-system must also be fully described and formulated in such a way that the commissioning engineer can clearly see how the installation is intended to work.

It must be clearly stated for which balancing method the installation has been designed. All measuring points, temporary and fixed, must be specified and their positions clearly marked in the drawings. It must also be stated which officially approved measuring methods are to be used.

THE DESIGN ENGINEER HOLDS THE KEY

Balancing work requires competent personnel and the use of recognized measuring methods. However, it can only be successfully carried out if the installation is designed to facilitate measurement and adjustment of air flows in the various branches of the air distribution system and in the ATDs. Installations in which relevant demands have been met will require less time for balancing and will, in most cases, also have lower operating costs.

The design engineer must, therefore, ensure that:

1. The installation can be balanced. The engineer must, where possible, specify symmetrical ducting so that excessive dimension changes in the ducting can be avoided. It must be clearly stated in the contract documents for which balancing method the installation has been designed.

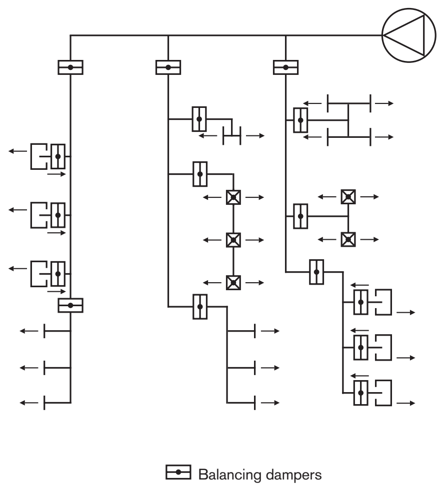


Figure 1. An installation in which the design engineer has taken into account the properties of the ATDs. Balancing dampers between the different groups of devices ensure that the pressure drops across them are minimized.

2. The installation can be checked on completion. The engineer must, therefore, choose suitable measuring methods for these checks. To simplify the measuring procedures, permanent measuring devices should be installed in the branch ducting and main ducting. Supply and extract ATDs should also have fixed measuring devices.
3. The installation can, at a later date, be checked by service and maintenance personnel without having to alter or remove any permanent fixtures or parts of the building.
4. The maximum allowable tolerances and deviations from the specified values are recorded, if they exceed the normal tolerance requirements. When an allowable deviation is specified, the probable measuring error must be included in the deviation.
5. The ducting system is correctly designed from a flow point of view. This means, among other things, that the shortest distance between a flow restriction, such as a 90-degree bend, and a branch duct must be at least six times the duct diameter. Supply ATDs must be connected to the ducting so that the specified design throws and distribution patterns are achieved.
6. No terminal devices are connected directly to a main duct.

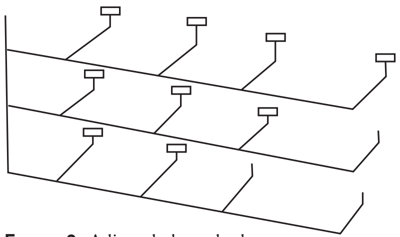


FIGURE 2. A linearly branched system.

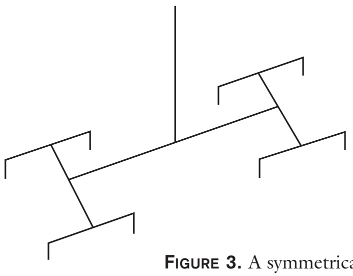


FIGURE 3. A symmetrically branched system.

Symmetry

Uniformity and, preferably, symmetry in ducting design are basic requirements for creating a balancing-friendly installation. Figure 2 shows a linearly branched (uniform) system and Figure 3 a symmetrically branched system.

These two alternatives are quite common and offer solutions that provide good air distribution and small pressure losses. In the Nordic countries, it has been noted that supply ATDs combined with measuring and adjustment functions offer great advantages. The installation of supply ATDs via connection boxes is now standard.

Figure 4 shows a complete, traditionally designed system with dampers and ATDs, with sound attenuators after every damper.

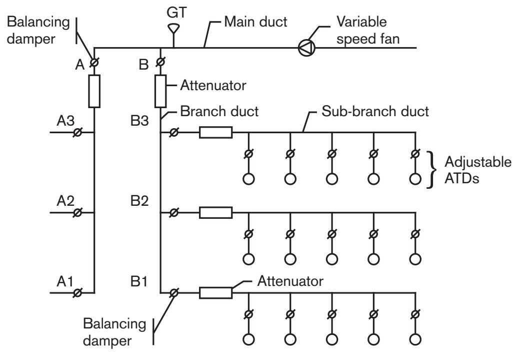


FIGURE 4. A balancing-friendly system structure.

Connection boxes

In Figure 4, it is assumed that the supply ATDs are connected to the ducting via connection boxes. A connection box should:
1. Include a damper to regulate the air flow
2. Include a measuring device for flow measurements
3. Be able to attenuate noise from the ducting system
4. Generate very little self-noise
5. Distribute the air to the supply ATD so that it can create the desired distribution pattern.

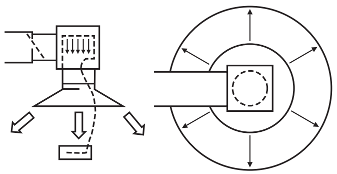
Dampers must be installed in main ducts, branch ducts and sub-branch ducts, so that the necessary pressure drops can be distributed throughout the system. The pressure drop across every damper can then be minimized, which is an advantage from a noise point of view. A balancing damper in the connection box also makes it possible to distribute the correct air flow through an ATD without causing noise problems.

When fitting connection boxes to the ducting system, it is essential to follow the installation instructions supplied by the manufacturer. Depending on the measuring method and the distance to the nearest upstream restriction, different lengths of straight sections of ducting are required, if measuring tolerances are to be met. If these requirements regarding straight sections are disregarded, large flow measurement discrepancies will result [CEN 1999].

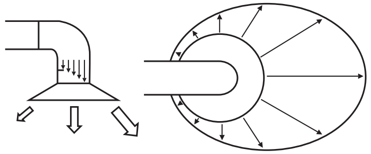
Correct distribution patterns

Another important task for the design engineer is to ensure that the supply ATDs can create the correct distribution patterns, so that the desired air movements in different rooms can be achieved. This will require steady air flows and it is therefore not advisable to attach a supply ATD directly onto a 90-degree bend or a T junction.

Supply air terminal devices should, therefore, always be attached to suitable connection boxes.



Supply air terminal device attached to a connection box



Supply air terminal device attached to a 90-degree bend

FIGURE 5. The distribution patterns from supply ATDs with and without connection boxes.

DIFFERENT BALANCING METHODS

The commissioning engineer cannot normally influence the choice of balancing method, as this will already have been chosen by the design engineer when planning the installation. However, if the commissioning engineer discovers faults in the installation, it is his duty to make this known before the balancing work starts.

A number of different balancing methods are used today, including:

1. The proportional method.
2. The presetting method, i.e. the dampers and terminals are preset according to pressure drop calculations.
3. A combination of these, comprising proportional balancing and presetting techniques
4. Own methods used by commissioning engineers.
5. Methods used for modern system solutions.

Proportional balancing method

The proportional balancing method [NVG, 1981] is a systematic balancing method. Unfortunately, knowledge of the method among commissioning engineers is limited and this means that it has often been used incorrectly. If the instructions for this method are not strictly followed, step-by-step, it will, in principle, be impossible to keep within normal flow tolerances. Simplified versions of the proportional balancing method should, therefore, always be avoided.

Presetting method

The presetting method requires carrying out very accurate pressure drop calculations for the ducting system. The system must also be installed exactly according to design. The commissioning engineer's task, when this method is used, is to set the dampers and ATDs according to the design engineer's instructions. The only other work that the commissioning engineer then has to carry out is to check that the design flows have been achieved. If any deviations are noted, these must be discussed with the design engineer, so that necessary steps can be taken. As it is often difficult to know exactly how the ducting will be installed in a building, this method is seldom used. On the other hand, it could be advantageous to use the presetting method for extract air systems with relatively low air speeds in the ventilation ducts and in which the pressure drops across the extract ATDs are relatively high.

Proportional and presetting methods combined

This is an excellent combination and one that should be used more often. However, it does require the design engineer to be very accurate in his work. It also means that the design engineer's choice of products must not be changed.

Own methods
A commissioning engineer's own private methods must not be used. Only a systematic, step-by-step method must be applied, otherwise it will be impossible to maintain the required flow tolerances.

Methods used for modern system solutions
Methods used for modern system solutions, i.e. those that contain pressure control devices, often mean that the commissioning engineer is only required to check that the system is functioning as intended.

Today, we know that we can advance quite far along the road to balancing-free installations. These require active components that are able to manage the distribution of the air and to compensate for any faults and operational disturbances. A method has been described in which the balancing work can be reduced to a minimum [Engdahl, 2002]. This method also leads to more flexible and energy-saving installations.

How the balancing of these systems is carried out varies depending on how the installation has been designed.

THE PROPORTIONAL METHOD

In Sweden, the proportional method has been used for many years, though it was originally developed in the 1960s in England. The method is based on the relative magnitudes of the air flows in the branch ducting being constant, even if the air flow in the main ducting is changed, see Figure 6.

This principle is extremely useful when applied to systematic balancing procedures. It means that the ratios of measured air flows to design air flows through the different ATDs and branch ducts can be gradually

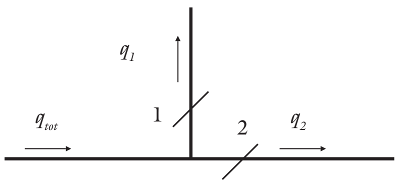


FIGURE 6. The principle behind the proportional method of balancing. The relationship between q_1 and q_2 is constant and independent of variations in the flow q_{tot}, provided that the settings of dampers 1 and 2 are not altered.

adjusted to their correct values. Consequently, during most of the balancing work, it is not necessary to measure the absolute air flows. Relative air flows are quite sufficient, which simplifies the measuring procedures and reduces the time needed for balancing.

One advantage of this method is that it can be used for balancing a number of different types of ventilation systems. The method also means that:

- The balancing work can be divided into a number of independent steps. A group of ATDs can, for example, be balanced independently of the conditions in other parts of the system. Separate parts of the system can, therefore, be balanced before the whole installation has been completed.
- The pressure drop in the system will be the smallest possible and the noise level can, therefore, be kept to a minimum.
- It can be seen at an early stage whether a particular device or part of the system cannot achieve the design air flows.
- Dampers and ATDs, once balanced, will not require any further adjustment during the balancing process.

The method is extremely simple and efficient but does demand a great deal of the components used in the system. For instance, dampers and flow measuring devices must be easily accessible, if the balancing work is to be carried out quickly. The easiest way to achieve this is to fit all the supply and extract ATDs with permanent measuring devices and dampers. If these functions are not installed, the ATDs must be connected to connection boxes fitted with the necessary functions. Branch ducts and main ducts must also be fitted with balancing dampers and, preferably, with measuring devices.

Balancing ventilation installations requires precision work. It is therefore important that all the dampers and terminal devices can be locked in their balance positions and that no unauthorized persons can alter them. However, experience does show that adjustments are often necessary after the system has been in operation for a while. There are numerous reasons why and one of them, little known to most, is due to the effects of thermal forces on the balanced flows. Unfortunately, notice is rarely taken of these changes, caused by the different thermal forces created in summer and winter. If the balancing work cannot be carried out during more neutral periods, such as in spring or autumn, it will be necessary to compensate for these forces, especially in high-rise buildings. This is often one of the reasons why adjustments often have to be made to damper and terminal device settings, to deal with complaints about draughts and alike that often occur in winter and summer.

The thermal force Δp can be calculated from the following equation:

$$\Delta p = \rho \times g \times h \times \Delta t / T \quad \text{Pa} \qquad (1)$$

where:
ρ is the density of the indoor air in kg/m^3
g is the acceleration due to gravity in m/s^2
h is the height of the building in m
Δt is the temperature difference between the indoors and outdoors in K
T is the temperature indoors in K

In winter, if the outdoor temperature varies between –5 °C and –15 °C, the equation above can be approximated to:

$$\Delta p = 0.045 \times h \times \Delta t \quad \text{Pa} \qquad (2)$$

This means, for example, that in a four-storey building with a storey height of 3 m, the thermal driving force, when it is –15 °C outdoors and +22 °C indoors, will be about 20 Pa. Depending on the pressure drop in the system at this particular time, this force can have an effect on the flow balance.

If this force affects the supply air system that has been balanced during the warmer part of the year and the balanced pressure drop is 50 Pa, the pressure drop across the most distant ATD, in winter, will be 30 Pa. This will cause a flow reduction equal to:

$$q_w / q_s = \sqrt{\frac{30}{50}} = 0.78 \qquad (3)$$

where:
q_w / q_s is the ratio of winter to summer air flow rate through the remotest ATD, resulting in a 22 % reduction of the air flow through this device.

This flow change calculation is approximate, but clearly shows that the thermal driving forces can affect the flow balance to a significant extent.

The proportional method, applied to a branch duct with four ATDs, is illustrated in Figure 7. When balancing a group like this, it is always the most remote terminal from the main duct that is used as the *reference terminal*, R, and the other terminals in the group (CE 2, 3, and 4) are then compared to it. When balancing the flows, the reference terminal is assumed to have the lowest ratio, i.e. the lowest value of q_m/q_d, where q_m is the measured and q_d is the design air flow.

If another terminal in the group has a lower ratio, this terminal is called the index terminal, I. The first step is to set the reference terminal so that its ratio is the same as that of the index terminal. The other terminals are then adjusted, in turn, to the same ratio as the reference terminal.

After balancing, the index terminal must be opened completely to ensure that the lowest possible pressure level in the installation is attained.

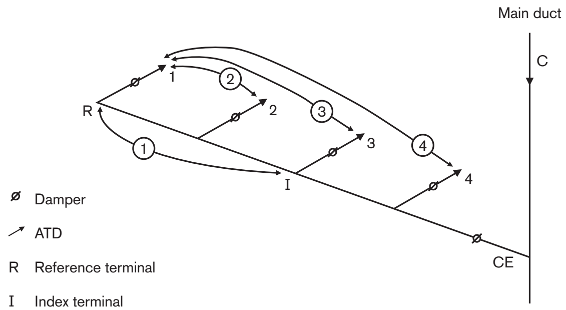


FIGURE 7. Balancing four ATDs on a branch duct.

This means that the commissioning engineer will be able to use relative measurements during the whole balancing process. The following relationship will apply:

$$\frac{q_m}{q_d} = \sqrt{\frac{p_m}{p_d}} \qquad (4)$$

where:
p_m is the measured reference pressure across the device
p_d is the reference pressure obtained at the design flow

This is important to note, as it means that the commissioning engineer only has to record pressures and does not have to calculate the air flows for each terminal. This saves a great deal of time and, consequently, reduces commissioning costs.

ATDs and connection boxes should, therefore, as far as possible, be fitted with measuring devices and be readily accessible to the engineer.

On completing the balancing work, the air flow is derived from the following equation:

$$q_m = k \times \sqrt{p_m} \qquad (5)$$

where:
k is the correction factor, specified by the ATD manufacturer, for determining the correct air flow.

As pointed out above, it is essential that the damper in the index terminal, in each sub-system, is fully open after balancing, as the pressure drop in the system has to be minimized. If the flow ratio for any of the index terminals is too low, the commissioning engineer must not increase the air flow to reach the design flow without considering the consequences. First of all, the cause of the low air flow must be investigated. Sometimes, it might be necessary to leave an index terminal as it is. The engineer must, however, motivate why this has been done in his report. If the pressure in the system is increased, so that the index terminal reaches the design flow, it could become too high and create noise problems.

The following relationships show how sound levels depend on flow and pressure:

$$\Delta L_A = 10 \log \frac{q_2}{q_1} + 20 \log \frac{\Delta p_2}{\Delta p_1} \qquad (6)$$

where:
ΔL_A is the change in sound level in dB(A)
q_1 is the flow before a flow change
q_2 is the flow after a flow change
Δp_1 is the pressure drop before a flow change
Δp_2 is the pressure drop after a flow change

The equation shows that a 10% increase in flow will cause a 2 dB(A) rise in sound level. An air flow increase of 15% will cause a 3 dB(A) rise.

STEPS BEFORE BALANCING

In order to carry out the balancing process correctly, a number of preparatory steps are required. The following steps should be carried out when the proportional balancing method is used and are applicable to both CAV and VAV systems.

Preparatory work – documents and data

- Make sure that all the necessary contract documents, revised drawings and specifications, as well as technical data for fans, units, devices etc, are readily available.
- Prepare schematics of every system and sub-system, if these are not already included in the contract documents provided by the design engineer.
- Plan the balancing work based on the above documents and facts. If necessary, revise any previously specified measuring methods, measuring points, schedules etc.
- Prepare the report forms for recording readings.
- Make sure that all the required instruments have valid calibration certificates.
- Inform the client and contractors on the building site when the balancing work will be carried out.

Preparatory work – checking the installation

- Check that the construction work has been completed to a satisfactory degree, i.e. that all necessary partition walls, doors and windows have been installed.
- Draw up a check list for each system. Note any special irregularities.
- Check that all the changes made during installation have been noted on the drawings.
- Check that the sizes of all supply air terminal devices are according to specification and that they have been correctly installed.
- Check that all fixed measuring devices have been correctly fitted in relation to the direction of flow.
- Check that the measuring devices have been fitted at the specified distances to any flow restrictions.
- Check that the installation is clean and operational.
- Check that all inspection panels/covers have been properly closed.

- Fully open all dampers and other balancing devices in the ducting.
- Angle slats in all supply air grilles horizontally/vertically.
- Check that fire dampers are open and fully functional.
- Set all fans to supply an air flow that exceeds the design rate by about 10%.
- Make sure that any automatic regulation devices in the system cannot affect the balancing of the air flows.
- Set the system to provide the maximum required air flow.
- Make sure that all supply and extract air systems that can affect the flow distribution are always in operation during the balancing process.

Preparatory measurements

- Measure the total air flow and flows through every terminal device in the system.
- Analyse the measured values. If they deviate significantly from the design values, this could be due to faults in the installation. If the deviations are due to faulty design, it is the design engineer's responsibility, in consultation with the commissioning engineer, to decide whether remedial work has to be carried out.
- Draw up a programme for the balancing process based on the preliminary measurements.
- Note the positions of all index and reference ducts/terminals on all schematics and schedules.

After completing all the preparatory steps, described above, the balancing work can begin.

PRACTICAL BALANCING WORK

The following list comprises the steps to be taken when balancing a complete installation. The different points refer to the system structure shown in Figure 8.

1. The balancing procedure for a CAV system using the proportional method:
   - Start in the branch duct or sub-system that has the highest ratio between measured and design air flow. (Branch duct C)
   - Find the sub-branch duct, from this branch duct, that has the highest ratio between measured and design air flow. (Sub-branch duct CE)
   - Use the proportional method to balance the terminals on this sub-branch duct.
   - Continue by balancing the terminals on the other sub-branch ducts connected to this branch duct. Balance the terminals on the sub-branch duct with the lowest ratio last. (Sub-branch duct CC)
   - Balance the sub-branch ducts, on which all the terminals are now balanced, leading from the branch duct. (Branch duct C). Use the proportional method and find the index and reference ducts.
   - Continue by balancing the terminals on the sub-branch ducts leading from the remaining branch ducts. In the example in Figure 8, there are two remaining branch ducts, of which one has no terminal devices shown.
   - Balance the sub-branch ducts leading from the all the branch ducts. Balance those on the branch duct with the lowest ratio last. (Branch duct A)

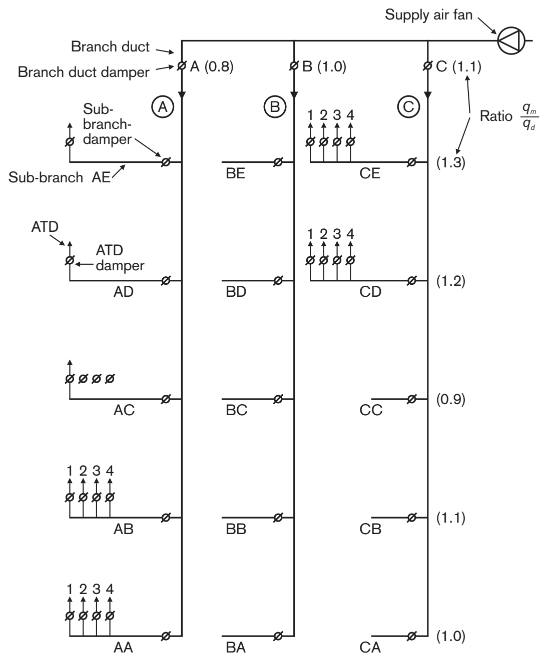


FIGURE 8. The system structure of the ventilation system used to illustrate the balancing procedure in a CAV system when using the proportional method.

- Balance the branch ducts. Use the proportional method and find the index and reference branch ducts.
- Complete the balancing work by adjusting the speed of the fan so that the ratio of measured to design air flow is 1.0. Measure the air flow, for example, at a convenient point close to the fan.

2. The balancing procedure for a VAV system using the proportional method. It is assumed that the flows are regulated using thermostats. The following steps must be taken:
   - Adjust the system so that the maximum air flow is obtained. This can be done by setting all the thermostats so that the control system delivers maximum cooling.
   - Check the air flow and static pressure across the remotest terminal and the terminal closest to the fan. If necessary, adjust the total air flow in the installation.
   - Balance all the supply air terminal devices connected to the VAV unit closest to the fan according to the proportional method. Check the total air flow through the VAV unit according to the proportional method and if necessary adjust it to the design value. Note that if the installation has been designed with a simultaneity factor less than 1.0, full air flows will not be obtained from all the supply air terminal devices.
   - Adjust the distribution patterns from the supply air terminal devices. Secure and record the position of the dampers and adjustment devices in the supply ATDs.
   - Change the thermostat from maximum cooling to maximum heating and check that the VAV device functions as intended.
   - Reset the thermostat and continue with the next VAV unit.
   - Use the same procedure as above for the other VAV units. This means that the balancing process must start at the VAV unit closest to the fan, where the static pressure is greatest, and continue towards the remotest VAV unit, where the static pressure is lowest.
   - When all VAV units have been balanced, the static pressure before the terminals that had the lowest pressures, the index terminals, is measured. Check that these are subject to the specified minimum pressure. All thermostats must have been set to maximum cooling.
   - Adjust the capacity of the fan so that the index terminals are subject to the necessary static pressure.
   - Together with the control systems contractor, adjust the static pressure regulator that controls the fan speed, normally located in the main ducting.
   - Measure the total air flow delivered by the fan and check that the fan motor is not overloaded.
   - When the extract air system has been balanced using the proportional balancing method, the extract and supply air systems can be balanced against each other.

Measuring methods

To obtain satisfactory balancing results, it is important that the measuring methods used are recognized methods and have few sources of systematic errors. Demands regarding systematic errors must be reasonably related to demands regarding flow tolerances. The systematic errors for the recommended measuring methods used in Sweden are given in [Johansson and Svensson, 1998]. These are generally followed by a recommendation regarding the choice of measuring instruments and the requirements specified by each method regarding acceptable levels of interference, etc. Practical balancing work requires, primarily, pressure measurements. However, when carrying out the final checks of the air flows, it is essential that standardized measuring methods be used.

The recommended measuring methods have been given special symbols, so that it is possible to identify which methods the design engineer has specified on the drawings.

The design engineer should focus on simplifying the balancing work. Primarily, this means choosing products that are fitted with built-in measuring devices.

If we are to rely on measured values, it is important that the instruments used have valid calibration certificates.

Probable measurement error

The probable measurement error, m_n, must be shown and calculated according to the following:

$$m_n=(m_1^2+m_2^2+m_3^2)^{1/2} \qquad (7)$$

where:

m_1, m_2 and m_3 are random errors as desribed below.

m_1 is the instrument error, expressed in %. Even when a measured value has been corrected with respect to the calibration curve, random errors will still occur. These are due to hysteresis effects for which corrections cannot be carried out.

m_2 is the error of method, expressed in %. This is due to deviations from the calibration method used for the measurement method in question. Only methods that are carefully specified should be used.

m_3 is the reading error, expressed in %. This depends on the scale divisions and type of instrument.

Experience shows that it is very difficult to achieve field measurements for which the probable measurement error is less than about 7%. To attain a maximum specified error of ±15% (including measurement errors), the measured deviation from the specified flow must not be greater than about 7%. This requirement can be time-demanding and difficult to achieve, unless the installation has been specially designed to facilitate the distribution of air in the system.

The requirements specified in Sweden have been adapted to levels deemed suitable for the majority of the installations where they could be applied. However, it is always possible for the client to change these requirements to ones that are better suited to the quality levels for the project in question. Consequently, it must be considered whether negative tolerances can be accepted in cases where air flows have been specified for reasons of hygiene.

Some recommended measuring methods

To minimize the time required for measurements, it is strongly recommended that the installation be fitted with fixed flow measuring devices at strategic points [CEN, 1999]. Flow measurements in ducting using Prandtl tubes or hot-wire anemometers should, in most cases, be avoided. This is primarily due to the time required to carry out the measurements but also because of the difficulties encountered, in awkward flow conditions, in finding suitable places to carry out the measurements. If the locations of any flow restrictions are at an adequate distance from the measuring point, i.e. more than about 8 or 10 duct diameters away, then no problems will arise. If measurements are to be made at the supply and extract ATDs, it is strongly recommended that they be fitted with permanent measurement devices. Flow measurements using anemometers with capture hoods should, in most cases, be avoided. The reason for this is that they are cumbersome and, therefore, difficult to use. Depending on the pressure drops across the terminal devices and the measuring devices, it might also be necessary to correct the readings to avoid large measurement errors. A simple reading of a typical pressure in a terminal is by far the simplest and quickest way to check the air flow through the device.

REPORTING ON BALANCING WORK

The contractor must fill in a report for the balancing work according to the stipulations in the contract documents and, immediately on completion of the work, submit copies to the design/project engineer and the client.

A balancing report should comprise the following details:

1. The name and location of the property and the client's name and address.
2. The name and address of the ventilation contractor.
3. The name and address of the commissioning engineer and the name of the person who carried out the actual balancing work.
4. Which ventilation system the report refers to.
5. Date and time when the balancing work was carried out.
6. The balancing method, the measurement methods and the instruments used.
7. The calibration certificates for the instruments.
8. The design values and allowable deviations for the total flow and the flows through all the terminals.
9. When using the presetting method for balancing, all calculated values and design settings for dampers and terminals must be stated.
10. Measurements of:
    a. Total air flows.
    b. Flows in all ducts where measurements have been made.
    c. Flows through all terminals.
    d. The calculated probable measurement errors.

The deviations in relation to the design values must be given for all measured flows. Measured values greater than the allowable deviations must be pointed out or noted in a special column and, if possible, the reason for each deviation stated.

For every measurement, the following must be given:

1. Date and time.
2. Outdoor temperature and barometer readings.

3. Wind conditions.
4. Notes on whether doors or windows were open or shut.
5. Other conditions that could be of importance for the balancing work.

SUMMARY

It can be seen from the procedures described above, regarding preparations for the balancing work and the different steps to be taken during the actual balancing process, that the commissioning engineer must be fully competent and well-acquainted the different steps. A commissioning engineer must have a sound knowledge of measuring and balancing techniques, as well as hydrodynamics and acoustics. The commissioning engineer must also be able to:

- Check whether the design engineer has taken the necessary steps to make sure that the balancing work can be carried out correctly.
- Carry out air flow calculations, based on the measured pressures, and measurement error calculations.
- Check the effect of driving forces on the measurements, etc.

This means that there are a number of requirements to be fulfilled, with respect to the commissioning engineer's competence. Unfortunately, these requirements are not always stipulated when recruiting new engineers.

One might suspect that steps can sometimes be overlooked, as the commissioning engineer often works against the clock. Balancing is the last phase of a ventilation installation contract and the day allotted for the final inspection work is often decided on before the engineer has commenced his work. Bearing this in mind, it is not surprising that there are often imbalances in ventilation installations.

Ventilation systems play an important role for our health and well-being. It is, therefore, the responsibility of the ventilation industry to improve system designs, so that better guarantees can be given regarding the stability of design air flows over long periods of time. A natural step would be to provide installations with automatic devices to maintain the correct distribution of air flows.

One example of a suitable solution is mentioned above, in which the control system maintains constant pressures in the branch ducting. This not only means that a great deal of the balancing work can be avoided, saving costs, it also makes it easier to a create a functional installation in which the pressure rise is kept to a minimum and energy use is thereby reduced.

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30. EFFICIENT CONTROL OF AIR HANDLING EQUIPMENT

NILS SPETZ Product Manager, Swegon AB

The subject of control technology has been addressed and discussed by numerous experts and there are a number of excellent textbooks that describe the basic principles and functions involved. A great deal has also been written about recent technological advances and the automation of building services. These books and other publications have often been written by authors with in-depth knowledge about control system design and the electronics employed. Further discussion of control technology from these points of view is therefore not necessary.

Instead, this chapter focuses on how joint development of control equipment and mechanical plant can be employed to make better use and improve the efficiency of the mechanical properties of air handling equipment. If joint development is to be really successful, then it is also necessary to be aware of, to understand and to be able to control parameters such as properties of individual components and the state of the air (humidity, density, etc), and even its thermal properties. This knowledge can then be used to maintain the specified tolerances for the system. It must be remembered that sensors and measuring equipment always work within given tolerances and if these devices are not tested together, then it is not reasonable to expect that specific tolerances can be met.

Individual adaptation of some of the parameters and components to the complete installation and the air handling unit itself is also important. For example, if the air heater is to be used in conjunction with waterborne heating, the flow control valve must have a suitable operating range, i.e. the valve must be designed so that it will operate fully open at the design water flow. Even the dynamic properties of the system, such as the running time of the valve actuator, the length of the piping to the air heater, the response of the hot water/air heat exchanger and the length of the ducting system will affect the settings of the regulator's control parameters. If the temperature of the room air or the extract air is regulated, the size of the room will also be a factor that affects the settings of the regulator.

The dynamic properties of the system are described using the terms 'deadtime' and 'time constant'. Deadtime is the time it takes from a change being made until a sensor registers it and the time constant is a measure of the responsiveness of the system. Examples of factors that affect the response of a system are room size, the capacity of the heat source and the responsiveness of the sensor, though this is of little importance where temperature controlled ventilation is concerned. System response can also be affected if a direct expansion air cooler is installed. In this case, the dynamic properties of the cooling equipment must be taken into account as well. Output is controlled by letting the cooling compressor start up in steps. Here, it is necessary to adjust the step time to compensate for the deadtime between the starting of the compressor and the coolant reaching the air cooler. This means that even when air treatment units with jointly developed control equipment are used, it must still be possible to adjust them individually, if they are to function properly in an installation as a whole.

In general, the integration of control systems with the plant and equipment (boilers, air handling units, coolers, etc) normally found in buildings has been very slow compared to that seen in many other fields. This is especially true for air handling units, most of which are still sold without any form of jointly-developed or jointly-tested control equipment. A simple analysis of products from other industrial sectors is a relevant starting point.

GOOD EXAMPLES FROM OTHER SECTORS

There are many different industrial sectors in which successful product development has taken place alongside a significant reduction in energy demands. And, at the same time, the products have become more user-friendly, offer better performance and safety, and have been given other new and valuable properties. This would not have been possible, if the products and their control systems had been developed independently of each other.

A few examples of consumer-oriented products are given below. Comparisons with these are especially relevant, as the functions and properties of the products are well known by the general public.

Washing machines

In older types of washing machines it is only possible to set the temperature of the water and choose between full and half loads, and whether the pre-wash and spin drying functions are to be used. The latest machines, on the other hand, have a number of new functions that reduce energy needs and give better washing results. A special sensor now measures how quickly the textiles absorb water. The machine then automatically regulates the amount of water, to provide the correct drop height in the drum. You do not have to think about putting in a whole or a half load – the machine measures the load itself. The total amount of water required is reduced, which means less water has to be warmed up, and the washing sequence takes less time. The speed of the spin dryer has also been increased, which means that water can be emptied even faster, again reducing time and energy use. As the washing then contains less moisture, less energy will be needed for drying. The spin-drying speed has been increased by fitting a balance control, which makes sure that the washing is evenly distributed in the drum before the program starts. If there is still an imbalance, the machine will automatically reduce its speed. To provide the most efficient rinsing possible, by using as little water as possible, some washing machines have built-in rinse sensors to measure the purity of the waste water. This is an example of a machine in which the mechanical functions and control equipment have been tested and developed together, providing a more energy-efficient product and increased customer satisfaction.

Cars

There is most probably no other product group in which the advances made thanks to the integration of electronics are so obvious. Today, microprocessors control just about every function in a car and numerous different types of sensors are used to supply information so that all systems can work optimally.

Fuel consumption has been significantly reduced with the help of electronic injection, which optimizes the mixtures of fuel and air. The performances of engines and their power ratings have also been increased with the help of electronic controls.

Another good example is the brakes. If you look at the brake pedal itself, no great changes have been made for a long time. On the other hand, the difference between putting your foot on the brake pedal in a new car and in an old one is enormous. A previous braking distance of 50 metres at a given speed has been reduced to 30 to 35 metres – valuable metres that have helped reduce accident rates. And there is now no need to regulate the braking force applied by your foot, as antilock braking systems provide an optimal braking force, even on wet roads. Many cars are now fitted with anti-skid systems, also controlled by advanced electronics.

Thanks to successful product development, cars now offer us increased performance, lower fuel consumption, increased safety and improved roadworthiness. This would not have been possible, if the electronic and mechanical systems had not been developed side by side.

There are numerous other examples of products from other sectors, which, with the help of integrated control equipment, have become considerably more energy-efficient and more user-friendly. HVAC systems can be seen in the same light and it is quite obvious that system components should be fitted with purpose-designed and jointly-tested control equipment, if present and future demands for operating efficiency and energy efficiency are to be met.

In the following sections the functionality of important parameters, determined by the properties of the equipment and the air, is discussed. Control equipment can then be developed based on functionality and not the other way round, as is often the case. For example, control systems are often supplied with pre-programmed functions that can be adapted to a wide range of air handling systems. These systems are then installed together and compromises struck to create an acceptable solution. Functionality tests and the joint testing of sensors and measuring equipment are, unfortunately, still performed in the field using components that have seldom or never been tested together before.

FOCUSING ON FUNCTIONALITY AND ENERGY EFFICIENCY

In new directives and standards, the focus is on energy use. A performance indicator that is often used is the so-called SFP_v rating of a fan, which, among other things, gives an indication of its energy efficiency when installed in an air treatment unit. This is a reasonable criterion, but will a fan that has the best SFP_v rating also require the least energy in the finished installation?

What happens if it turns out, when a control equipment contract has been negotiated, that the speed of a fan motor is to be controlled using the cheapest frequency converter? How efficient will the frequency converter be and will it be possible to adjust it correctly and adapt it to the fan motor on site?

Today, many fans are fitted with some form of air flow measuring device and it can, for instance, be installed in the fan's air intake. Measuring errors could be in the region of ±5%. Even pressure sensors have varying measuring errors, usually around ±5%. If the pressure sensor installation is carried out on site, the total measuring errors could be around ±10% for each air direction and this, at worst, could mean a difference between supply air and extract air pressures of up to 20%. This will, of course, greatly affect energy use and is hardly acceptable. The fans will supply more air than they were designed for and imbalances in flows between supply and extract air might give rise to undesirable pressure differences in the building. More energy will be required, comfort problems could arise and moisture might affect the walls and roof of the building. There are many good reasons why control and mechanical systems should be integrated with each other and some fundamental arguments are discussed below.

Fans

A fan is a centrifugal machine with a load curve that is proportional to the square of its speed. When a fan is fitted with speed controls it is important that the control unit, the motor and the impeller have been jointly developed and tested to optimize total performance. Although a frequency converter has a number of different parameter settings that can affect its operation, the ratio of supplied voltage to frequency is the parameter that has the greatest effect on the power requirements. By studying the fan curve and adjusting the operating parameters of the frequency converter, it is possible to reduce the power required and to ensure that the fan delivers the required torque.

Modern fans often have direct drives to avoid power losses, otherwise associated with belt driven fans. The fans are usually fitted with some sort of speed control device and this is often in the form of a frequency converter. However, the motor might then become sensitive to the leakage currents that arise when frequency converters are used and these could damage the motor bearings. If fans, motors and frequency converters were tested together, it would be possible to eliminate leakage currents by changing the switching patterns in the converter. A fan that has been developed in this way requires less energy and offers longer operational life.

Air flows

If energy-efficient operation is to be guaranteed, it must be possible to control the air flows. As mentioned earlier, imbalances in air flows can lead to an increased use of energy and comfort problems. It is therefore important to have jointly-tested measuring devices that take into account measuring errors and that can verify the measuring accuracy.

Flow measuring devices are often installed in fan intakes. If the air treatment unit is fitted with a rotary heat exchanger, the venting air must also be taken into account when determining the correct extract air flow.

Even the density of the air can affect the air flow rate. The density of air at −20 °C is 1.39 kg/m^3 and at +20 °C is 1.20 kg/m^3. When a ventilation system is commissioned the temperature of the air in the ducting is normally between 20 and 25 °C. If commissioning is carried out in warm weather, and the air treatment unit is fitted with a highly efficient heat recovery unit, and with a flow measuring device situated in the fan's air intake, the extract air flow in winter could be 10 to 15% higher than the design value. This will cause an excessively large under-pressure in winter and cold unheated outdoor air will infiltrate into the building. The reason for this difference in air flow is that the exhaust air temperature in the heat exchanger varies depending on the season. In summer, the temperatures of the extract air and the exhaust air are roughly the same but in winter, when the heat exchanger is used to recover heat, the temperature of the exhaust air could be −10 °C or lower. This means that the densities of the extract air and exhaust air will be different, causing a flow rate discrepancy. If, on the other hand, a system is commissioned in cold weather, there could be an over-pressure in the building in the summer. By taking into account the temperature of the air when the control system measures the air flow, it is possible to compensate for changes in flow due to the varying density of the air. A well-tested flow measuring method with carefully thought-out functionality will ensure that the design air flow will always be attained, no matter what the time of year or the outdoor temperature.

Heat exchangers

Present and future energy-saving demands mean that air handling units will always have to be fitted with some sort of heat recovery system. Rotary heat exchangers offer the highest temperature efficiencies (80 to 85%) and can also provide full heat recovery in winter, as there is no danger of them freezing up if the extract air is not too humid. It is even possible to use this type of heat recovery unit in summer to help cool the supply air.

The maximum speed of a rotary heat exchanger is about 6 to 10 rpm. The relationship between speed and temperature efficiency is not linear and even at very low speeds the unit can recover heat. Its temperature efficiency at 1 to 2 rpm is already quite high. In the spring and autumn, when outdoor temperatures are close to the design supply air temperature, it is important to be able to control the heat exchanger so that the required torque can be maintained at low speeds, thus avoiding variations in the supply air temperature.

To ensure that the leakage direction is as intended and that the venting sector in the heat exchanger is functioning properly, the extract air section must be subject to a small under-presure so that extract air cannot leak into the supply air. In addition to these pressure conditions, the size of the venting sector is the parameter that will determine the size of the air flow. In air handling units that handle large air flows, it is sometimes possible to adjust and set the venting sector manually. This is not normally the case in air handling units that handle small amounts of air.

In demand-controlled ventilation systems, which are becoming increasingly more common, the pressure difference in a rotary heat exchanger between the supply air side and the extract air side is reduced when the air flow is reduced. A consequence of this is that there will be too little time to vent the venting sector when flows are small and full recovery is required. This can be compensated for by making the venting sector larger. The disadvantage of this solution is that the temperature efficiency of the heat recovery unit is reduced. As an alternative, the pressure balance can be adjusted to satisfy the lowest air flow rates and this will ensure correct venting at all higher rates. The disadvantage here is an increased pressure drop in the whole installation. A third alternative is to let the speed of the heat exchanger vary as a function of the air speed through the unit. In this way it will be possible to ensure that the venting function works properly, as the speed of the rotor is reduced when the air flow is reduced. This low air flow will result in a slightly increased temperature efficiency, which, in turn, compensates for the negative effect caused by the lower rotor speed. This third alternative is viable when all, or nearly all, parameters are known, and would be quite achievable, if the mechanical and control systems were developed side by side.

Filters

Filters are the parts of an air treatment system that require the most maintenance. They must be changed on a regular basis, so that excessive pressure drops and subsequent increases in energy use are avoided. Changing filters is costly and it is important that they are not changed more often than necessary. In some systems there is no filter monitoring unit at all and the filters are just changed at regular intervals. This is a viable solution but will most probably entail unnecessarily high costs, both for replacement filters and energy.

The installation of a pressure sensor, to check that the differential pressure does not exceed a certain set level, is a good solution in systems where there is only one fixed operational flow rate. This type of solution, however, is not usually satisfactory in a system with variable flows. In this case, a sensor should be chosen that allows the monitoring system to continually measure the differential pressure across the filter. The alarm level for the sensor is set when the air handling plant is started up with a new filter in place or when it has been changed. When the plant reaches the design air flow, the pressure drop is recorded and stored in the memory of the control equipment. This value is then used to calculate the different alarm levels corresponding to the pressure drops for the whole of the flow range. In this way the filter can be monitored to provide optimal results with respect to energy and filter costs.

Reheating and cooling

Air heaters and air coolers in waterborne systems require valves, pumps and actuators to control the flows and temperature of the water. Valves and pumps must be designed very carefully to ensure their correct authority in the water circuits.

When electrical air heaters are used in systems with variable air flows there could be a risk of the heating rods overheating, if the air speed is too low when the flows are low. If technical solutions and control systems were developed along side each other, problems like these would be avoided. As the area of the air heater is known, the speed of the air at a given air flow rate is also known. If air speeds were too low, the power supplied to the heating rods could be reduced and their operational life extended significantly.

Other functions in control systems

Control systems for air handling units also comprise a number of functions that are not directly connected to specific mechanical features. Nonetheless it is important to include them when the control equipment forms an integrated part of a system in order to provide sufficient functionality to deal with other needs that might arise. These extra functions could include time controls based on weekly or seasonal settings, different types of temperature control and air flow control, and numerous other functions.

Verification
One of the greatest advantages of integrating controls into a piece of equipment is that it makes it possible to guarantee quality and define tolerances beforehand – not only for specific parts of the measuring process but for the system as a whole. Equipment functionality can be tested in the field and the experience gained can then be implemented in future product versions.

The fulfilment of demands for electro-magnetic compatibility, EMC*, is easier to achieve when the control equipment is integrated into the equipment. The whole unit can then be sent to a laboratory for testing and approval, thereby guaranteeing that interference levels are not exceeded. It is not possible to do this if the control equipment is added on at a later stage. If this is the case, it will have to suffice to ensure that all the individual components meet the necessary EMC demands in the hope that they, together, will not affect or be affected by any surrounding equipment.

AIR HANDLING – ONE OF THE MANY SYSTEMS IN A BUILDING

The air handling system is one of the many different systems found in a building and these systems are very often connected together under a main control system. This means that the control equipment must have a communication interface to the main system and, as there are a number of different standards for interfaces and communication protocols, more than one type of interface and protocol will be needed. This is especially important in control equipment integrated into an indoor climate system, so that compatibility with a wide range of main systems can be ensured.

Air handling units with integrated control equipment are still regarded as novel ideas, despite having been available for a number of years. Strong traditions and commercial arguments are the main reasons why control equipment and air handling units are still purchased as separate units.

* EMC stands for electromagnetic compatibility and is used in connection with unwanted interference effects that a piece of equipment can create and to what extent the equipment itself is shielded from surrounding interference.

Supplying air handling units with an integrated control system places much greater responsibility on manufacturers. They must make it quite clear what is included and what has to be assembled on delivery. Some components, such as room sensors, exteriorly mounted ducting dampers and other parts cannot, of course, be integrated into the air handling units but have to be installed and connected afterwards. Manufacturers must therefore take great care when stipulating what is included in a delivery. Greater cooperation will also be required between the design engineers involved in planning the ventilation system and those involved in planning the power supplies to the building and to the control and monitoring systems.

Consultants and companies involved in building services automation and monitoring systems must be willing to develop a greater understanding of each other's fields, if air handling units with integrated control equipment are to function satisfactorily in combination with other automated systems in a building.

SUMMARY

Future demands for less energy use will mean that the development of air handling solutions will never be allowed to slow down. And the only way in which it will be possible to meet these demands is by initiating serious joint efforts aimed at the development and integration of control systems and air handling equipment. This is a challenge that will drive development forwards, towards even more energy-efficient indoor climate solutions.

31. THE INTELLIGENT BUILDING – A MATTER OF CHOICE

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There has long been a desire to create intelligent buildings; buildings that are capable of making necessary and intelligent decisions that enable optimal indoor climate and optimal energy utilization, etc. Today, this is more or less a reality; however, it is based on the assumption that various systems in a building, e.g. for lighting, ventilation, heating, cooling, access, fire safety, carbon dioxide monitoring, etc., can be combined in a single network. How can this be possible when the various systems come from different suppliers? As a customer, can I be certain that I am investing in a long-term solution with a reasonable lifecycle cost without leaving myself at the mercy of the supplier? Here, I wish to describe what it means to create good conditions for an intelligent building with a focus on freedom of choice.

WHY INTELLIGENT BUILDINGS?

Naturally, not all buildings need to be intelligent, and the fundamental challenge is to find the right level of intelligence for each building. We have to consider the costs of adding intelligence and the functions we wish to enable. In most cases, the intelligent building allows the property owner, the property manager and the users to improve their situation and even lower their costs. Consider the following example. Each of us has at some time or other probably had something to say about our indoor climate; that it is too hot or too cold, the air is poor, it is too dark or that it receives too much sunlight. It is easy to overlook the fact that indoor climate is important for our ability to perform. Suppose a company were able to achieve optimal indoor climate. In practice, this would mean that the indoor climate was continuously adapted according to the type of activity, how the premises are used and to suit the needs of the people who use the building. In addition, we assume that each person's individual desires are satisfied to the extent that this is possible. This would mean that the employees feel physically and mentally livelier and more alert and, consequently, that each individual can increase his or her productivity. If we assume a productivity increase of 5 to 10% and that the company has 400 employees, this would have a significant impact on the company's financial outcome. Furthermore, this may also affect a number of dynamic values which in turn result in greater profitability. Examples of the latter include improvements in the quality of the company's products and services, a greater degree of service to customers, and fewer absences due to illness, etc – something that is perhaps worth a little extra consideration.

Another benefit of intelligent buildings is that they allow us to influence energy use. We all wish to reduce energy use, not least for the sake of the global environment. Energy use is largely determined by how we use the building; e.g. when we feel the room is too warm we open a window to admit cooler air, while at the same time the radiators continue to pump out expensive heat. Or, when we neglect to switch off the lights when we leave the room. Or, when ventilation and cooling systems continue to run at full effect, despite the fact that everyone in the company has left for the day. Normally, a building's total energy use is not utilized to the full. As much as 10 to 20% of the energy may be lost as a result of sub-optimal interaction between the systems in the building.

An intelligent building in which different systems work together therefore offers several benefits, not least if the systems are coordinated with the aid of a supervisory monitoring system. It is not always easy to create an intelligent building. First and foremost, the systems must be able to communicate with each other.

COMMUNICATION BUS SYSTEMS AND PROTOCOLS

In an intelligent building, the systems for ventilation, heating, cooling, lighting, access, etc. must therefore be able to communicate with the supervisory monitoring system. For this to be possible, everything must be integrated in a common network. All systems linked in the network can send information to each other, depending on how the network is configured when it is taken into operation. This type of network is usually referred to as a *communicating bus system*. Quite simply, the equipment is connected via a so-called computer bus. This set-up resembles an office in which all workstations, printers and servers are linked in a common network and where data is continually sent between units in the network. In the context of automation, the Ethernet network corresponds to the bus.

Consider this example from a building. A motion detector senses motion in a room and sends information to the access system. The same motion detector can also be used to send information to other systems to inform them that no one is in the room. The lighting system can then extinguish or dim the lights, and the heating and ventilation systems can adjust their output to the right level. In this simple example, the same information about motion is thus sent to different systems that control access, lighting, ventilation and heating.

Communication between different units is decided by a so-called protocol. In simple terms, a protocol is a set of rules governing data transfer between the various units in the network, how the data stream is built up, and for ensuring that the data reaches the desired destination, etc. A protocol can also contain much more, such as procedures for security, encryption, etc., but we will not go into detail here. Since each protocol describes in detail how information is to be transferred, there is no easy way of connecting products with different protocols in the same network. This often requires custom solutions and should be avoided.

CLOSED SYSTEMS

Formerly, manufacturers of equipment for buildings developed their own protocols, thereby creating their own standards for how their products would communicate with other devices in the building environment. This actually implied a limitation in the environment, since only the manufacturer's own products, with the same type of protocol, could communicate with each other in the same network. Therefore, a larger building with equipment from several different suppliers might contain several smaller closed systems, each with its own protocol and each living a life of its own. To control and monitor the systems, there was often a supervisory system in the form of a graphic user interface on a PC. Similarly, each supplier delivered its own supervisory system, which matched its own equipment. These companies supplied what is now known as closed systems. Very often, each closed system managed its specific task perfectly adequately. The problem was that it was virtually impossible to integrate the different systems into a single network. One had to live with the fact that the building was equipped with several closed systems installed in parallel.

As a building user, one may have had a relatively intelligent building,

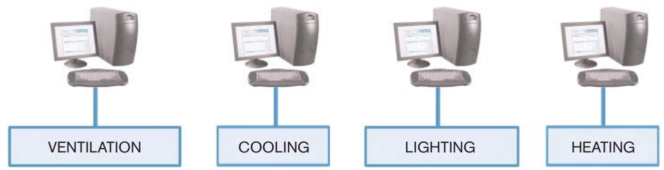

FIGURE 1. Closed systems.

but not without problems and often rather high costs. The suppliers sold inexpensive systems, but subsequently commanded higher prices when it came time for spare parts, new components and service. Furthermore, integrating several different suppliers' systems into a single network was complicated. A total integration with a supervisory monitoring system normally requires a PC equipped with specially adapted software that brings everything together in a single network. Alternatively, so-called *gateways*, which can bridge two different protocol standards, were used. This was something that proved very costly to develop, but even costlier to maintain in the future. At the same time, a common drawback was that these customized solutions could only handle a limited amount of information, which meant that much of the desired intelligence was lost.

When it came time for expansion or alteration, there was often no recourse but to contact the original supplier and order more components. This could prove expensive, considering the fact that there were no alternatives – that is to say, if the original supplier was even still in business. This was rather advantageous for the supplier, who was free to set his prices accordingly. At the same time, each system from the respective suppliers required unique expertise for installation and commissioning. This also applied to daily operation and maintenance. Again, only a limited number of companies were up to the task. In other words, the building owner was at the mercy of the supplier and in no position to invite tenders from competitors. One simply had to accept the situation or replace the entire system and start anew. The possibilities for providing for any future need to change the building were thereby restricted.

OPEN SYSTEMS

Over time, the end user has gained a growing awareness of the negative aspects of dependency on suppliers. This resulted in new demands from the market, which eventually led to the development of what we now call *open systems,* which are obviously the opposite of closed systems. Just what constitutes an open system may vary depending on who you ask. In simple terms, an open system allows the freedom to choose products from different suppliers, e.g. systems for ventilation, lighting, access, etc. Despite the diversity implicit in freedom of choice, it must also be possible to integrate the products in a manageable and cost-efficient way in a common network.

Open systems make use of established, available standards in all phases, in everything from product design to installation and integration with other products, to enable the exchange of information in a common network. For a system to be open, the standard must be fully available to all who will use the system. An open system is based on an open protocol, which is applied by manufacturers when they develop their products.

This is also seen in the computer world, where many manufacturers develop their products according to open standards and protocols such as TCP/IP. This is why it is possible to choose freely from a range of computers, servers, hubs, printers and other peripherals from different suppliers, and then integrate them into a single network. This has contributed to the rapid development and widespread use of office networks that are now able to use standard components from different suppliers at a reasonable cost.

If we turn again to our open systems in buildings, we see that the user is therefore free to choose products that comply with the standard, regardless of supplier. Compatibility also means that, should the need for replacement arise in the future, the individual products can be exchanged. Compare this to the closed system, where normally only the original supplier could solve the problem!

The market now offers a few open systems that comply with different standards. Just because a system is open does not necessarily mean that two products with different protocols can communicate with each other, even though they both use open protocols. For this to work, they must be of the same type. It is also important to understand that when a system is procured with the requirement that it must be open, the installation, configuration, integration and documentation must be provided in the correct manner; otherwise, many of the benefits of an open system are lost and we end up back at square one. It may seem complicated, but a simple rule of thumb is that if a building is based on open systems, and thereby on accepted standards, full integration at a reasonable cost is possible.

Obviously, there are numerous advantages with an open system. Several of these advantages are described here.

Lower installation cost

Open systems allow many sub-systems to interact in one and the same network. This implies a lower material cost, since fewer networks are used to integrate the building's various systems. In turn, less cabling is required, which represents a further reduction in the installation cost. Compare this to the way installation used to be, when every supplier had his own closed network! Each system then requires a separate installation including the necessary cabling, etc.

Lower lifecycle cost

Open systems present new possibilities with respect to operation and maintenance of the building. An integrated building with open systems has fewer specific networks. This means that one or a few service companies can assume responsibility for the entire operation, which is more cost-efficient than the old way, when it was usual for each supplier to service his own system. Instead, one service call now suffices to maintain several systems in the buildings.

Energy savings

Sub-systems can provide information that can help to improve the operating efficiency of the building, and thereby reduce energy use. Here, a supervisory control and automation system fulfils the important function of analyzing all the information, which we will discuss in a subsequent section.

Flexibility and expandability

A lot can happen to a building over the course of its life. Perhaps it is extended or renovated. Maybe the function of the building changes when new tenants move in. This often means that the building-automation system must be upgraded. With closed systems, alterations were often costly – if it was at all possible to retain intelligence in the building. Even a minor change could entail enormous expense, especially if the building had custom-made systems that had to be revamped.

But with the freedom of choice offered by open systems, changes can often be made at a lower cost than previously, while the intelligent functions of the building are retained. The lifecycle cost of the building is kept at a reasonable level, even if the building or its use changes.

Freedom of choice

Perhaps the most important effect of open systems, and a feature that is more or less implicit in all of the advantages mentioned here, is the freedom of choice they afford. With standardized open systems, a greater range of compatible products is available from the various suppliers on the market. This allows greater freedom of choice during the procurement of an intelligent building. The knowledge that products are interchangeable often induces suppliers to take better care of their customers. The same applies to companies that provide services for installation, systems integration, maintenance and operation. When many suppliers are able to offer components and services, the increasing competition means lower prices. At the same time, quality may improve, since each supplier must make a greater effort to maintain his market position. The customer is able to reject the less competent suppliers in favour of those who offer demonstrably good quality.

In other words, when the need for building alterations or service arises, it's all about freedom of choice and not being bound to a potentially expensive long-term agreement. With closed systems, it can be very difficult to change suppliers once the system has been installed. In addition, products and systems in the original supplier's range may no longer be available, the supplier may have gone bankrupt or perhaps drastically increased his prices.

Open systems make a building intelligent, creating a basis for improved indoor climate and helping to reduce energy use. The installation cost is lower and the lifecycle cost is reduced. The end user is free to choose suppliers of both components and services for installation, maintenance and operation. Open-systems users can feel secure in the knowledge that they have invested wisely when they are not dependent on merely a few suppliers. Greater flexibility means that the building can be developed to keep pace with changing needs. You have doubtless noted that the key term in this section has been 'freedom of choice'. Freedom of choice brings long-term security.

BUILDING MANAGEMENT SYSTEMS (BMS)

A large intelligent building often has a so-called BMS (Building Management System), which is normally a PC with advanced software. These are also referred to as presentation systems, control and monitoring systems, maintenance systems, etc. All systems in the building, e.g. ventilation, lighting, etc, are connected to a BMS via the common control network that enables the systems to communicate. The purpose of a BMS is to allow the user, via a computer monitor, to oversee the operation of equipment and systems and thereby ensure that all the elements are working together correctly and efficiently. It must also inform the user about the status of the building by presenting important information on operating parameters, alarms, etc, and notify the user as to maintenance requirements.

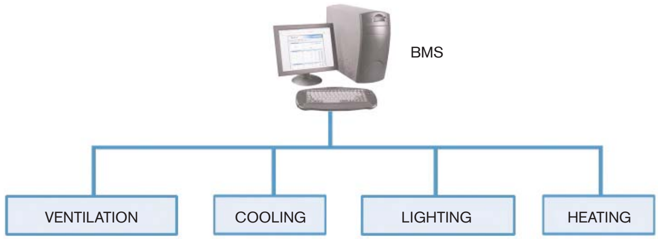

FIGURE 2. Open systems – Building Management Systems.

A BMS can collect information from the systems over a long period, thus allowing the user to learn about the building's "behaviour" in different situations. An intelligent building that is based on open systems generates a lot of information and operating data that can be managed with a BMS. An increased awareness of the building's behaviour makes it easier for the user to discover changes or abnormal patterns, which facilitates identification and preventive action to eliminate problem sources.

GREATER UNDERSTANDING OF THE BUILDING SAVES ENERGY

The greater awareness of the behaviour of the building gained via a BMS vastly improves the possibilities for saving energy and thereby reducing operating costs. An invoice from the power company really only shows how much electricity has been purchased during a given period and at what cost. With a BMS, it is possible to determine where and when the energy was used. In this way, we can pinpoint undesirable energy use, e.g., outside office hours, or via a specific piece of equipment that is used incorrectly or excessively. An example might be a ventilation system that is running at full capacity after office hours, even though the premises have been vacated. Defective sensors may lead to excessive heating, and consequently an increased demand for cooling. In other words, the BMS can be a tool for finding hidden costs.

I can give a concrete example of how greater awareness of what is happening in the building can result in better economy. During the 1990s, a study was conducted in Borlänge, Sweden, to investigate how awareness of energy costs affects the use of electrical appliances. In each rented apartment in a group of buildings, a small wall-mounted display showing current energy use was installed. The interesting feature was that the figures displayed were in Swedish kronor, instead of kWh. The people participating in the study suddenly became aware of the relationship between their use of electrical appliances and their electricity bill. By turning on the range, they were able to note instantaneously how much their energy use increased in monetary terms. Many people soon realized that, for example, the range, laundry machine and dishwasher use rather a lot of electricity and subsequently that one should give some consideration to how these appliances are used. The results of the study showed an average reduction in energy use of 15%. This can only be achieved if users are aware of how they are using the equipment.

There are several advantages of linking the systems together with a BMS via the control network. More information about how the systems in the building are working means they can be used optimally and correctly. When new equipment is installed or when alterations are made, set-up and fine-tuning of the systems is facilitated. Once the systems are operating, the operator can check to ensure that they are working at just the right level to perform their tasks. At the same time, a BMS can indicate when a specific system is in need of service, which means that the working life of the equipment can be prolonged. This also enables better planning of maintenance and remedial action, as well as future investments.

Note, however, that if an installation has been poorly planned and executed, it makes no difference how sophisticated the open control system is; it cannot solve the fundamental problem. On the other hand, an open BMS provides information about problems or history, making it easier to identify and remedy problems.

WHY ARE THERE SO MANY NETWORK STANDARDS?

We will conclude with a classic problem. Today, many closed networks have been phased out and replaced with open networks together with all the attendant benefits. But why are there so many different open network standards? Why has there been no consensus on a single network that can be used for all systems in a building? If this were the case, integration would not pose a problem. Several of the many reasons are discussed here.

First and foremost, there are several major market players who have long played a decisive role in their respective geographic areas. They may have previously worked exclusively with closed networks and now offer a combination, or perhaps only open networks. Because these players are so dominant, they also have a big impact on the market since, for example, they have large resources for marketing one or more of the networks they recommend. Consequently, a geographic division of the market arises whereby, over a long period, a single large supplier sells and installs a certain type of network. As a result, new players adopt products and services that are associated with this network.

Technological advancements are made continually, affording new possibilities for improvements in communication solutions for building automation. Internet and wireless technologies, for example, have spurred the development of new standards.

Several suppliers of open systems also have a strong profit motive, since they manufacture and own components that are necessary for designing a product for a certain type of network. The supplier may also be responsible for performing the requisite certification, which normally entails an expense. Consequently, since he profits when his network technology is spread and used, the supplier markets his product actively. Note that, even though a supplier has an ownership interest in a network, the network can still be open and accessible.

In addition to political reasons, there are also a number of practical reasons. Different types of systems in a building must comply with different types of requirements, which means they have different features and characteristics. Safety and security-critical systems, e.g., fire protection and access control systems, must satisfy stringent requirements for delivering a message to the correct recipient within a given time. At the same time, there must also be some form of encryption to protect sensitive information. Normally, devices such as a heat pump or outdoor thermometer are not subject to such strict requirements. This may be one reason why manufacturers of different types of systems base their solutions on different standards.

In addition, there may be requirements with respect to the speed at which information is exchanged. Communication between a circuit breaker and a light fixture must be fast enough. If it takes too long for the lamp to light, we instinctively hit the switch again, which turns the lamp off. Temperature measurement, which often has a much slower progression, is not subject to the same type of speed requirement.

All in all, this probably means that there will always be several open networks on the market. The gradual demise of the closed network is a big step in the right direction, and different manufacturers now offer products and systems that are compatible with each other.

As a user, one should always demand open networks and choose suppliers that offer the possibility to communicate according to a palette of different standards. This ensures freedom of choice in the event that equipment must be replaced or complemented. At the same time, it is essential that the equipment is installed, configured, integrated and documented in accordance with the open-systems approach. With the freedom to choose from among a range of open alternatives, we can create a future-safe building that can be managed and developed over time at a reasonable cost. When creating an intelligent building, there are also good reasons not to become too focused on trying to minimize the initial material costs. An intelligent building is so much more, which is why it is always better to focus on the building's lifecycle cost and the functions one wishes to enable.

32. QUALITY ASSURANCE FOR A GOOD INDOOR ENVIRONMENT

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FIGURE 1. The provision of a good indoor environment in a building, one that does not cause ill health or discomfort, is a basic demand made not only by the residents but also by society in general.

QUALITY ASSURANCE – AN ONGOING ASSIGNMENT

Quality assurance of buildings and their functions is not a clear-cut concept. Some people think that quality assurance just means extra paperwork and the creation of pages of documentation to be filed away and not used for any practical purpose. Hopefully, most people regard quality assurance as a way of ensuring that the goals that have been set are actually reached. The idea behind quality assurance is to reach one's goals in the most efficient way possible, by improving work and production methods, technical solutions, etc. An important aspect of the quality assurance process is that goal fulfilment can be checked and verified.

An example of quality assurance work in the building and property management sector is the following up of goals to ensure that the indoor environment in a building is, or will be, acceptable. One of the most important aspects of the indoor environment is the quality of the air. Other important aspects include acoustics, lighting and thermal comfort. A good indoor environment is assured by carrying out quality assurance work throughout all the phases of a building project, i.e. when drawing up the building program, when carrying out planning and construction work, and when managing the finished building.

QUALITY ASSURANCE DURING THE BUILDING AND MANAGEMENT PROCESSES

The foundations for a good indoor environment are laid down at an early stage in the building process, although this can then be ruined by poor management. On the other hand, good management can, over time, improve a building with a poor indoor environment through structured activities aimed at clearly defined goals. In this case, management, operation and maintenance are planned based on an inventory of the faults that are in most need of remedial action.

THE CLIENT'S QUALITY ASSURANCE WORK DURING THE BUILDING PROCESS

The client is the player who, throughout the building process, makes the final decisions regarding the execution of the quality assurance work. The client's work to quality assure a good indoor environment mainly involves:

- Ensuring that there are people in *the client's own organization* who have the necessary competence in the field of indoor environment. In some instances this competency could be held by one of the project managers, while in others there could be one or more specialists who are responsible for and who monitor indoor environment issues. How this work is organized depends of the size and complexity of the building project.
- Stipulating clear *requirements* for a good indoor environment. The level of ambition will vary from building project to building project. In a rebuilding project, the requirements should be made based on what is feasible after carrying out a survey of the building. Note that it is the client's responsibility to comply with official regulations and, consequently, requirements regarding good indoor climate. Clients sometimes choose to go beyond the standard requirements for good indoor environments.
- Clarifying *how responsibility has been assigned* with respect to fulfilment of the requirements. The responsibility for fulfilling the client's requirements lies with those involved in the project. They must be able to verify that the stipulated requirements have been fulfilled. The client must demand that the consultant's and contractor's documentation and reports be comprehensive and complete.
- Checking that those involved are *sufficiently competent* or by ensuring that relevant competency is available. This can be done by evaluating their previous experience, reference projects and routines for self inspection as well as by looking at how much they have cost.

• *Following up the requirements* and checking that verification work has been carried out and reported and that the requirements have been fulfilled. The client should also carry out random checks and measurements.

The quality assurance work carried out to ensure a good indoor climate should, if possible, be coordinated with other quality or environmental work related to the project.

Figure 2. A suitable time for the client to follow up the quality assurance work is in connection with planning or site meetings.

Figure 3. Fulfilment of functional requirements stipulated for the indoor climate must be verified by carrying out appropriate measurements. The picture shows equipment for measuring emissions from furnishings.

THE CLIENT SPECIFIES THE REQUIREMENTS LEVEL

During the planning, building and management stages of a project, requirements must be stipulated for the different functions in a building, even from an indoor environment point of view. These requirements are dealt with in more detail in Chapter 14/The client and the building process. The following are examples of areas in which clear and verifiable requirements must be stipulated:

1. Sources of pollutants and the spread of pollutants
Sources that emit pollutants must be eliminated or their occurrence reduced. Important aspects to bear in mind include:

• Emissions from building materials. These can occur, especially if the materials have been subject to moisture or have been handled incorrectly. In some cases, however, materials can emit significant amounts of chemical substances without having been subject to moisture or incorrect handling (primary emissions). Choosing low-emitting materials can contribute to better air quality.
• Structures and production processes. These must be protected against moisture damage, as this could cause fungus growth on building materials or chemical emissions due to reactions with, or decay of, the materials. Carefully planned moisture prevention measures will help eliminate unsuitable structural solutions and production methods.
• Radon from the ground. This is an air pollutant that should, primarily, be dealt with at source, i.e. by preventing it from reaching the indoor environment. This can be done by making the foundations airtight.
• The domestic hot water system. This must be designed so that the risk of legionella growth is avoided. The temperature of the water should be checked regularly when the building is in use.

Figure 4. Requirements for achieving a good indoor environment can include the provision of weather and moisture protection measures for materials and components, to protect them against moisture and dirt.

2. Ventilation
Adequate ventilation must be ensured to remove pollutants emanating from the activities in the building and from the occupants, or to remove surplus heat. See Chapter 16/Air change and air flow.

3. Supply air quality
It is important that the air introduced into a building is sufficiently clean. The quality of the supply air is determined by the location of the air intake and its design. Supply air from outdoors might have to be filtered. See Chapter 18/Outdoor air intakes, and Chapter 20/Air filters and air filtration.

4. Airtightness

The airtightness of building components is important for achieving the planned ventilation of the all the rooms in a building, for ensuring that polluted air does not infiltrate into the building, for avoiding moisture damage, for avoiding poor thermal comfort as a result of draughts and cold surfaces, and for an efficient use of energy. Although theses requirements apply primarily to the building envelope, they are also important for protection levels between fire cells. See Chapter 12/Airtightness.

Figure 5. An airtight building envelope is another important requirement for good air quality and energy efficiency. There is sometimes a misunderstanding about whether or not leakage and infiltration through the building envelope is beneficial. A building must, of course, be allowed to breathe but this is a job for the ventilation system.

5. Thermal comfort

Thermal comfort depends on both the outdoor climate and how the building envelope and the building services have been designed. The requirements specification must make it quite clear how much the thermal climate can be allowed to deviate from its optimal value, for example, with respect to high indoor temperatures during hot and sunny days. See Chapter 14/The client and the building process and Chapter 10/Thermal climate.

6. Sound and sound attenuation

The acoustic environment is important in all buildings and depends on, among other things, the ventilation system, traffic noise and other noises from the surroundings. See Chapter 11/Building acoustics and Chapter 26/Sound and sound attenuation.

7. Cleanability

The quality of the indoor air is affected by how easy it is to clean the building and how cleaning is carried out. The cleanability of a building is a measure of how easy it is to reach different surfaces for cleaning and

also depends on the number of surfaces that collect dust and whether these surfaces are easy to clean.

If requirements regarding cleanability are stipulated, it is a good idea to include recommendations relating to the management process. Comments provided by property management organizations could lead to solutions that would be of great importance to care and maintenance routines.

THE PLANNING PROCESS

During the planning process, the design/project engineers must be able to show that it will be possible to fulfil the building's technical requirements. This must be clearly shown in appropriate documents with specific references to drawings, descriptions and calculations.

The client must also have an agenda for measures to be carried out during the planning process to ensure that the goals will be met in the finished building. Steps that are often taken in connection with quality assurance work during the planning process include:

- Providing information regarding the stipulated requirements to those taking part in the planning work.
- Carrying out self inspections to verify that the requirements have been met.
- Carrying out combined inspections to ensure that different technical solutions do not conflict with each other.
- Providing instructions to those engaged in the building process.
- Compiling information for the property managers about operations and maintenance, and about weak points to which extra attention must be paid while the building is in use.

THE BUILDING PROCESS

During the building process, all work must be carried out so that the stipulated requirements are met. Verification that the requirements have been met involves carrying out appropriate measurements and certifying that the project documents have been followed. Quality assurance work during the building process often means:

- Checking with the respective engineers that important operations have been completed.
- Providing on-site training for the contractors, informing about important stages in the building process and clarifying requirements that must be fulfilled.
- Documenting self inspection work.
- Documenting measurements and verification work.

• Compiling information for the property managers with regard to future operations and maintenance, as well as indicating weak points to which extra attention must be paid while the building is in use.

Figure 6. The airtightness of the building envelope can be easily assessed by carrying out airtightness tests. When these tests are carried out there is also often a search for leaks, so they can be identified and sealed. If the degree of airtightness is insufficient, this can affect the ventilation, the quality of the air, moisture protection measures, thermal comfort and energy use.

THE MANAGEMENT PROCESS

The property owner's management team can ensure that a good indoor environment is also maintained after the building has been completed. If the building is misused and poorly maintained, the quality of the indoor environment might decline. Some property managers strive to improve the indoor environment by systematically working towards clearly defined goals. What should a property owner/manager do to maintain or improve the indoor environment? The following suggestions might help:

- Create a management organization that has *clearly defined areas of responsibility,* not only to draw up routines, shown below, but also to make sure that they are followed and properly documented.
- *Provide personnel with the right tools,* so that they can tackle the necessary assignments. Cleaning staff and operating staff should also be offered training when necessary. Even the tenants might require training or regular information.
- Draw up *routines and checklists for operations and maintenance*. These can be based on information from the building process and recommendations from material and system manufacturers. In existing buildings, an inventory of the building will reveal weak points and identify risks that should be given extra attention when carrying out maintenance work. Routines and checklists should be drawn up for ventilation systems, heating systems, the building envelope and cleaning.
- Draw up *a maintenance plan*. This should comprise measures to be taken to eliminate weak points and to reduce any risks identified in the inventory.
- In certain instances, it might be advisable to provide *information to the users/tenants* about how the premises and the building services should be used, i.e. if there are any operational limitations. For example, the ventilation system in the building might be dimensioned for a certain number of people and the quality of the air will be affected, if this number is exceeded. Cleaning extract air terminal devices and kitchen fans will also contribute to good air quality.
- Create *routines for handling complaints* so that user's/tenant's points of view can be dealt with expediently. Complaints should be registered, assessed and quickly attended to when necessary. All documentation should be carefully managed, to avoid forgetting to deal with important complaints. Comments and complaints about measures that should be tackled immediately, for example, replacing light bulbs, need not always be registered.
- Keep *records* that show that operation and maintenance routines and maintenance plans have been followed. To avoid being caught between two stools, it is vitally important that unique responsibility is assigned to each and every aspect.
- Create a complete overview of the indoor environment situation by holding regular meetings to discuss relevant issues. In this way the users/tenants, cleaners and maintenance staff can report on the cur-

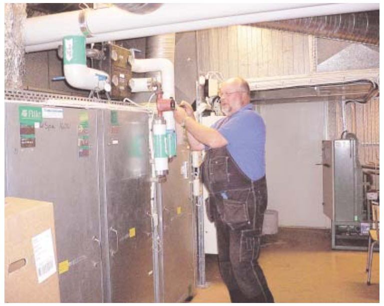

Figure 7. Staff must always be properly prepared, so that a good indoor environment can be maintained when the building is in use. Training them in their areas of responsibility and creating routines and checklists for their work is often a good investment.

rent situation and put forward *suggestions for improvements*. These could lead to improved routines, changes in maintenance plans and other measures of a technical nature.

Figure 8. Issues put forward by the users/tenants, cleaners and maintenance staff can be discussed at regular meetings. This also provides a good opportunity to put forward suggestions for improvements. These could lead to improved routines, changes in maintenance plans and other measures of a technical nature.

A typical building and building services inventory

An inventory can reveal a building's weak points, risk factors and faulty systems. These details, together with the proposed remedial measures, are an excellent basis on which to base operational and maintenance routines and, above all, for drawing up maintenance plans. If a building is to be rebuilt, it is a great advantage if remedial measures can be planned in time and not discovered during the rebuilding work or, even worse, after its completion.

When carrying out conversions or alterations it is often necessary to make an inventory of the building's properties, to assess whether it will be possible to run it efficiently with regard to energy use. To avoid sub-optimization, it is often a good idea to carry out energy-saving measures together with any remedial measures designed to improve the indoor environment.

Improvement measures are normally identified and planned based on an inventory, which mainly comprises the following:

- A review of all present documentation regarding the design of the building, the building services and the indoor environment. This often includes drawings, questionnaires, complaints records, ventilation test certificates and comments and, if necessary, energy use statistics.
- Details collected from maintenance staff, property managers, cleaners etc.
- An inventory of the building, carried out as an on-site investigation of the factors that could affect the indoor environment. These might include *the ventilation system, moisture conditions, the existence of any moisture-damaged materials, emissions and smells, structural risks (design drawings will be needed), factors that affect thermal comfort (for example, the condition of the windows and the building envelope, and the design of the heating system), airtightness, noise problems and radon.* The inventory should also include an overview of the factors that can affect the use of energy, such as the *condition of the building envelope and the design of the heating system and control system.* The initial inventory is carried out using non-destructive testing. Further measurements, tests and sampling can then be carried out if required.
- Remedial measures, based on the information above, can then be proposed to correct any faults or defects. Some conditions in a building might be difficult to correct so that all the indoor environment risks are completely eliminated. In cases like these, the property owner might have to consider accepting some of the faults and, instead, keep an eye on them by inspecting them regularly when the building is in use.

Figure 9. An inventory carried out before rebuilding will reveal critical construction solutions and faulty functions that could constitute risks for the indoor environment and the air quality. This means that major problems can be discovered in time, with great cost savings as a result. The photograph shows moisture measurements being carried out with a moisture meter, to find floor areas with excess moisture content. If floor coverings are placed on top of damp floors, chemical substances could be released and affect the quality of the air.

Concluding remarks

All over Europe, clients and property owners are trying to create good indoor environments and a number of systems for quality assurance of the indoor environment are now available. One of the systems used in Sweden is described below.

'P Certified' indoor environments

The quality assurance method described above is, to a great extent, sim-

FIGURE 10. Quality assurance of the indoor environment, in which third party inspection is included, is used in schools, day nurseries, office buildings, blocks of flats, shops etc. When third-party inspections are carried out checks are made to ensure that the requirements for good indoor environments have been fulfilled and that the management routines have been followed, so that good indoor environmental standards are maintained.

ilar to that which is used to quality assure good indoor environments according to the Swedish system for 'P Certified' indoor environments. In 'P Certified' indoor environments, a yearly third-party inspection is carried out to check that the requirements for a good indoor environment have been fulfilled, that management routines have been followed and that complaints have been properly dealt with.

It is a good idea to combine steps aimed at creating a more efficient use of energy with measures to ensure a good indoor environment. When drawing up energy declarations for buildings, an inventory of the building and the building services can also be carried out and, at the same time, it ought to be possible to evaluate the quality of the indoor environment. One of the key factors for the efficient use of energy is the existence of well-functioning building services, which are also of great importance where the indoor environment is concerned. The system for 'P Certified' indoor environments has therefore been extended to include energy use as well.

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EPILOGUE

The *Swegon Air Academy* hopes that this book will, to some extent, contribute towards an increased understanding of how a good indoor climate can help to make a building comfortable, efficient and profitable. We also hope that this book will increase understanding between the different players in the building industry and pave the way for an even better dialogue between property owners, architects, builders, consultants and installation contractors.

The chapters that discuss the present energy situation and how we ourselves affect the outdoor climate will, hopefully, cause us to reflect on these issues and stimulate active engagement. We can hardly prevent climate changes but we can join together to try and reduce them.

We would like to take this opportunity to thank all those who have made this book possible. In addition to all the contributing authors, we would like to thank Dr. Per-Erik Nilsson and his colleagues at CIT Energy Management for their valuable advice and especially Dr. Lars E. Ekberg, both for contributing three chapters and for his careful scrutiny of the other author's contributions.

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numerous ideas and suggestions, as well as a good dose of encouragement when I was almost ready to give up.

With your help and that of the authors, the result has not been a dry textbook but a timeless publication with a soul of its own. You don't have to read it from cover to cover in one go: read it as you please - just pick out a suitable chapter and satisfy your curiosity!

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The *Swegon Air Academy* is a forum for objective and company-neutral sharing of knowledge and experiences related to air handling and indoor climate issues.

One of our primary goals is to explain complex relationships in an intelligible way, so that those who are interested in a subject can understand it at a deeper level.

Via seminars, newspaper articles and literature, the *Swegon Air Academy* contributes to a greater awareness of the importance of indoor air quality for health and well-being, to an increased understanding of the energy issue and to a higher level of involvement in matters concerning the outdoor environment.

The *Swegon Air Academy* provides information and educational activities all over Europe and co-operates with well-known experts in relevant fields.

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