Technical Indoor Climate Guide

2014-03-13

Foreword

The aim of this Technical Guide is to serve as a reference work that is useful to both practising engineers and during different training courses. The Technical Guide deals with technical issues related to ventilation, air handling, indoor climate and energy housekeeping, but does not claim to be comprehensive.

It is fully possible to build poorly working systems with the best products available on the market. Consequently, it is extremely important to have knowledge of what is required to build well-functioning systems with maximum possible comfort and the lowest possible energy consumption. A client rarely has the necessary skills to assess and predict the consequences of a specific decision. It is therefore the responsibility of the consultant to present the technical and financial consequences of decisions made during a project.

The client, of course, does not want to pay more than necessary for a particular action or solution. And here lies a danger — you get what you pay for! Today the most common procurement methods focus primarily on the price at the time of procurement. We tend to forget the very purpose of the air handling installation — to provide a good indoor climate that creates ideal conditions for working and residing in the served premises.

As there is also always considerable focus on realising as low energy consumption as possible, the likelihood of disregarding the requirement of good indoor climate increases. One condition for being able to achieve a good indoor climate without using excessive energy, is to have an understanding of the complete picture. This Technical Guide aims to provide essential basic information and knowledge about how the relationship appears.

Included at the end of the Technical Guide are definitions, designations, conversion factors, formulas, thermal data for air and water as well as further reading tips, publications which in neutral terms describe matters that may be useful to know from a designer, installer or user perspective.

Laws, standards and regulations may differ in different countries and it should be noted that this Technical Guide in many cases refers to Swedish standards and regulations. It should also be noted that standards and regulations change with time.

A large number of industry experts have contributed with content and have reviewed and proofread the guide, but Swegon assumes no liability or responsibility for any errors.

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Content

Purpose of the climate installation

Ventilation air is utilised for different purposes. Its main task is to remove contaminated air and replace it with clean and tempered air. Contaminated air can refer to impurities in the form of gases and particles, but excess heat can also be regarded as an impurity in some cases.

Another important task of ventilation is to create a good indoor climate without draughts and with small temperature differences in the occupied zone. The room climate is composed of a number of factors including air velocity, air temperature and radiation temperature. An imbalance of one or more of these factors in rooms used by people may cause draughts.

In order to maintain these basic functions the air handling installation must be designed so that:

- it is stable against interference. Interference may be partly external, mainly wind and temperature as well as partly internal interferences, for example, convection currents from various heat sources. Stability against external interference requires, among others, that the installation is designed with a minimum pressure drop.
- they are easy to control and measure. The installation must be designed for air terminals with fixed measuring tappings or fixed measuring devices in main and branch ducts. The installation of fixed measuring devices significantly reduces the time needed for commissioning as well as the overall cost.

A fully functional total solution requires collaboration between installation technicians, builders and architects. The ventilation consultant needs to enter the construction process at an early stage to specify requirements for requisite areas, the consequences of the building design, etc. so that the demands on the indoor climate can be met at the lowest possible cost.

Why ventilation?

The word ventilation comes from the Latin, 'ventilare', which means being exposed to the wind. Today, we have given the word vent a more specific meaning, namely air exchange. We replace used air with clean air.

To replace used air with clean air requires energy. New, more stringent energy requirements risk leading to reduced airflow, which in turn can result in a repeat of the mistakes made in the 1970s in Sweden. When a new building code (SBN 75) came into effect in the mid-1970s, airflows were restricted at the same time as demands on the air tightness of buildings were strengthened. The result was what came to be known as "sick buildings" and SBS ("Sick Building Syndrome"). These sick buildings have been shown to have the following negative effects on the health of individuals:

- Irritation of eyes, nose and throat
- Rhinitis and nasal congestion
- Repeated respiratory infections
- Headaches
- Tiredness
- Heavy headedness, concentration difficulties
- Nausea and dizziness

The cause of "sick buildings" is a complex issue. The following points represent an attempt to compile the most significant causes:

- An attempt to minimise production costs
- Inadequate documentation
- Procurement procedures
- Built-in moisture in houses
- Neglected operation and maintenance
- Energy savings through reduced airflow
- New and unproven building materials
- Inadequate cleaning

Lessons

What we can learn from history is that demands on energy savings must never be pushed so far that demands on adequate airflow are jeopardized. Accordingly, good ventilation is not only important for the indoor climate to be perceived as good, but also central to the health of people.

Why a good indoor climate?

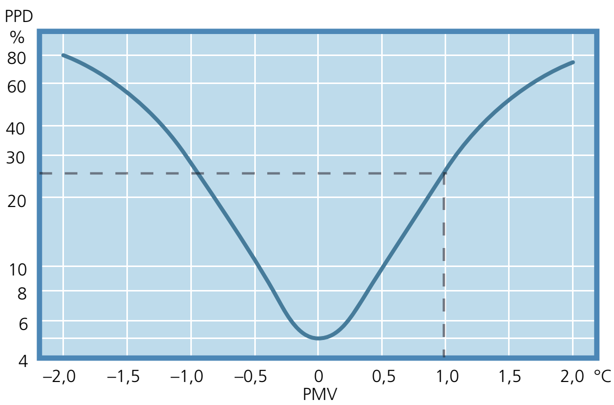
Experienced indoor climate

An individual's indoor climate experience is an interaction of several factors that affect our thermal comfort:

- Activity level, the body's heat production
- Clothing's heat resistance
- Ambient air temperature
- Ambient surface temperature
- Relative air velocity
- Relative air humidity

According to professor P O Fanger an ideal indoor climate is produced when individual experiences thermal comfort, i.e. when a person is thermally neutral. However, one of the problems always faced when creating a good indoor climate with the help of a climate installation, is that people perceive the environment differently. No matter how well you succeed, about 5% will still be dissatisfied. The percentage of those dissatisfied then increases for each degree of variance from the average person's most ideal temperature.

The diagram shows the percentage of dissatisfied when the temperature varies from the average ideal temperature.

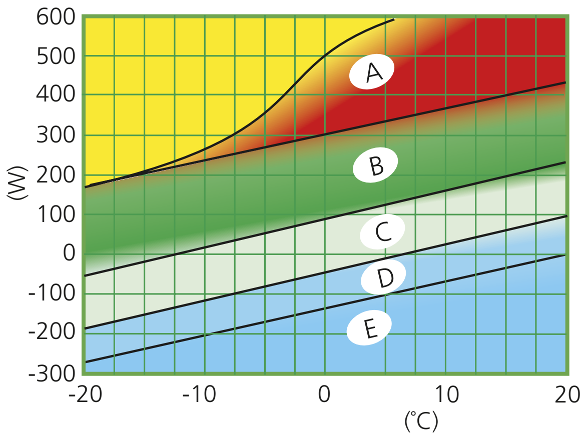


Expected percentage of dissatisfied (PPD = Predicted Percentage Dissatisfied) as a function of the thermal comfort experienced (PMV = Predicted Mean Vote).
Example (dashed line): If the temperature is 1 degree warmer than the ideal temperature, 25% will be dissatisfied.

Heating and cooling load in a normal office

Here is an example that illustrates the heat balance in an office. The room has windows with energy-efficient glass, U-value approx. 1.3 W/m²,K. The graph shows the heating and cooling needs in the room are affected by different heat sources that usually affect an office.

The person emits about 100 W. When the lighting is switched on a further 120 W is added. When the person, lighting, computer and solar incident radiation are taken into account the load amounts to about 650 W in the room. At an outdoor temperature of –18°C, the heat loss through the exterior wall, window, etc. for the room is approx. 180 and 300 W respectively, depending on how the room is located in the building. The higher value refers to a corner room. The room's cooling load is, in almost all cases, greater than the room's heating requirement even when the outside temperature is –18°C.

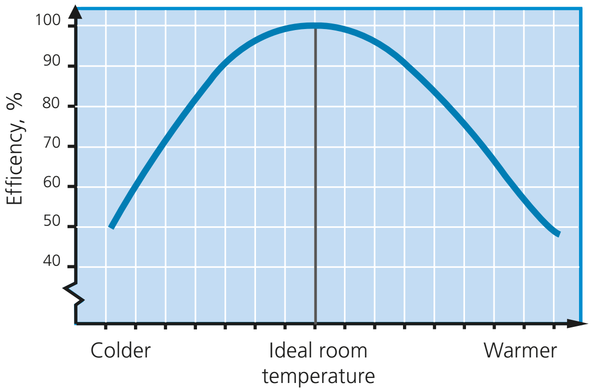


Heating balance in a normal office
A = Solar incident radiation
B = Computer
C = Lighting 120 W
D = Person 100 W
E = Transmission

Performance capacity

What we perceive as an ideal indoor climate is always individually from person to person, and studies show that the performance capacity follows the perception of an individual. In his research Professor David Wyon has, among others, studied people in different work situations.

The diagram shows that the deviation from the ideal temperature is important. You can measure a decline in efficiency, even after a few degrees deviation from the ideal. This means that it is possible to make a calculation of what an investment in the indoor environment can signify, and in doing so, motivate that the experience is not only positive but also efficient.



Efficiency as a function of the deviation from the ideal temperature.

Is comfort cooling motivated to create a good working environment?

Several independent trials and research projects have demonstrated significant economic benefits, linked to improvements in the working environment. The results relate among others to normal office work (Seppänen and Fisk as well as Wyon and Wargocki) but also students' school work (Wyon and Wargocki).

As school work is difficult to measure in money, we satisfy ourselves here by focusing on how the thermal indoor climate affects those who work in an office environment. The following effects have been possible to establish:

- A too high indoor temperature distracts those working in the room, creating complaints, which result in increased maintenance costs.
- A too high indoor temperature makes us lethargic, aggravates sick building symptoms, and have a negative impact on our mental capacity.
- A too low temperature gives cold fingers so that we become less dexterous.
- Quick changes in temperature have the same effect on office work as a slightly increased room temperature, while slow temperature changes only causes discomfort.

Studies of how the performance capacity of office employees is influenced by the room temperature have also come up with the following results (Seppänen et al. 2006b):

Most people who work in an office environment with typical office duties, perceive 22°C as the optimum temperature. Greatly simplified it is possible, based on existing studies, to conclude that for every degree the temperature deviates from the ideal (up or down), our efficiency drops by about 1%. At an indoor temperature of 30°C, we have lost approximately 10% of our capacity. If you consider the wages at a workplace and the number of employees and put the reduced performance in relation to the cost of investing in comfort cooling, you will soon conclude that it is a very profitable investment. The payback period is often no longer than approximately two years.

From requirement to technical solution

Indoor climate requirements

In order to select and design a ventilation system that creates both good air quality and good comfort, it is important to take into account all relevant factors - above all the demands made by the client/user. The climate concepts you normally think of include:

- Air quality
- Air velocity
- Thermal climate
- Acoustic environmental
- Lights/lighting
- Air humidity

The Swedish Indoor Climate Institute's "R1 Classified indoor climate systems - Guidelines and specifications", describes in detail how the different requirements need to be formulated.

Indoor air quality requirements are usually stated according to the air quality classes AQ1 and AQ2. It is assumed for both classes that the air does not contain contamination in concentrations that can be considered to have a negative effect on the health of humans. Even though the level of carbon dioxide in indoor air does not constitute a major problem, it has been shown to act as an indicator and for some time been used to evaluate the quality of indoor air.

Thermal climate requirements are stated based on the current version of SS-EN ISO 7730. Several factors need to be foreseen and defined for each situation/room, for example, the users' physical activity level and clothing. The physical activity is measured in the unit "met" and the affect of clothing in the unit "clo". We speak about two different classes for thermal climate, TQ1 and TQ2. It is important to set target values for the floor temperature and limits for the vertical difference in temperature, radiation temperature symmetry and risk of draught. Air velocity in the occupied zone and the operating temperature have a major effect on the perceived comfort.

Noise is one of the environmental disruptions indoors that is perceived by most as extremely troublesome. Target values for noise from installations are specified in the Swedish standard SS 025267 and SS 025268. There are good reasons to try to produce indoor environments, which with a good margin meet the authorities' sound requirements. Three sound quality classes - NQ1, NQ2 and NQ3, where NQ1 is the best class, are used to specify the demands on sound requirements.

The lighting conditions are of great importance for how to perceive a room. The perception of the visual environment is largely subjective and also dependent on our age. An elderly person needs significantly more light than a younger person to experience the light as sufficient. The lighting system in a room must be performed according to the current versions of the Swedish standard SS-EN 12464-1 and SS-EN 12665. A requirements specification must be drawn up in accordance with the standard, which prescribes appropriate target values for illuminance, glare index and colour reproduction index for a large number of rooms and activities.

All factors that contribute to the perceived indoor climate, also affect to some extent the energy requirement, which is why the indoor climate should be considered from a holistic perspective.

Air quality

General

Obviously, the ventilation system is a very important element in maintaining good indoor air quality. Sometimes you can suspect poor air quality, but the problem is in fact due to a too high temperature. In order to ensure good air quality in a room it is important to take into account the airflow requirement, as it will take away both contamination and excess heat. How well this is carried out has a great impact on comfort, which in turn affects both work performance as well as safety.

Uncertainty is greater with respect to people's perceptions of air quality than for thermal environment. Danish studies show that the number of dissatisfied users is approximately 14%, if everyone is exposed to a carbon dioxide level of 800 ppm*. A carbon dioxide level of 800-1000 ppm has also become a widely used requirement level for air contamination.

The carbon dioxide level is detected by a carbon dioxide sensor, which is usually placed in the extract air. There are a number of other emissions that can be generated in an occupied zone and an air quality sensor of the type VOC** detects in addition to carbon dioxide numerous other contaminants.

The air quality is affected by many factors, such as, emissions from building materials, secretion from people, etc. Smoking is an example of pollution that has a negative effect on air quality.

Air humidity

Air humidity is the parameter that normally has the least effect on the experienced comfort. At times, it may even be so that other parameters such as air contamination are associated with the air humidity.

However, if the capacity of the chiller to dehumidify the supply air is insufficient, moisture in the air will be a troublesome factor during the summer.

*) ppm = parts per million
**) VOC = Volatile Organic Compounds

Room air, air quality

It occurs, for example in published European guidelines, which target values for good air quality based on an evaluation in terms of the expected percentage of dissatisfied. However, in the Swedish "R1 Classified indoor climate systems - Guidelines and specifications" the meaning of both the air quality classes with qualitative terms are described according to the following table:

Air quality class	Meaning
AQ1	Normally very little risk of disturbances in the form of odours
AQ2	Normally little risk of disturbances in the form of odours However, odours are perceived briefly immediately after the premises are occupied.

It is assumed for both quality classes, in addition to that set out in the table, that the air does not contain contamination in concentrations that can be considered to have a negative effect on the health of humans.

Air quality class	Premises
AQ1	The carbon dioxide concentration of the room air should not permanently exceed 800 ppm during normal room usage (about 400 ppm above the carbon dioxide level in the outdoor air)
AQ2	The carbon dioxide concentration of the room air should not permanently exceed 1000 ppm during normal room usage (about 600 ppm above the carbon dioxide level in the outdoor air) This class corresponds to the Swedish authorities' general advice.

The airflows required to achieve the desired air quality class, are highly dependent on a number of external factors. Generally, it applies that regulatory requirements and general advice concerning the maximum permitted concentrations of airborne contaminants and hygiene airflows must be followed. Some guidelines regarding the maximum permitted contaminant concentrations to reduce the risk of ill health, are also given in "R1 Classified indoor climate systems - Guidelines and specifications".

Airflow, air quality

A properly planned and dimensioned ventilation system gives a slightly higher air exchange efficiency, if the system is implemented according to the principle of displacement ventilation compared with a mixing system. However, in practice the differences are not so great. The same airflows can therefore be used for both options.

Generally, the following formula can be used to determine the requisite airflow from an air quality perspective:

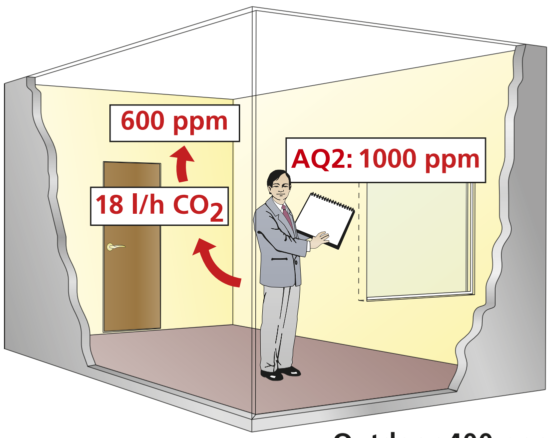
$$q_v = \frac{m \cdot 10^6}{C - C_{in}} \quad \text{(l/s)}$$

q_v = ventilation airflow (l/s)
m = generation of contamination (l/s)
C = recommended highest level of contamination (ppm)
C_{in} = the initial concentration of contamination (ppm)

General CO_2 release (m, generation of compounds) from a human per kg bodyweight

Activity	CO_2 l/**h**/kg	CO_2 l/**s**/kg
Rest, lying	0.17	0.00005
Seated	0.26	0.00007
Standing	0.30	0.00008
Walking	0.35	0.00010

General initial concentration (C_{in}) is at its lowest about 350 ppm. Significantly higher values can occur in a city centre.



The requisite airflow for an office where sedentary work is performed and with air quality AQ2

Example:

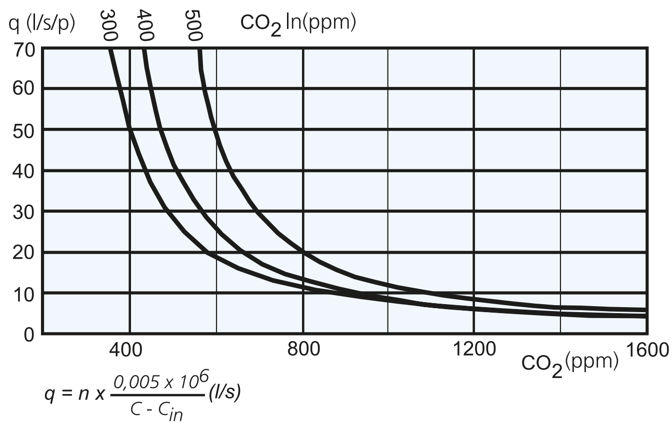
Office with individuals as the dominant source of contamination. A seated individual weighing 70 kg produces about 18 litres of CO_2 per hour, i.e. 0.005 l/s. At an initial concentration of 400 ppm CO_2 the required airflow per person would thus be:

$$q_v = \frac{5000}{C - 400} \quad \text{(l/s, person)}$$

If the requirement of the highest permitted CO_2 level in air quality class AQ1 is 600 ppm and in air quality class AQ2 1000 ppm, the requisite air flows will be:

AQ1 (≤ 10% are expected to be dissatisfied with the climate) = 25 l/s, person

AQ2 (≤20% are expected to be dissatisfied with the climate) = 8.3 l/s, person



Max. airflow as a function of the air quality.
(Seated persons, 70 kg)

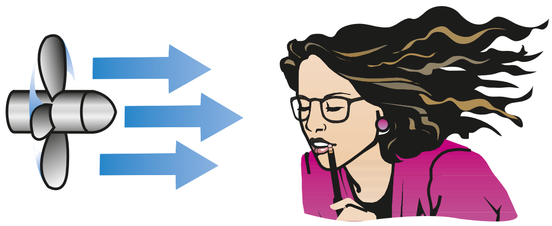
Air velocity

That the air velocity has an impact on the perceived indoor climate is a known fact, but perhaps not how it relates to the interaction between air temperature and air velocity. A simple analogy can be made by stretching out your arms through the side window on a hot summer day and let the speed of the wind cool you. The experience will not be as enjoyable if you do the same experiment in the winter when it's -10°C in the air. You can then understand that it is not only the velocity of the air that is interesting, but also the temperature. ISO 7730 demonstrates this through something called a draught rating index (Dr-index), a ratio that is made up of:

- Air velocity
- Turbulence intensity
- Air temperature

The Dr-index can be calculated e.g. for a given position in a room by measuring the air temperature and air velocity over time (and calculate the turbulence intensity, which is a measure of how much the air velocity varies during the measurement procedure, how “agitated” the air is) The result is that the colder it is in a room, the lower air velocity can be tolerated and vice versa, a higher air velocity can be accepted if the temperature is higher.

The aforementioned is the reason why you often have different maximum velocity requirements in the summer and winter. In the winter, it is preferable that the air velocity in the occupied zone does not exceed 0.15 m/s. The corresponding velocity in the summer should not exceed 0.25 m/s.



Air velocity has an impact on the perceived indoor climate.

Thermal indoor climate

General

When formulating the requirements for the thermal indoor climate, the target value for the operative temperature should be determined according to the current version of the standard SS-ISO 7730. A temperature range that is normally expected to be experienced as comfortable needs to be determined. What is a suitable range, depends on users' physical activity and their dress. Accordingly, these factors must be predicted for each situation/room. The physical activity is measured in the unit “met” and the affect of the clothing in the unit “clo”. In order to determine these parameters, see “R1 Classified indoor climate systems - Guidelines and specifications”.

The target value for the operative temperature should be stated in the requirements specification as well as details to what extent you are prepared to accept deviations from this. The ventilation planner should show the client what various alternative levels will mean when it comes to the complexity and cost of the installation.

Temperature concept

- Temperature: typically shows clear differences between people.
- Operating temperature: approximate mean value of the surrounding room surfaces’ temperature and the room air temperature.
- Directed operative temperature: large difference due to the radiant heat exchange. For example, local heat deficiency is created at a cold window, which is perceived as draught.
- Temperature radiation asymmetry: the difference between the plane radiant temperature on the two opposite sides of a cross-section of an individual's physical centre.

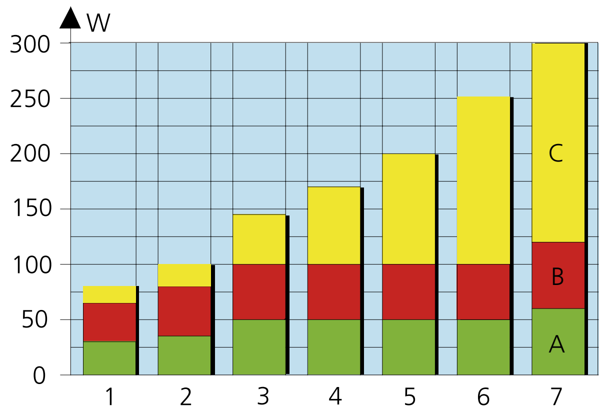
Heat transfer

An individual’s heat exchange with the surroundings occurs mainly in three ways, which are:

- Heat emissions through radiation to the surrounding surfaces or to free space.
- Heat emissions through convection to the surrounding air.
- Heat emissions through evaporation of liquid, which is mainly due to sweat, evaporative release.

A fourth form of heat exchange can also occur through conduction to solid or liquid objects in direct contact with the body surface. However, in normal cases this element is so small that it is completely negligible.

As the diagram shows, the heat emissions from the body vary with the type of activity. Especially the evaporative emissions which increase with higher activity.



Heat emissions from the body during different types of work.
A = Convection (green)
B = Radiation (red)
C = Evaporative emissions (yellow)
1 = Total rest
2 = Light office work
3 = Normal office work
4 = Computer work
5 = Light physical work
6 = Gentle walk
7 = Painting work

Radiation

Radiant heat is emitted continuously from warm to cold surfaces and increases with the temperature difference between these. In summary, radiant heat exchange is dependent on the following factors:

- Size and placement of surfaces = solid angle relative to each other
- Temperatures of individual surfaces
- Condition of the surfaces which determines emission and absorption indices, i.e. the ability to emit and receive radiant heat

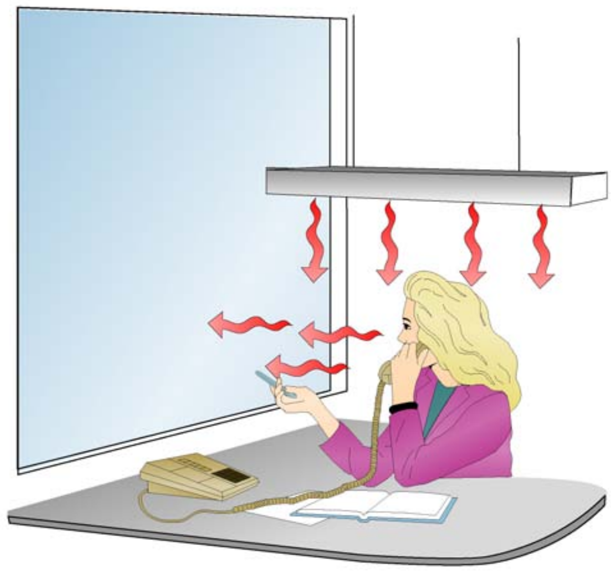
There are two different types of radiant heat:

- High temperature radiation from bodies with temperatures above approx. +500°C
- Low temperature radiation from bodies with temperatures below approx. +250°C

The above temperature limits are only approximate specifications.

As room surfaces and conventional heaters have relatively low temperatures, an individual's heat exchange with ambient indoor surfaces occurs in the form of long-wave low-temperature radiation. In the event of low temperature radiation, the structure and colour of the surfaces have practically no effect on the ability to emit and absorb thermal radiation, with the exception of pure untreated metal surfaces. Examples of low temperature radiation sources' are panel radiators and ceiling and underfloor heating systems.

For calculations of the indoor climate within the temperature range - 50 to + 100°C the heat exchange between the room surfaces is made up of invisible long-wave low-temperature radiation. One can also completely neglect the radiation exchange through emission and absorption from and to the room air. Accordingly, for room heat balance calculations consideration should only be given to low temperature radiation from the different surfaces of the room, walls, floors, ceilings, furniture, heaters, etc., and from any incoming high temperature radiation from the sun.



Radiation exchange occurs between all surfaces that have different temperatures independent of direction.

The heat exchange via radiation between separate room surfaces is usually expressed with the following equation:

$P_S = \alpha_S \cdot A1 \cdot (t_1 - t_m)$ [W]

P_S = exchange of heating capacity in W between surface A_1 in m^2 which has the temperature t_1 with all other room surfaces which combined have an average surface temperature t_m

α_s = thermal exchange constant for radiation in W/m^2 K

With good accuracy (deviation less than 2.5%) a_s within the temperature range 0 to + 100°C can be noted for t_1 and t_m.

$\alpha_s = \varepsilon_0 \cdot (4.38 + 0.034 \cdot (t_1 + t_m))$ [W/m^2 K]

ε_0 = emissivity

Convection

If a surface is warmer than the room air, it emits heat to the room air. Similarly, the room air emits heat to a surface that is cooler than the room air. This form of heat transfer is called convection, and is divided into:

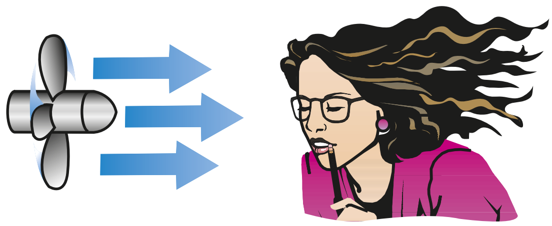
- Natural convection
- Forced convection

Natural convection is obtained through the differences in density in the different layers of air, which are created by the temperature difference between the air and the different bodies that the air flows against or around. The air closest to the body surface has a different temperature than the air outside of its boundary layer, and on account of this, the individual layers have a different density and different types of flows occur in relation to the shape, temperature deviation and location of the bodies.

Forced convection is obtained for example, by the action of a fan, i.e. the fan determines the air's flow. In this case, the air is driven around the body surfaces by forces other than differences in density.

In summary, convective heat transfer is affected by the air's flow along the body surface, the size of the surface and the temperature difference between the body and air.

The heat exchange per unit area increases with increased air velocity, reduced size of the surface and increased temperature difference between the body surface and the air.



Forced convection

Induction

Induction is a form of forced convection that occurs when a high speed jet of air passes stagnant air which is then drawn by the air jet. The air jet then grows in volume.

The induction principle is used in both active chilled/climate beams and comfort modules in induction devices.

Evaporation, evaporative release

When liquid turns to gas vaporization heat is consumed. When a human sweats, this vaporization heat is mainly taken from the body surface which is then cooled. Humans emit heat through evaporation. The heat transfer through evaporation and convection also occurs via breathing.

Heat emissions due to evaporation depend on the relative humidity of the room air. At normal room temperatures between about +18 °C to +25 °C and normal relative humidity of approx. 20-50%, this effect is however very small.

Unless the relative humidity level is maintained at a good level, max 45% at 25 °C, but rises up to 60% and above, the skin surface will be moist. When this occurs vaporization is more difficult. It is therefore important that the climate installation is designed so the moisture content in the building is not too high. This is done by equipping the central supply air unit and comfort cooling installation with sufficient cooling capacity in order to dehumidify the incoming air before it is fed to the premises.

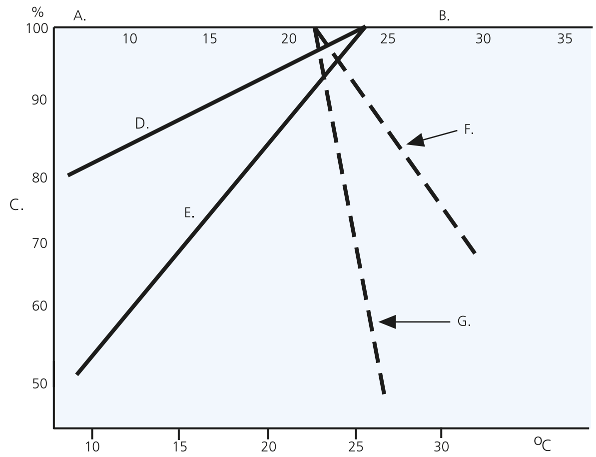
Temperature

The air temperature is the parameter that is easiest to understand and where most have their own experiences of human differences.

The sanitary inconvenience limit in terms of the air temperature in a room is below 18 °C or above 28 °C.

The temperature range for good comfort is considerably narrower and typically in the range 20-24 °C. It is difficult to satisfy everyone's need of appropriate temperature and you will always need to use clothes as a comfort controller.

A person's perception of the indoor climate is also vastly affected by the physical activity performed. The greater the physical activity, the lower the desired temperature.



Changes in work performance with the indoor temperature (according to Wyon).
A = Vigorous work (0.6 clo)
B = Sedentary (1.0 clo)
C = Work performance
D = Manual work
E = Finger dexterity, feeling
F = Mental performance (at effort)
G = Work rate (work in progress)
°C = for sedentary work (60 W/m²) and summer clothing

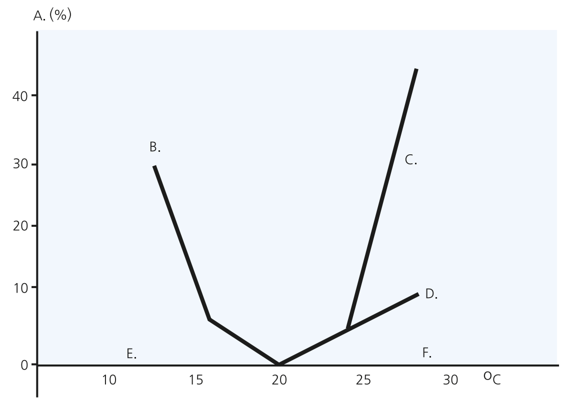
Work performance

The effect of the room temperature on a person's work performance is shown in the diagram in the top, right-hand corner. The diagram shows schematically and extremely simplified results from different trials with mental and physical performance. It is striking how quickly mental performance and the work rate fall with an increasing room temperature. It is therefore easy to show that it is profitable with a good air conditioning system.

Example:
Summer dress, sedentary work (office).
Room temperature 25 °C.
In relation to the comfort temperature the work rate has dropped to 70% and mental performance to 90%, i.e., the employer gets a most 70% from his employees at this higher temperature.

Assume that the cost per hour and employee including lost contribution margin = SEK 500.
Assume that the room temperature exceeds 25 °C, about 100 hours per year. The loss per year is then:
0.3 x 500 x 100 = SEK 15,000/employee.
Additional investment for a good air conditioning system is at most about SEK 300 /m².

With 20 m²/person results in an increases investment cost of SEK 6000/person, that is to say the investment pays for itself after the first summer season.



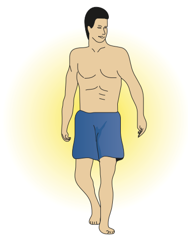
The accident rate for factory work changed with the indoor temperature (according to Wyon).
A = Increase in the number of accidents in %
B = Accidents
C = Men
D = Women
E = Vigorous work (0.6 clo)
F = Sedentary (1.0 clo)

Accident frequency

Another incentive for a good indoor climate is specified in the diagram to the bottom right. The diagram shows schematically and extremely simplified the relationship between workplace accidents and deviations from the comfort temperature.

Clothing

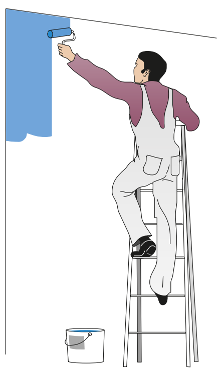
The effect of clothing is simple to understand, but no less complex to manage. A normal situation on a summer's day is that women are more lightly dressed than men, for example, a skirt and thin blouse in comparison to the men's trousers and shirt, and that this is a sufficiently strong parameter in itself to justify different temperature levels in the room. Thus, the summer situation can directly be a source of conflict in an office landscape where many people work together.



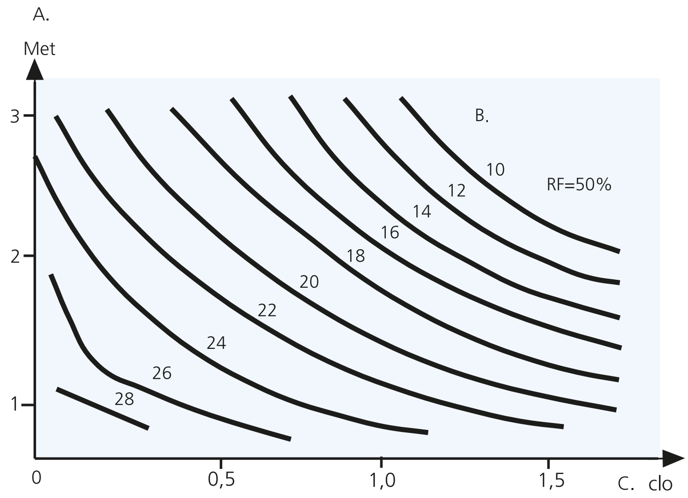
Clothing is extremely important for how you perceive climate.

Activity level

The activity level, or simplified, which work duties you perform, is also a highly significant parameter. More mobile work requires a lower temperature than sedentary work and combining different levels of activity in one and the same premises may therefore be inappropriate.



An individual's perception of the indoor climate is affected by the activity being conducted. The greater the activity, the lower the desired temperature.



Relation between the optimal operative temperature and met and clo values.
A = Activity level
B = Operative temperature (optimal) °C
C = Clothing clo

clo = unit of measurement for the insulation ability of clothes to experience good comfort at given external conditions. Usual indoor clothing during the winter has 1 clo while normal indoor clothing during the summer has 0.8 clo. Being naked is equivalent to 0 clo.

Operative temperature

The operative temperature is and approximate mean value of the surrounding room surfaces' temperature and the room air temperature. Consequently, the perceived temperature is affected equally by the temperature of surrounding surfaces and the room air. Since different room surfaces such as windows, exterior walls, interior walls, floors and ceilings, etc. have different and varying temperatures and orientation in the room, the operative temperature can also vary for different directions.

Directed operative temperature

You can perceive the radiation that arises at a cold window at the same time as perceiving a slightly higher temperature on the room's interior surfaces. The reason for this phenomenon is that the radiation heat exchange to the cold window is substantially greater than in towards the room.

If, in this way, the difference in directed operative temperature becomes large, the room will be cooled by the body surfaces that face towards the window. This local heating deficiency is perceived as draught. The cause of the draught perception does not need to be due to excessive air movement, but can also be caused by local parts of our body being subjected to a large radiation heat exchange to a cold room area.

Temperature radiation asymmetry

It is well known that it can be uncomfortable to remain close to a cold surface for a long period, for example, a large window of inferior quality, or a hot surface, such as a hot radiator. This relationship has been examined more closely at the Heating and Air-conditioning Engineering Laboratory at Technical University of Denmark. Where, in a climate chamber, they exposed subjects to different irregularities in the thermal radiation field.

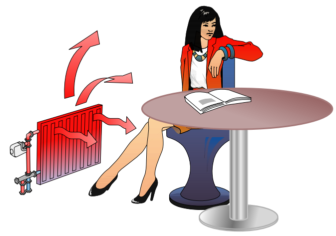
They discovered the limits where 5% of subjects experienced thermal discomfort. However, this is when they are in heat balance with the environment, i.e. when the comfort equation is satisfied. These limits are shown in the illustrations to the right.

Radiation asymmetry is defined as the difference between the plane radiant temperature on the two opposite sides of a cross-section of an individual's physical centre. The figures show the values of Dtpr* which in typical situations will result in 5% experiencing thermal discomfort due to the oblique radiation field. The air temperature is estimated to be equal to the surface temperature of the room's other surfaces.

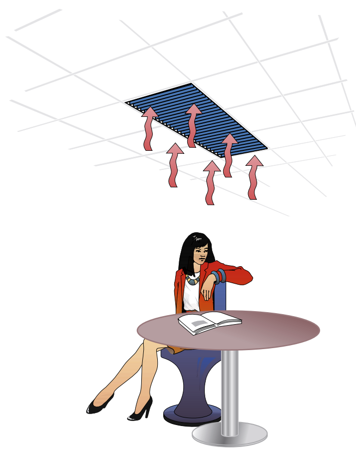
It is evident that people can accept large irregularities in the form of cold radiation from above and hot radiation from the side. However, they are less tolerant of cold radiation from the side and have very little tolerance when it concerns hot radiation from the ceiling.

**) Dtpr stands for the difference between the room temperature and the temperature of a surrounding area and should not be too large so that the operative temperature is experienced as unpleasant.*

Cold surface to the side
Dtpr < 10K



Warm surface to the side
Dtpr < 23K



Cold surface above the head.
Dtpr < 14K

Warm surface above the head.
Dtpr < 5K

Airflow, excess heat

The required ventilation airflow to remove excess heat is determined for both mixing and thermally controlled systems by the temperature difference between the extract air and the supply air.

For a mixing system, the extract air temperature (t_f) normally does not differ more than a degree or so from the room temperature 1.1 m above the floor (usually the reference point for the room's temperature). However, for a thermally controlled system, t_f can with normal ceiling heights, adopt values 3-5 °C above the room temperature.

The supply air temperature is generally limited to:

- 15 °C with mixing ventilation
- 18 °C with displacement ventilation

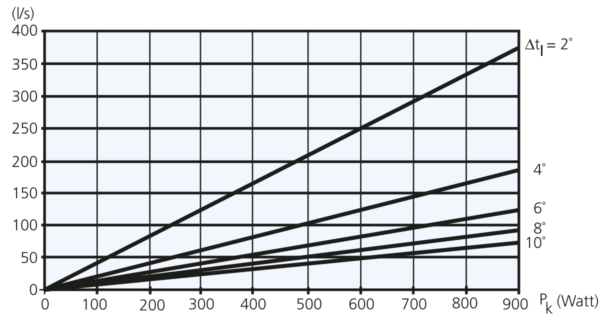
The temperature difference (t_f - t_t) is therefore at maximum cooling load virtually the same for the different ventilation principles and thus the required airflows. One condition in order to utilise large temperature differences between the extract air and the supply air is that:

- For mixing systems:
  The supply air diffusers are dimensioned correctly.
- For displacement ventilation:
  The supply air diffusers provide very good air distribution over the entire floor surface and that the air distribution can be controlled according to the appearance of the occupied zone.

The following equation can be applied to determine the requisite ventilation flow to remove excess heat:

$$q = \frac{P}{\rho \cdot C_p (t_f - t_t)} \quad \text{(l/s)}$$

where q = air flow in l/s
P = cooling load in W
t_f = extract air temperature °C
t_t = supply air temperature °C
ρ = air density (1.2 at 20 °C)
C_p = air's specific heating capacity (1.0 at 20 °C)



$$q = \frac{P_k}{1{,}023 \times 1{,}2 \times \Delta t_l} \quad \text{l/s}$$

Max. airflow as a function of the cooling load.

Sound/Acoustics

General
Today noise from the ventilation systems is one of the most common causes of complaint about the indoor climate at workplaces and homes, and a common source of disturbance to the surroundings. Unfortunately, standards and requirements are not always met in practice and the requirements are not always comprehensive. In order to design a fully functioning, quiet ventilation system with a high degree of comfort, all components must be of the highest quality and used correctly.

Totality
All quality products are designed based on their impact on the four climate areas, which first and foremost determine comfort in the room. In addition, energy efficiency and economic aspects are also weighed into the product design. The four climatic areas are:

- Air quality
- Thermal climate
- Acoustic climate
- Visual climate

Actions on the ventilation system directly affect the three top areas while actions in the room also affect the fourth. This working method means that the acoustics, noise from the ventilation system and other acoustic effects of the ventilation system, are assessed with the same weight as the main task the ventilation system is designed to perform.

Requirements specification for sound
The Swedish Indoor Climate Institute's "R1 Classified indoor climate systems - Guidelines and specifications", describes the following:

Requirements to limit noise from installations in the premises must be formulated in line with current version of SS 025 268 *Acoustics - Sound classification of spaces in buildings - Institutional premises, rooms for education, preschools and leisure-time centres, rooms for office work and hotels.*

Target values for different types of premises can be found in the standard.

Requirements concerning traffic noise, step sound insulation, airborne sound insulation and reverberation in rooms should be formulated based on SS 025268.

Requirements to limit noise from installations in the homes must be formulated in line with current version of SS 025 267 *Acoustics - Sound classification of spaces in buildings - Dwellings.*

Demands on acoustic comfort
In order to ensure that all noise levels are sufficiently low and at the same time have a character that is acceptable without a dominant low frequency noise, it is recommended that demands are made on both dB(A) and dB(C). It is important to check the impact of the products on the low frequencies. In documentation values are often specified down to the standard value of 63 Hz. Calculations are required for lower frequencies.

Example of requirement values
For an acceptable sound level in the room, it is recommended that the requirement on the dB(C) value is at most 15 dB higher than the requirement value in dB(A).

Residences	30 dB(A)	45 dB(C)
Offices	35 dB(A)	50 dB(C)
Classrooms	30 dB(A)	45 dB(C)
Conference room	30 dB(A)	45 dB(C)
Hospital ward	30 dB(A)	45 dB(C)
Cleaning room	45 dB(A)	60 dB(C)

It is important that all areas where individuals reside have a noise requirement. Values higher than 45 dB(A), 60 dB(C) should not be accepted in rooms other than plant room, machine rooms and the like with their own loud noise sources. In all contexts, sound levels should be sought so noise from the ventilation system does not become dominant.

Why dB(C)?
Historically three different weighting curves were formed, A, B and C. It was thought that A-weighting would be used to measure weaker sounds, B-weighting for the medium strength and C-weighting for the loudest noise. In all contexts, the loudness was to be measured in a similar way as the ear does. International comparisons showed that none of the suggested measurement values were better than any other. In order to specify loudness the measurements were simplified to only use A-weighting. In many contexts, this provides a reasonable understanding of the sound's audible strength. However, sound generated by noise sources with a very low frequency are judged incorrectly because A-weighting obviously attenuates low frequency too much. Especially when the assessment is not only loudness, but more the strength of disturbance in the room for individuals during their activities. Fans in the ventilation system can be a source of noise with strong low-frequency content.

When selecting products their low frequency properties should also be taken into consideration. Data for sound attenuators, throttle dampers, insertion loss from devices, etc. are stated in octave bands down to 63 Hz, which in most cases is sufficient to assess the noise generation and propagation in dB(C) from the fan and other components. Inherent sound generation from the devices is normally only stated in dB(A) as the device's inherent flow noise is usually dominated by mid and high frequencies and consequently does not contribute specifically to the C-value in the room. Background values in octave bands are available as corrections from the given dB(A) value, so for those who want to, it is always possible to calculate the C-value.

Energy requirements

Increasingly stringent requirements are made on energy efficiency, partly because of the resource savings and partly due to efforts to reduce climate impact. An energy efficient climate system usually means a slightly higher initial investment, but it pays for itself as a rule quickly through lower operating costs.

Specific energy consumption

For Sweden, the Swedish National Board of Housing, Building and Planning – BOVERKET's Building Regulations, BBR 19 (BFS 2011:26) apply from 2012.

The regulations state the maximum specific energy consumption and with that concerns all energy for ventilation, air conditioning and hot water, but not energy consumption for activities in the building. Dwellings have slightly more generous requirements compared to commercial premises.

It should be noted that at the same time as energy requirements, requirements on indoor climate, the building's usability and condition must always be met.

In the winter Sweden experiences large differences in regional temperature levels and requirements are therefore divided into three climatic zones according to the following table:

The building's specific energy consumption, kWh / (m^2· year)

	Residences	Premises
Climate zone I (northern Sweden)	130	120
Climate zone II (central Sweden)	110	100
Climate zone III (southern Sweden)	90	80

Similar demands are also made on the design of the building with the highest permissible average coefficient of thermal transmittance.

Optional requirement levels

In addition to BOVERKET's Building Regulations there are various optional requirement levels with stricter requirements. For example, the organization BELOK, an association of Sweden's 16 largest property owners, has stricter requirements than BOVERKET's Building Regulations.

In addition there are special requirement levels for e.g. Passive houses, Green Buildings, etc.

Specific fan power

The specific fan power (SFP), states how much power a fan requires to transport one m^3 of air per second and is stated in the unit kW/(m^3/s).

By placing demands on the value you can affect the electrical output requirement of a fan, an air handling unit or an entire building.

Important benefits are that both the selected equipment and flow system solutions are included in the valuation, and that value can be verified.

The disadvantages are for the valuation of alternative system solutions (constant flow versus variable flow) and for heat recovery, where you can reduce the use of energy for heating and/or cooling by choosing an energy recovery system with a higher thermal efficiency, but also with a higher pressure drop, which results in a higher SFP value. Note that in the cooling instance, the total use of electricity decreases for increased SFP as the electrical power to the cooling compressor decreases in the event of recovery. A holistic approach is sometimes missing for electricity usage to service systems in a building. In order to ensure proper decision making data and to avoid sub-optimization, a careful analysis is recommended where available opportunities are calculated and evaluated.

BOVERKET's Building Regulations recommend that the specific fan power for a ventilation unit or ventilation system must not exceed SFP 2.0.

The lifecycle of a building

General
The life cycle cost (LCC) is the total cost of a certain piece of equipment throughout its lifetime, from installation to finally being taken out of service. The most important components in a LCC calculation are:

- Energy costs during the life of the product.
- Investment costs for the product.
- Maintenance costs for the product during its lifetime

It is important when purchasing energy-consuming products, to not only look at which product is cheapest at the time, but also the product with the lowest energy costs and which is the least expensive to maintain. Energy costs during the life of the product are almost always more important in the overall cost than the investment cost.

Project planning
In order to achieve an optimal end result requires a holistic and system approach. A clear picture of responsibility should be sought when choosing a supplier. All calculations must include the investment cost as well as energy usage and the maintenance cost. The supplier must be able to provide guidance with the help of key figures, communication interface, energy calculations, etc.

Procurement
The supplier must be able to submit a tender with clear specifications, demarcation lists and provide all the necessary documentation. The tender must be possible to submit on a product level or system level, including all the required products. This will ensure the desired functionality.

Construction and commissioning
Generally speaking, the system's products are installed and commissioned in accordance with the supplier's instructions. The system should be inspected twice a year, once before the cooling period in the spring and once before the heating period in the autumn.

Upgrades and rebuilding work
If the system solution is designed using products that offer a high degree of flexibility, there are many change options that do not require products to be replaced.

Air throws and air distribution patterns from air terminals can be altered as required. Via the control system, the air volume between different zones and products can be redistributed via the control system, without any physical intervention.

Most high quality products have an additional tolerance that allows a slightly higher load than the design load. All products and accessories should be available for at least 10 years after the date of purchase.

Recycling and demolition
Complete lists of used materials and proportions should be included in the products' building materials declarations.

The service life of high quality products varies between 15–25 years, depending on usage and provided that regular and professional servicing has been carried out.

Demand-controlled ventilation

In order to cope with both the energy and comfort requirements often requires what is known as demand-controlled ventilation. The concept also includes demand-controlled heating and in appropriate cases also cooling.

Demand-controlled ventilation involves ventilating and conditioning the air in a room or premises precisely to meet our needs – no more and no less. This occurs through sensors for occupancy and/or air quality and with control systems and products that adapt to the current airflow and temperature requirements.

The potential for savings is substantial, especially in premises such as offices, classrooms and hotel rooms where there is considerable variation between high and low load conditions in rooms and during times when there are few or no occupants.

Using demand-controlled ventilation, it is possible to meet the ever-increasing requirements on energy efficiency. As much as 80% fan electricity and 40% of heating and cooling energy can be saved with demand-controlled ventilation compared with constant airflow and temperature.

At the same time, it gives the opportunity for maximum comfort for the people occupying the premises through demand-controlled air, heating and cooling regulation.

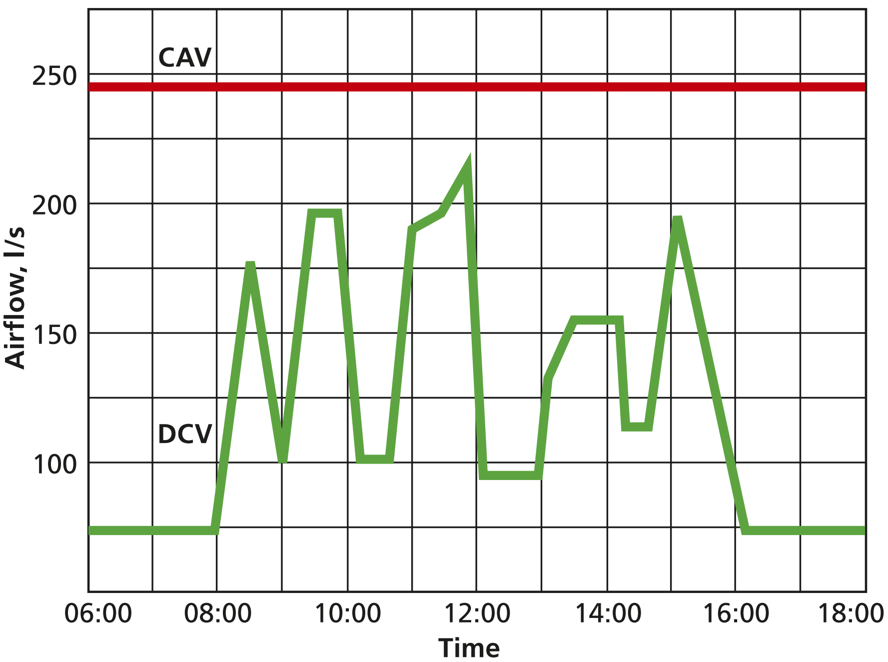
Other positive effects are that smaller units and dimensions can frequently be used for ventilation, heating and cooling as the maximum airflow and air conditioning can be reduced compared with constant flows and temperatures. Usually flexibility will also be greater if walls are moved or conversions are made in the future.

The diagram below shows an example of the difference between constant air volume (CAV) and demand controlled ventilation (DCV).

An empty room requires minimum ventilation and air-conditioning.

When the room is in use, ventilation and air conditioning is controlled to meet the needs of the occupants.

Example for an office floor



CAV

Constant airflow based on full occupancy

DCV

Demand-controlled airflow based on actual occupancy

Environment classes

In some cases, products and solutions are required that can withstand the effects of air pollutants, aggressive atmospheres, moisture and salt. Demands that can then be made are that products must meet a specific environmental class.

Environmental classes according to the Swedish National Board of Building, Planning and Housing's steel structural manual, BSK 99, are based on SS-EN-ISO 12944-2:

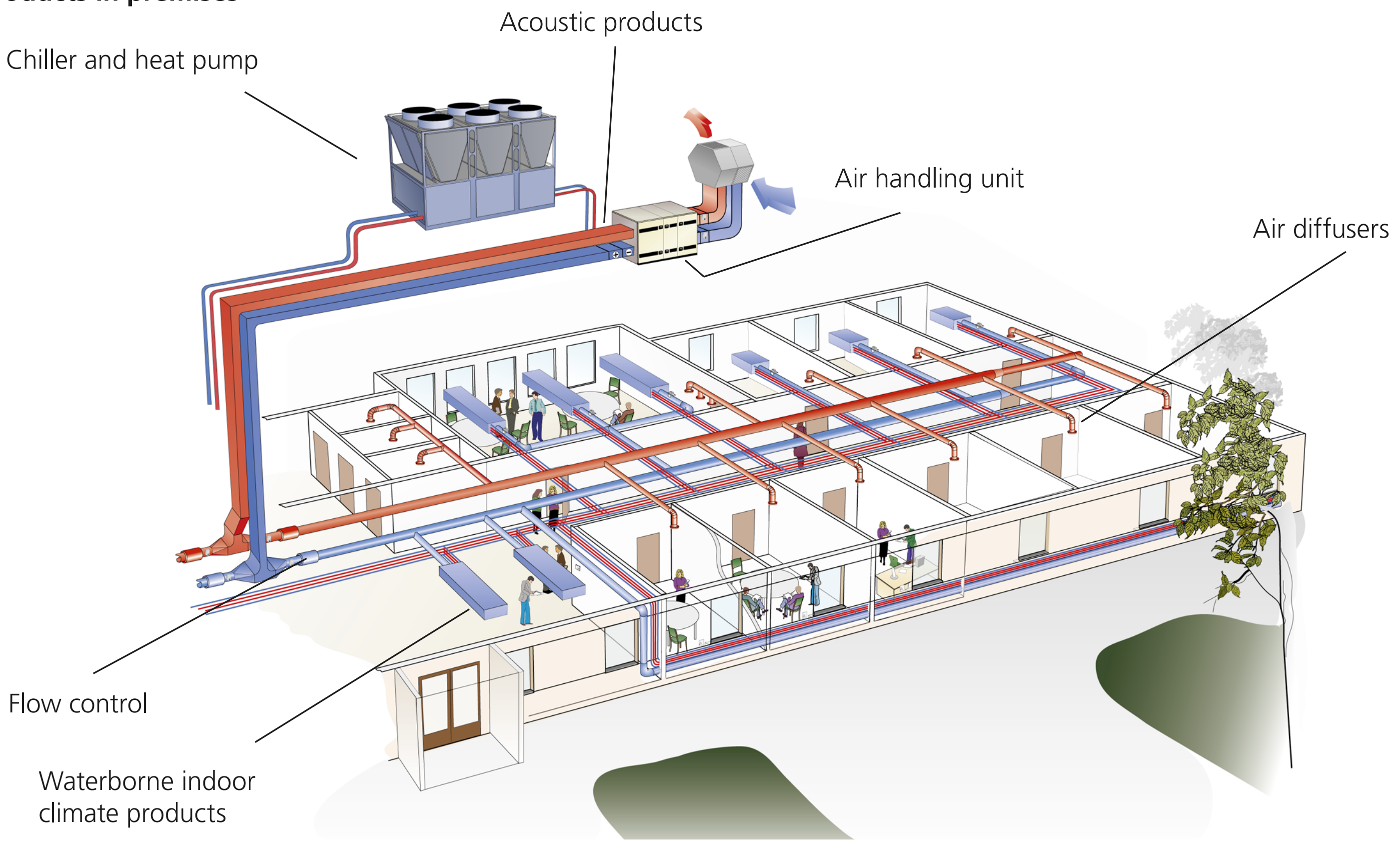
Environmental Class	Air aggressiveness	Environment example
C1	Very low	Indoors in dry air, such as in heated premises.
C2	Low	Indoors in air with varying temperatures and humidity and negligible levels of air pollution, e.g. in unheated premises. Outdoors in areas with low levels of air pollution.
C3	Moderate	Indoors at moderate effect and moderate levels of air pollution. Outdoors in areas with a specific amount of salt or moderate air pollution.
C4	High	Outdoors in air with moderate amounts of salt or substantial amounts of air pollution. Indoors in areas with high humidity and large amounts of air pollution, such as swimming pools and industrial premises
C5-I	Very high (industrial)	Indoors with almost permanent moisture condensation and high levels of air pollution. Outdoors in industrial areas with high humidity and aggressive atmosphere.
C5-M	Very high (marine)	Indoors, see above. Outside in the coastal and offshore areas with high degree of salinity.

Indoor climate products

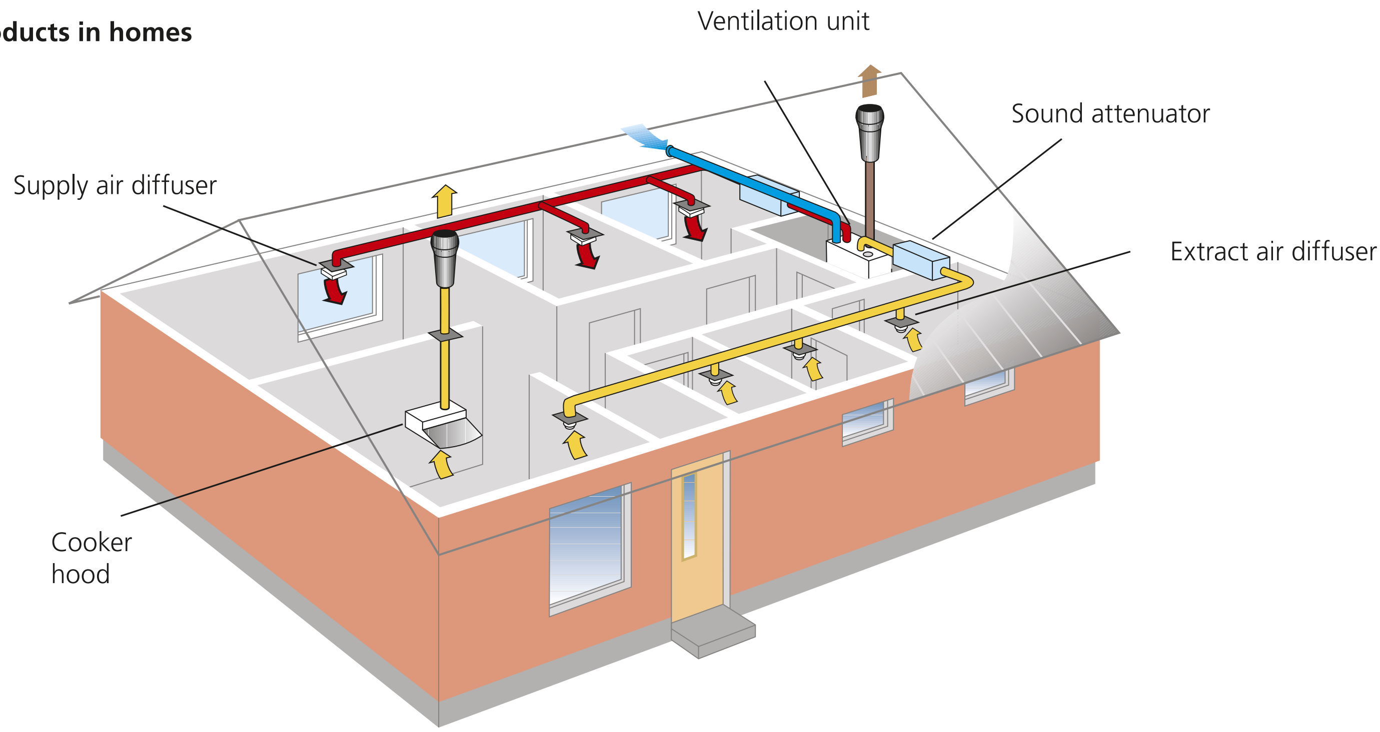
This chapter provides a brief summary presentation of the different products and product groups that can occur on different levels in an indoor climate system.

The presentation does not claim to be comprehensive. For details of products, refer to respective manufacturer's documentation.

Products in premises



Products in homes



Chillers and heat pumps

Water chiller
In a water chiller it is the liquid (pure water or water with antifreeze mixture) that makes up the cooling agent. The cooling agent is led out to the user, via tanks, pumps and pipes, usually a liquid coil in the air handling unit and/or chilled beams and similar room products.

Air-cooled water chiller
The units are designed for placement outdoors and the condensers are cooled by the ambient air via fans. Some models may be designed for free cooling when the fans at low temperatures blow cool air through a heat exchanger connected to the evaporator.

Water-cooled water chiller
The units are designed for placement indoors and have an externally placed cooling agent chiller.

Units with direct expansion
This type of unit cools the air that is either led directly through the evaporator via fans, or via a secondary circuit to the evaporator.

Standalone chillers with direct expansion are usually used in computer halls and equipment rooms. Another type of chiller with direct expansion is docked or integrated in the air handling unit.

Heat pumps
Heat pumps work in a similar way to a chiller with the difference that heat is emitted by the condenser and coolant cooling occurs in the evaporator.

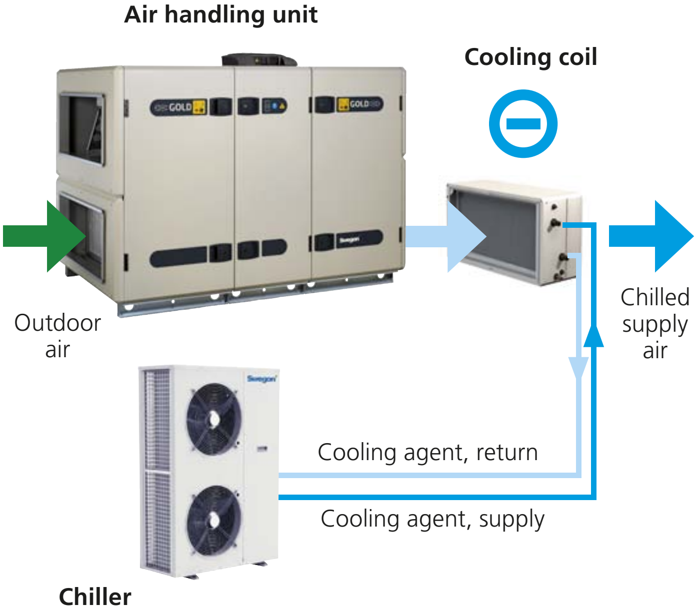
Reversible unit
Chillers and heat pumps are often reversible.

A reversible chiller is optimized for cooling, but can also be used in reverse mode to heat by switching the valves so that the cooling agent flows in the reverse direction.

Conversely, reversible heat pumps are optimized for heating, but can be used in reverse mode to cool.

Multifunctional unit
Multifunctional units are reversible heat pumps that can also produce domestic hot water.

They have a circuit for each function and can thus maintain cooling agent, heating agent and hot water simultaneously.



A typical application example within comfort ventilation is that the chiller supplies the air handling unit's cooling coil with chilled liquid.

Typical smaller air-cooled water chiller

Typically, large air-cooled water chillers

Water-cooled water chiller

Air-cooled water chillers with free cooling

Chillers with direct expansion for airborne cooling of the premises

Chillers with direct expansion for docking to the ventilation unit

Air handling unit

General
There are several different types of air handling units. Common to all is that they should transfer filtered air to and from the premises and with recovery of the relative heating, possibly also cooling, present in the premises.

One-piece air handling unit
The one-piece air handling unit is a complete air handling unit with supply air and extract air fans, supply air and extract air filters and heat exchanger.

If supplementary functional sections such as dampers, air heaters and sound attenuators are required, these are docked to the one-piece air handling unit or installed in the ductwork.

The heat exchanger can be of the type rotary heat exchanger, plate heat exchanger or coil heat exchanger.

As you can have integrated sensors for temperature and flow in an one-piece air handling unit, it usually also features in-built control equipment.

Modular air handling units
Modular air handling units are characterised by each function, or possible combination of functions, consisting of a separate module. Thereby permitting a relatively large range of optional functions.

The units are generally supplied in modules that are assembled on site. Control equipment can be included in the delivery or be provided by an external supplier.

Supply air and extract air handling units
Supply air and extract air handling units can consist of both standard units or modular units. They are used when, for different reasons, you cannot have both the supply and extract air in the same unit, usually for reasons of space.

Control equipment can be included in the delivery or be provided by an external supplier.

Room unit
Room units are complete air handling units with integrated control equipment located in the premises to be served. They are usually used in classrooms, day nurseries, conference halls, small offices, workrooms, shops, restaurants and similar public premises.

One-piece air handling units with rotary heat exchanger

Modular air handling units

Supply air unit

Room unit

Heat recovery in air handling units

Air handling units can be equipped with different types of heat exchangers. In many cases, one and the same unit can be chosen with different types of heat exchanger.

The three most common types of heat exchangers are presented below.

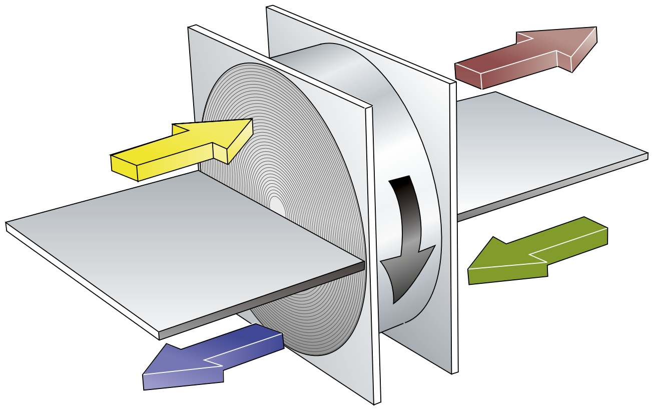
Rotary heat exchanger

The rotary heat exchanger consists of a rotating wheel with a multitude of small passages, 'ducts', made of aluminium. The warmer extract air heats the ducts and transfers heat to the colder supply air.

Its temperature efficiency may be as high as 85% if the supply air and extract air flow at the same rate. Normally, frost never forms inside a rotary heat exchanger. As a result, it can keep its high temperature efficiency regardless of the outdoor temperature.

The rotary heat exchanger recovers cooling energy just as efficiently and with treatment for hygroscopicity or sorption also recycles moisture, which saves the cost of cooling.

A rotary heat exchanger should not be used if the same air handling unit serves mixed activities, for example an office and restaurant, as the same surfaces come into contact with both extract and supply air.



Rotary heat exchanger

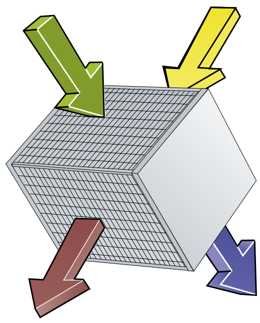
Plate heat exchanger

A cross-flow plate heat exchanger consists of thin aluminium plates that form air passages arranged at right angles to one another. The warmer extract air heats the plates and transfers heat to the colder supply air.

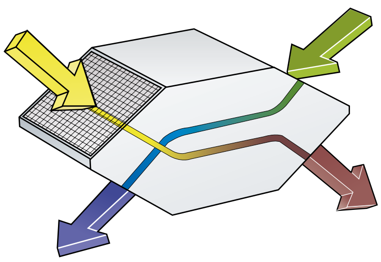
A counter-flow plate heat exchanger is designed in the same way as a cross-flow, however, it is constructed so that there is also a parallel section. This makes the contact area larger than a cross-flow plate heat exchanger.

Its temperature efficiency is as high as 65% in a cross-flow plate heat exchanger and up to 80% in a counter-flow plate heat exchanger, if the supply air and extract air flow at the same rate.

The supply air and extract air have completely separate air passages therefore possible odours or particles in the extract air cannot be transferred to the supply air.



Cross-flow plate heat exchanger

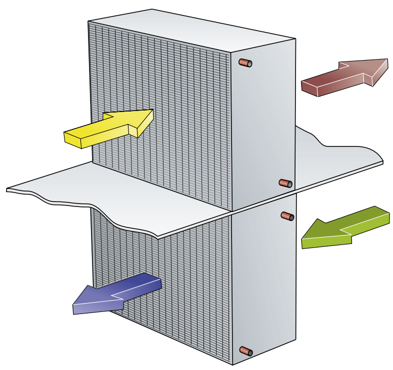


Counter-flow plate heat exchanger

Coil heat exchangers

Coil heat exchangers are generally used when you want separate air passages for high air flows. It has one liquid coil in the supply air and one in the extract air. The liquid in the extract air coil is heated by the extract air and is pumped to supply air coil which heats the supply air.

Its temperature efficiency is as high as 60% if the supply air and extract air flow at the same rate.



Coil heat exchangers

Outdoor air | Supply air | Extract air | Exhaust air

Air diffusers

General
Air diffusers are available in several different designs. The function differs between different main groups, but as it is a room product there are usually several designs available within each main group depending on the installation method and design.

Below follows a summary of the most common main groups of air diffusers.

Nozzle diffuser
These air terminals have a large number of nozzles, usually made of plastic, which the air passes through. This creates a large number of air jets, which promotes a good and rapid air mixture and jet diffusers are particularly suitable for the delivery of cooled air. The distribution pattern can be controlled for optimum comfort by rotating each individual nozzle.

Guide-vane diffuser
These air diffusers have some form of guide vanes, usually through the terminal openings being folded up at the rear of the diffuser. This creates vanes for the air, which is then distributed to each opening. Guide-vane diffusers are usually perforated with elongated openings or with some kind of hole pattern. Guide-vane diffusers withstand low temperatures and offer a high air flow capacity.

Perforated air diffuser
Perforated air diffusers have some form of perforated holes or grilles that the air passes through. These air diffusers have an impermeable middle section to direct the air towards the outer sides of the terminals. They are suitable for large airflows.

Slot diffusers
Slot diffusers or conical diffusers often consists of a type of disc, which distributes the air to the slots on the outer sides of the diffuser. The air diffuser type gives a high co-ejection with the room air.

Grille diffuser

The grille diffuser is made up of some form of grille. Grille diffusers can be designed for a fixed distribution pattern, but can also be equipped with adjustment options by turning the grille's fins.

Duct diffuser

The duct diffuser can consist of grilles or nozzles mounted directly on the visible air ducts in the premises.

Jet diffuser

Jet diffusers are used for large airflows and often in industrial premises, stadiums and the like. The jet diffusers are particularly suitable for delivery of heated air.

Displacement diffusers

Displacement air terminals are designed for cooled air supplied at low air velocities. Usually it is possible to direct the air flow more or less sideways.

Extract air diffuser

Extract air diffusers are usually simpler in design than supply air diffusers as draughts, distribution patterns and the direction of airflow do not have to be taken into account. They are usually designed with some form of disc or grille.

Air transfer grille

Air transfer grilles are usually designed with some form of sound attenuator to reduce crosstalk between rooms.

Waterborne indoor climate products

Passive chilled beams

Passive chilled beams emit the cooling capacity primarily through the room air self-circulating through the chilled beam's water coil. The warm room air is "drawn" into the top of the chilled beam, through the cooling coil and flows downward.


Passive chilled beams for surface mounting

Chilled/climate beams

Active chilled beams are a combination between air terminal and cooling unit. They provide the room with the designed level of primary air. Room air also circulates though the integrated flange coils by means of induction and in doing so cools the room air. Different flow patterns can be created depending on the form of the room, the placement of the heat load and how the chilled beams are placed.

This type of beam can be used for both cooling and heating of premises which is why many manufacturers call them climate beams.


Climate beam for flush mounting in suspended cassette ceilings.

Comfort modules

Comfort modules are a hybrid between a traditional chilled beam, an air diffuser and a radiator and combine the following properties:

- The high cooling capacity of the chilled beam at low airflows.
- The rapid mixing of supply air with room air offered by the nozzle air diffuser
- The radiator's heating capacity

Spreading the supplied air in four directions creates a large mixing zone. This means that you can supply high cooling capacities with a product, which requires significantly less space in the ceiling than a traditional climate beam.


Comfort modules for flush mounting in suspended cassette ceilings.

Induction units

Wall placement

Induction units are usually installed along the inside of exterior walls and are therefore also called facade or window units. In principle, they work as climate beams where the primary air creates both mixing and circulating room air through induction. They ventilate, cool and heat.

It is usually possible to choose different casings according to requirements and appearance and are often combined with space for electrical and data cables.

Floor placement

Induction units can also be placed horizontally and lowered into raised floors.

Induction units for installation along the facade

Ceiling placement

A modified type of induction units can also be used for ceiling placement. Installation examples are hotel and ward rooms, where the unit is placed in the ceiling angle between the hall and room.

Acoustic products

General
Acoustic products for comfort ventilation consists mainly of unit attenuators, duct sound attenuators, and sound attenuating exterior wall grilles.

Modern attenuators often feature materials on the inside that can be cleaned. They can be designed to be opened for access for cleaning or that the duct system before and after the attenuator provides access for cleaning.

They can also be designed to support different fire-resistance grades.

As a rule, manufacturers strive towards the damping capacity being as high as possible, but that air resistance - and hence the pressure drop - shall be as low as possible.

Example of unit attenuators

Unit attenuators
The unit attenuators are generally placed as close to the air handling unit as possible. They are often docked directly to the air handling unit or installed in the immediate vicinity of the unit.

The unit attenuators usually have sound absorbing materials on the insides and some form of beams with sound absorbing materials in the middle section.

They are usually available in both rectangular and circular designs.

Example of duct sound attenuators

Duct sound attenuators
The duct sound attenuators are designed for installation in the duct system. They usually have sound absorbing material along all interior sides and are usually designed to take up as little space as possible.

They are usually available in both rectangular and circular designs.

Sound attenuating exterior wall grilles
Sound attenuating external wall grilles can be used as intake or exhaust grilles for fan and machine rooms. The sound is attenuated on passing the sound absorbing fins in the grille.

Example of sound attenuating exterior wall grilles.

Flow control products

General
The basic idea with products for flow control is to ventilate and condition air precisely to meet our needs – no more and no less. Demand-controlled ventilation involves optimising the amount of power consumed to operate the fans in the air handling unit and minimising the costs for heating or cooling the building.

Active damper at zone level
The active zone damper is used to manage flow changes faster in large systems. The motorised damper controls the airflow according to a signal from the pressure sensor to maintain a constant duct pressure in the zone. The duct pressure can be kept low in the zone, which gives a quiet ventilation system and economical operation.

Active damper at room level
Active dampers are available in several different designs. A common feature is a motorised damper to regulate airflow. The active damper at room level usually regulates the airflow according to a signal from sensor for either the occupancy or air quality.

Active air terminal at room level
In principle the active air terminal works as an active damper at room level. Air flow regulation occurs through a built-in motorised damper or by motorised slot opening on the terminal.

Active comfort modules at room level
Active comfort modules regulate both the airflow and temperature according to the requirement. Airflow regulation occurs through a built-in motorised damper according to a signal from sensor for either the occupancy or air quality. The temperature is regulated by a motorised valve actuator, which increases or decreases the fluid flow according to a signal from the temperature sensor.

Control
Products for flow control are as a rule always controlled from a central system where you get an overview of all the products in the installation and where you can also configure settings on a product level. The control system communicates with the air handling unit and with a main control system.

Active damper at zone level

Active damper at room level

Active air terminal

Active comfort module

Products for residential ventilation

General

Products for residential ventilation are designed for small airflows. The ventilation units are compact to take up little space and can often be installed with wall or ceiling brackets.

They are usually fitted with integrated control equipment. Airflows can, in the simplest form, be regulated by a number of fixed airflows by switching cables on a transformer. On quality products, airflows van be variably adjusted, and other functions can be selected and set via some type of terminal.

Ventilation unit with rotary heat exchanger

Ventilation units

Ventilation units are complete with supply air and extract air fans, supply air and extract air filters and some form of control equipment.

The most common heat exchangers are rotary heat exchangers, counter-flow plate heat exchangers and the cross-flow plate heat exchangers. Also see the previous section "Air handling units".

Ventilation unit with counterflow plate heat exchangers

Cooker hoods

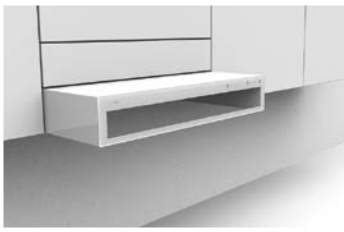
The cooker hoods can be connected via a specific bypass duct to the ventilation unit. The exhaust air from the cooker hood does not pass the heat exchanger. Therefore the cooker hood does not require an own extract air duct or fan. In such instances, the air handling unit is controlled from the cooker hood control panel.

The cooker hoods can also have its own built-in fan or be connected to a ceiling fan.

Cooker hoods are in available in a variety of designs and variations. From previously serving as a functional kitchen appliance, they have developed into a design product to give the kitchen the desired character.

Cabinet integration

The classic hood is mounted under a kitchen cabinet that is positioned over the cooker. These cooker hoods are often equipped with a hood, which give very good odour extraction. Cooker hoods for cabinet integration are also available in a version that can be pushed under the kitchen cabinet when not in use.



Classic cooker hood for cabinet integration

Flush mounted

Recessed cooker hoods are mounted above the cooker and between other kitchen cabinets. The front is equipped with standard kitchen cabinet door. When in use, the cover is lifted out and forms a hood that provides excellent odour extraction.

Wall mounted or suspended

Wall mounted or suspended cooker hoods are often called designer hoods. They are usually made of stainless steel. They are equipped with some type of screen or expansion to provide good odour extraction.

Recess mounted cooker hood

Designer cooker hood for installation against the wall or suspended above

Air diffusers

Air diffusers for homes are designed for small airflows. It is usually possible to adjust the airflow and in some cases also the direction of airflow.

Indoor climate systems

Ventilation principles

General

A lot of people have probably at sometime complained about the ventilation at the workplace, in a lecture hall or in any other large building. It is too hot or too cold, blows too strongly, the air does not feel fresh, etc. It is difficult initially to install the perfect ventilation system in a given local, as there are no rules for all common installation cases.

Full scale tests in a laboratory, where e.g. an office is built with the intended ventilation system, is the safest way to examine how a planned system will work in reality. The laboratory for full scale testing gives designers and developers the opportunity to try a hypothetical ventilation system early in the building process.

A distinction is made between two main principles for the ventilation of premises:

- Mixing ventilation
- Thermally controlled ventilation

We can also mention:

- Piston flow
- Short-circuit flow

The principles are presented in detail on the following pages of this chapter.

The chapter also presents practical guidelines for the principal supply air system normally used in premises in connection with comfort ventilation.

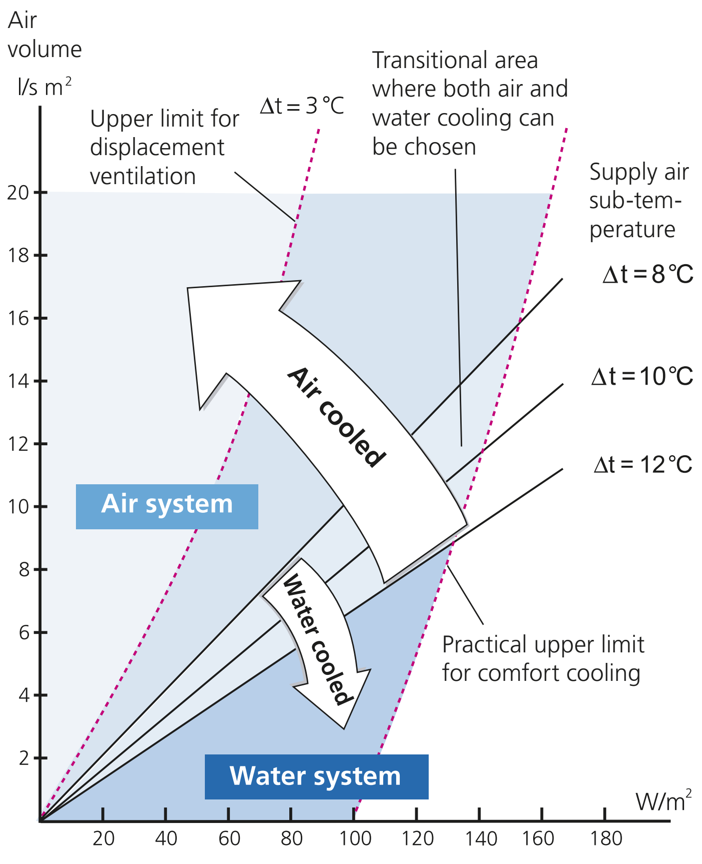
Climate systems air/water

Distinguishing characteristics for different climate systems

Air is used in all system solutions, as only this medium can create good air quality. Air can also be used to manage the temperature, if a sufficiently large flow is used.

Water is a good energy carrier which at comparable flow rates and volumes with air, is capable of transporting significantly larger amounts of energy. The diagram below shows when cooling with air or cooling with water is appropriate.

Accordingly, when selecting a system air is always used for the ventilation requirement. When it comes to temperature, both air and water can come into question. Which medium is best to transport the energy depends on a large number of factors. The building's design, operations, media access, as well as the client's special requests are examples of causes that can govern system selection in one way or another.



With knowledge of the specific cooling capacity load and the specific airflow requirement, the diagram shows whether cooling can be done with air or whether supplementing with water should be done.

Cooling capacity

The subsequent section, Air conduction in premises, includes more information about different principles' possibilities to cool premises.

The given limits are fluid and highly dependent on, among others, the extent of the occupied zone. When specifying comfort requirements, it is important to make a connection to where in the room the requirements must be met. Since the requirements can change over time as the type of activity changes, it is important to be able to influence the distribution of air in the room in a simple way. This is done by selecting supply air diffusers with a flexible adjustable distribution pattern.

Stated cooling capacities for the different ventilation principles only apply in the instances when air is the energy carrier. If water is used as the energy carrier, i.e. the use of chilled ceiling of a radiation ceiling type or chilled beams for convective cooling of the air, a significantly larger heat load can be removed.

The figure gives an indication of when it is necessary to select either an airborne or waterborne system. Based on the requisite airflow (l/s, m²) for different types of premises, it is then easy with knowledge of the required cooling load, to determine the most appropriate option.

Three types of climate system for comfort cooling

The excess heat that must be removed from the building to maintain the indoor temperature below a predetermined maximum permitted temperature, called the cooling load. The climate systems used to actively cool the buildings, can generally be divided into three types.

- Systems with airborne cooling
- Systems with waterborne cooling
- Combined systems

It is important to distinguish the sensible cooling capacity load and the total cooling capacity including wet cooling. Sensible cooling capacity refers to the capacity that corresponds to the temperature difference between the desired temperature and the temperature you would have without comfort cooling. The total cooling capacity shall also include the latent cooling load that include wet cooling. This concerns the enthalpy difference that must be accomplished in order for the supply air to be dehumidified in the ventilation unit's and/or fanned air cooler's cooling coil, see the figure. If one include the latent cooling load, the total design cooling load generally increases by over 100%.

Other factors that affect system selection:

- Customer requests
- Access to water and air
- Number of persons in the room/house
- Type of activity
- How the building is built

Units	Water	Air
Density d kg/m^3	1000	1.2
Energy c_p kJ/kg C	4.18	1.0

Rule of thumb:

P_{tot} = W/m^2

q_{air} = l/s m^2

If P_{tot}/q_{air} >15 — use a waterborne system

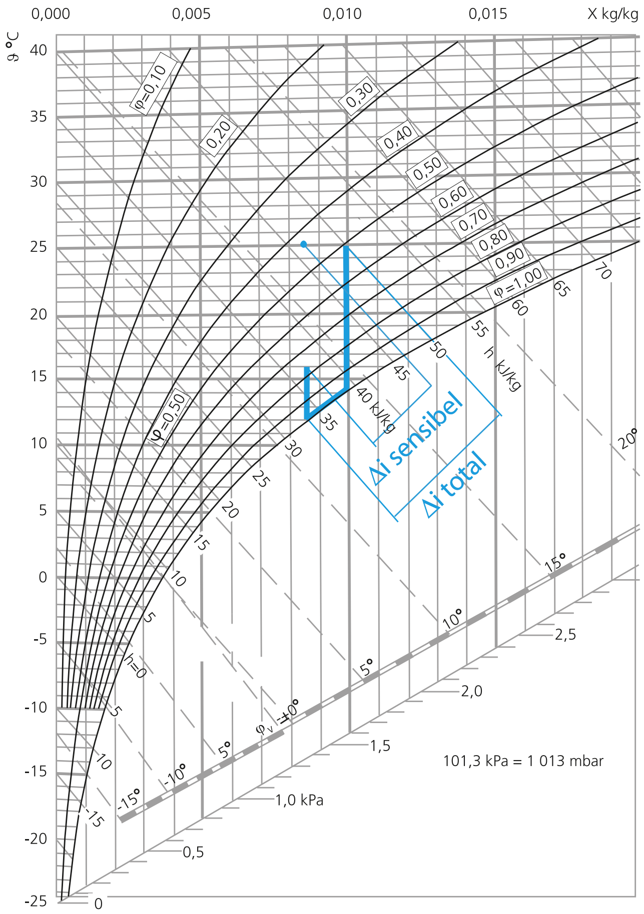
If P_{tot}/q_{air} >10 — use a airborne system

If P_{tot}/q_{air} =10-15 — use a airborne system

Example

500 Watt cooling capacity
water 0.04 l/s, 12 mm pipe
air 42 l/s, 160 mm duct

- Water — Takes up less space
  Smaller unit
  Improved energy carrier
- Air — Simple, inexpensive installation
  Larger volumes of air (better quality)
  Free cooling can be be used



The sum of the latent and sensible cooling capacity gives the design cooling capacity.

Systems with airborne cooling

In these systems the design air flow is determined by the cooling load. Accordingly, it is not the air quality requirements that determine the size.

In existing buildings, it is usually difficult and expensive to change the duct system. If you cannot transport sufficiently large air flows in the existing ducts to meet the cooling load, a waterborne cooling system is usually installed during rebuilding. The cooling system must be able to take care of the variation in cooling load, both during the day and over the year. The two basic types of systems with airborne cooling are constant flow systems or variable flow systems (combinations of the two methods are also available).

Typical application areas with air where high to medium airflows and maybe even variable air volumes come into question:

- Conference rooms
- Light industry
- Shops
- Clean rooms: hospitals
- Classrooms
- Restaurants
- Shopping centres
- Sports halls

Systems with waterborne cooling

These types of system provide individual rooms with waterborne cooling. The existing air system is solely used to meet the air quality requirements.

In a rebuilding or renovation situation this type of cooling system is often preferred. When installing the system, there is usually space in existing suspended ceilings to install the pipes required for the distribution of chilled water in the building.

Systems with waterborne cooling may also be preferable in new builds, since the space requirement above the ceiling is less than with airborne cooling. In tall buildings, in some cases, it can even mean that a certain building height can "serve" extra floor levels.

Typical uses of waterborne cooling where low to medium airflows with a high cooling capacity come into question:

- Offices
- Conference rooms
- IT rooms
- Shops
- Hotels
- Classrooms
- Banks
- Restaurants

Cooling system

Chillers in comfort ventilation are generally used to cool the supply air delivered by the air handling unit. The chiller can also supply cooling directly to room products, for example, comfort modules.

In this section the following methods are described to generate cooling:

- Conventional electrically powered compressor cooling is the traditional way to produce cooling. It also offers great flexibility.
- Free cooling can be used in waterborne systems and require some type of heat exchanger to the outdoor air.
- District cooling is being offered to a greater number of individual properties. Cooling water is supplied to a substation and secondary water is then distributed to the building to be cooled. The production units can range from cold seawater to heat driven chillers.

Two less common methods to cool air are also described briefly:

- Evaporative cooling lowers the air temperature as it passes a wet surface that creates water evaporation. There is a cooling capacity as long as the air is not saturated with water vapour.
- Sorption cooling is in principle the same as in evaporative cooling, but the supply air is dehumidified before it is moistened again.

Electrically driven compressor cooling

General
Cooling effect with a compressor cooling unit is the classic way to produce cooling. When machine cooling for comfort purposes is discussed, this is what is normally referred to.

A compressor-driven chiller offers great flexibility in terms of the method of supplying cooling to the building. It is possible to deliver cooling from the chiller to either the cooling coil in an air handling unit and/or cooling equipment placed directly in the rooms, such as chilled beams.

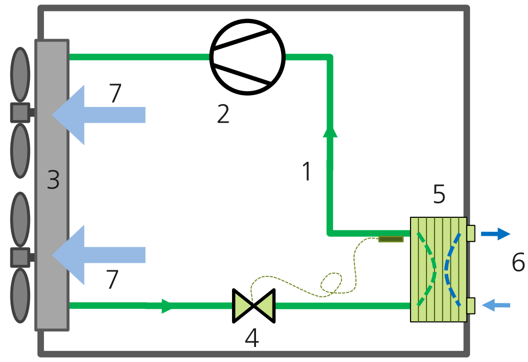
Definitions
Chiller: Unit that produce cooling.

Refrigerant: The energy carrier which, via a compressor, circulates between the evaporator, expansion valve and condenser in the cooling unit. Usually some type of HFC (see below).

Cooling agent: Usually water or water with antifreeze mixture which has been cooled in the chiller and carries cold to the user.

Heating agent: Usually water or water with antifreeze mixture which has been heated in the heat pump and carries heat to the user.

User: Liquid coil on an air handling unit, chilled beam or the like that receives the cooling agent.



Principle for air-cooled water chiller
1) Refrigerant
2) Compressor
3) Condenser with fans
4) Expansion valve
5) Evaporator (heat exchanger)
6) Cooling agent (water)
7) Ambient air

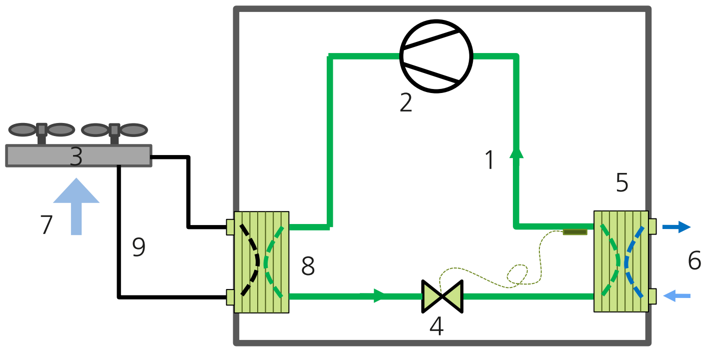
Principle of operation for all cooling units
The cooling unit has a circuit filled with refrigerant, usually some type of HFC (HydroFluoroCarbons). The compressor increases the pressure of the refrigerant, which is then heated and transforms into gas state. When the refrigerant passes the condenser it is cooled down and condenses, i.e. returns to a liquid state. It then passes through an expansion valve which quickly lowers the pressure, which causes the refrigerant to cool. The refrigerant then passes through the evaporator. This acts as a heat exchanger where the refrigerant transfers its coldness to the cooling agent.

Air-cooled water chiller
The cooling unit is placed outdoors and the condenser is cooled by ambient air.

Water-cooled water chiller
The cooling unit is placed indoors. A condenser made up of a heat exchanger via a fluid circuit from an outdoor dry cooler is used to condense the refrigerant.

Heat pumps
The same basic principles apply as for the cooling unit, but the condenser is used instead to transfer heat to the heating medium.

Multifunctional unit
These units can heat, cool and produce domestic hot water. They consist of a reversible heat pump with several circuits to be able to simultaneously produce heat water, cooling water and hot tap water.



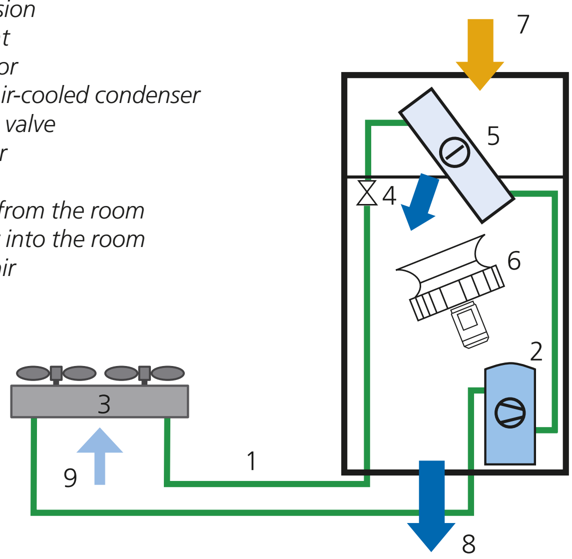
Principle for water-cooled water chiller
1) Refrigerant
2) Compressor
3) Outdoor dry coolers with fans
4) Expansion valve
5) Evaporator (heat exchanger)
6) Cooling agent (water)
7) Ambient air
8) Condenser/heat exchanger
9) Fluid circuit

Chillers with direct expansion, fan equipped

The evaporator transfers direct cooling to the through-flowing air. The air-cooled condenser is placed outdoors.

This type of chiller is usually used in computer rooms, equipment rooms, and the like.

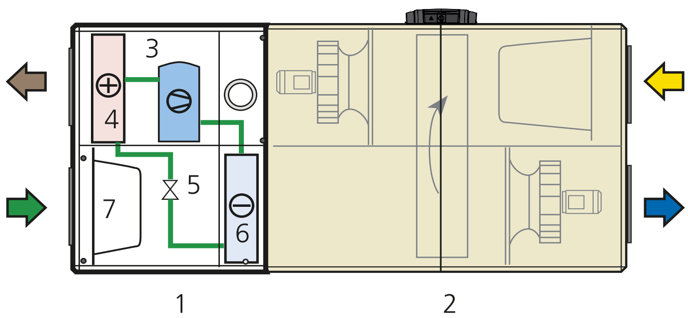
Principle for fan equipped chiller with direct expansion
1) Refrigerant
2) Compressor
3) Outdoor air-cooled condenser
4) Expansion valve
5) Evaporator
6) Fan
7) Warm air from the room
8) Cooled air into the room
9) Ambient air



Chillers with direct expansion for docking to the air handling unit

Another type of chiller with direct expansion is docked or integrated in the air handling unit. The evaporator is then placed in the unit's supply air and the condenser in the exhaust air.

One advantage of this type of unit is that only the actual docking to the air handling unit needs to be done on site. The limitation may be capacity.



Principle for chillers with direct expansion docked to the air handling unit
1) Chiller
2) Air handling unit
3) Compressor
4) Condenser
5) Expansion valve
6) Evaporator
7) Air filter

Free cooling

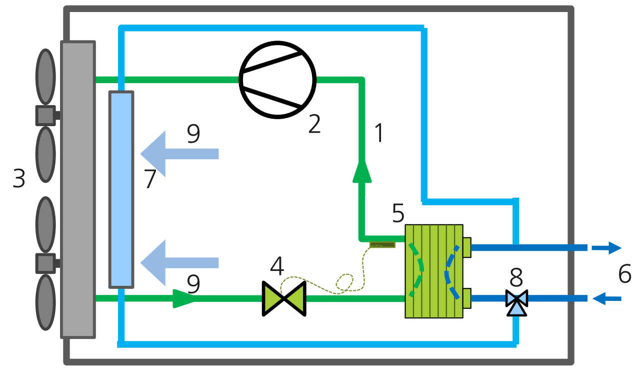
Definition

Free cooling refers to the possibility to supply cooling when there is a cooling requirement without having to pay for the source of cooling. However, there are always other costs, such as for pumps and/or fans.

Waterborne free cooling

For waterborne cooling systems there is a possibility to use so-called free cooling. The most common is that some form of heat exchanger to the outdoor air is installed. This is usually carried out integrated in the chiller and heat exchanger is connected between the chiller's refrigerant and cooling agent circuits.

When free cooling is utilised when using waterborne cooling, it is common that, at a predetermined outdoor temperature, you allow all water to be cooled to the outdoor air. Thus, at temperatures lower than this the chiller is not used. The outdoor temperature at which switching occurs normally lies around 10°C.



Principle for an air-cooled water chiller with free cooling
1) Refrigerant
2) Compressor
3) Condenser with fans
4) Expansion valve
5) Evaporator (heat exchanger)
6) Cooling agent (water)
7) Heat exchanger for cooling agent
8) Three-way valve for cooling agent return
9) Ambient air

Waterborne free cooling can also be used in connection with cooling towers, watercourses or geothermal pipes.

Free cooling without a chiller

In its simplest form, free cooling can be used without a chiller. The air handling unit's outdoor air is then used for cooling and supplies cooled air to the air diffusers in different rooms. With a heating coil in the supply air, the supply air temperature is regulated during the winter so it does not become too cold.

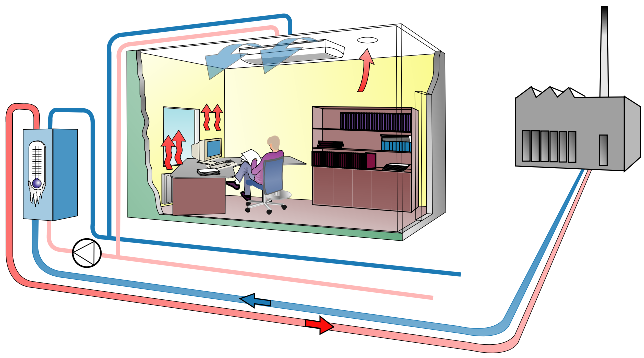
This works as long as the outside temperature does not exceed about 16°C, which makes the method useful especially in countries with a Nordic climate. In e.g. Stockholm, the outdoor temperature is normally lower than 16°C approximately 7800 hours (out of a total of 8760 hours) per year. Higher interior temperature are accepted on hot summer days or chilling is carried out in another way. Some air handling units have control functionality for so-called summer night cooling, when a higher airflow, without the heat recovery unit being operational, cools the premises during the night.

District cooling

It is becoming increasingly common for energy companies to offer their customers district cooling. Cooling is produced and distributed in different ways in different places depending on the individual energy company's prerequisites in terms of production capabilities and customer base structure and density. A district cooling system, production units can consist of anything from "free cooling"(e.g. cold sea water that can be used directly for cooling purposes), over compressor cooling machines, to heat driven chillers (absorption chillers).

It is relatively common to utilise cold from existing heat pumps that are already used to supply heat to the district heating network. Previously it was common for customers with a relatively large cooling load to be connected to a district cooling network, for example, a hospital site or a shopping centre. However, it is now more common for individual properties to be offered the chance to be connected.

For customers, cooling water is supplied to a substation, basically in the same way as in a subscriber station for district heating. From there, the secondary water is distributed to the building or buildings to be cooled.



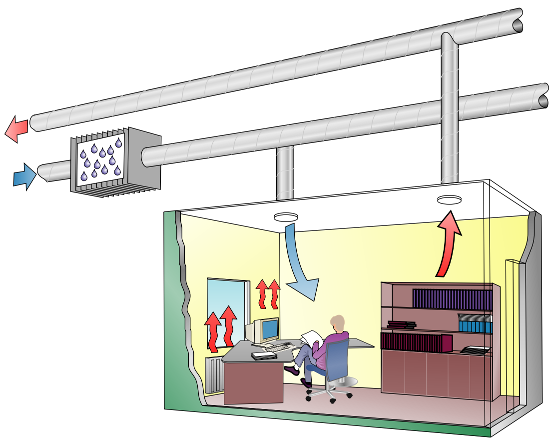
Principle for district cooling

Evaporative cooling

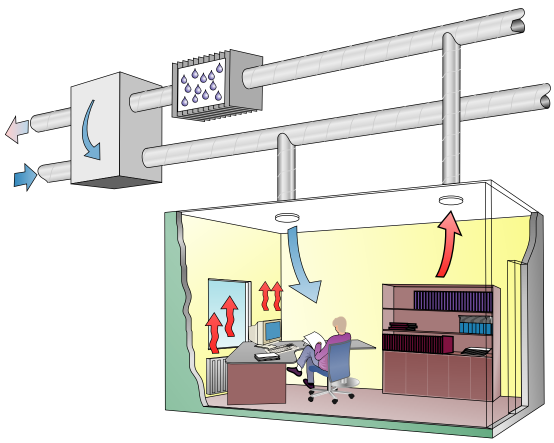
Evaporative cooling of air utilises the drop in air temperature by moistening the air using water evaporation from a wet surface that the air passes. Cooling is possible as long as the air is not saturated with water vapour. The lowest temperature the air can take on with this type of cooling is limited by the air's wet temperature, which is sometimes called the air's cooling limit.

Direct evaporative cooling refers to a process where the supply air is moistened and the temperature is lowered. At the same time the supply air's moisture content increases. With indirect evaporative cooling, the extract air is moistened, whereby the air temperature is lowered. Thereafter, a heat exchange occurs (not moisture-transfer) between the extract and supply air where the heat from the supply air can be transferred to the extract air.

The possibility to cool is basically determined by the current state of the outside air. The more moisture (the higher the value of t_{wet}) it contains the poorer its ability to cool. The method therefore is considered to be of limited use in offices and other commercial premises.



Principle of the direct evaporative cooling

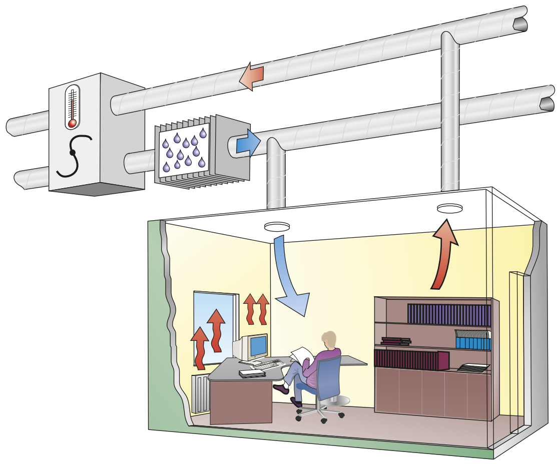


Principle of the indirect evaporative cooling

Sorption cooling

In order to lower the supply air temperature as much as possible, it is advantageous to have the driest possible air when moistening process begins. In the sorption cooling process, moistening from the evaporative process is supplemented by drying of the supply air before it is moistened.

Accordingly, a sorption chiller consists of a dehumidifier section that dries the air and a section that cools the air (the evaporative part). The supply air is dehumidified with a moisture-absorbing rotor. On the extract air side, the absorbed water is driven out of the rotor. Heat is used for this. Accordingly, heat must also be supplied to a sorption chiller.



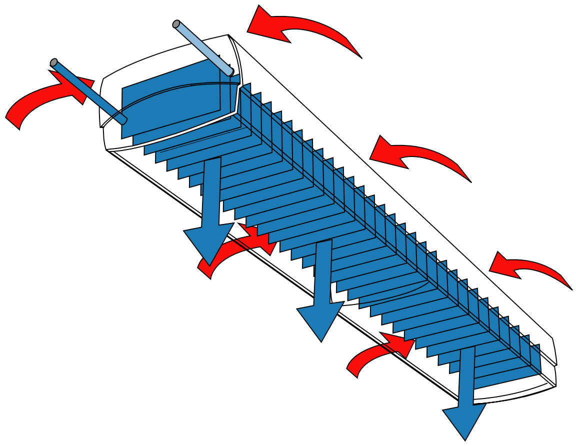
Principle of sorption cooling

Passive chilled beams

A passive chilled beam delivers its cooling effect primarily through convection, i.e. circulating room air flows through the cooling coil.

The chilled beam's water load is usually regulated on/off or variably with a control valve, sometimes several chilled beams are controlled depending on the required cooling capacity and flexibility.

Chilled beams work according to the principle of "dry" cooling (above dew point). The supply temperature of the cooling agent should always be higher than the dew point temperature of the air in the room to avoid condensation.



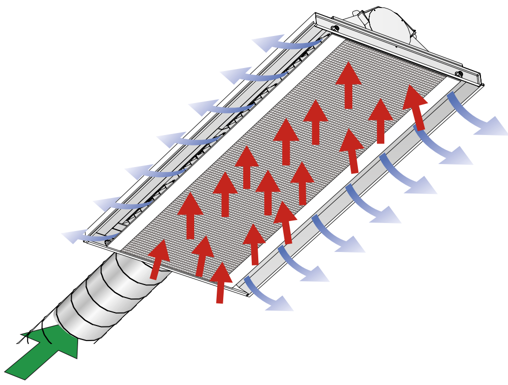
Principle of passive chilled beams

Active chilled and climate beams

A chilled beam with a connection for supply air is called an active chilled beam and can work at the same time as a supply air diffuser and in many cases increase the cooling effect through the so-called induction effect. This type of beam can be used for both cooling and heating of premises and is therefore also called for climate beam.

Distinguishing characteristics for chilled/climate beams

- Cost effective solution with supply air and room cooling in the same unit
- High cooling capacity
- It is important that the supply air is dehumidified so that condensation precipitation and reduced capacity are avoided.
- Under certain circumstances, can also be used for heating as an alternative to radiators



Principle of active chilled/climate beams

Comfort modules

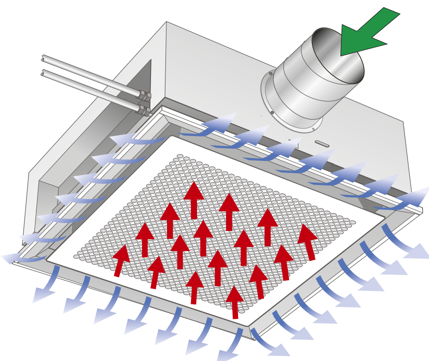
The comfort module can be described as a hybrid between a climate beam, a supply air diffuser and a radiator.

The comfort module combines the high cooling capacity of the climate beam at low primary air volumes with the supply air diffuser's ability to quickly mix the cooled air with the room air. This creates the possibility to supply large cooling capacities with a much smaller unit than with a climate beam. While the same function creates better conditions to heat the premises from the ceiling.

Some comfort modules are also equipped with the option of variable airflows based on the occupancy and energy requirement.

Distinguishing characteristics for comfort modules

- Small dimensions give greater space for other installations in the ceiling
- Four-way air distribution allows large mixing zone and high comfort
- Conditions and ventilates premises for high comfort regardless of the season (ventilation, cooling and heating)
- No moving parts, give low maintenance costs
- Great flexibility throughout its life



Principle for comfort modules

Cooling panels

A horizontal cooled surface that is suspended from the ceiling is called a cooling panel. Cold water passes through pipes connected to an aluminium plate in the cooling panel. Heat is transported from the plate to the cold water. The cooling panel then cools the hot air in the room as well as absorbing heat from the room through low temperature radiation. Cooling panels can be surface mounted on the ceiling, suspended or integrated in suspended ceilings.

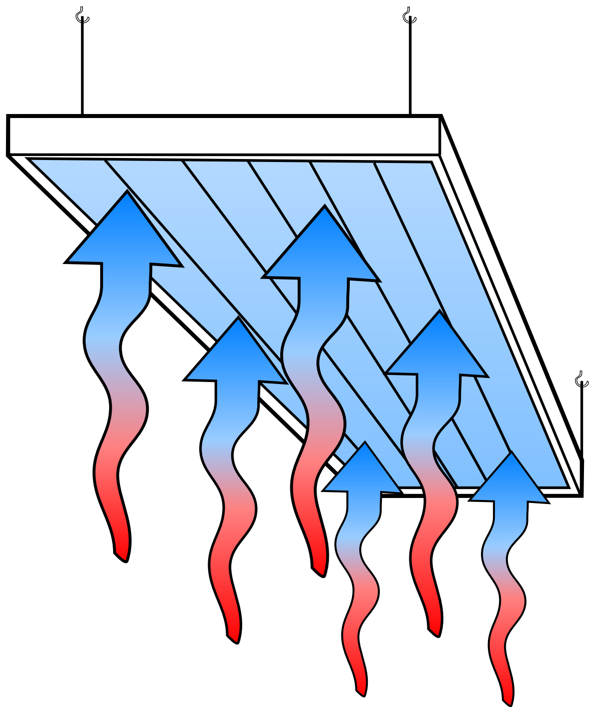
The output from the cooling panels is usually regulated on/off with a control valve, sometimes several cooling panels are controlled depending on the required cooling capacity and flexibility.

Cooling panels work according to the principle of "dry" cooling (above dew point). The supply temperature of the cooling agent should always be higher than the dew point temperature of the air in the room to avoid condensation.

Cooling panels provide good thermal comfort, but there are also some disadvantages that need to be considered. As the panels cover a large part of the ceiling, they become slightly inflexible and difficult to combine with other ceiling installations. They also constitute an acoustically hard material, which can give rise to echo.

Distinguishing characteristics for cooling panels

- Discrete solution
- Give cooling in the form of radiation
- Cooling effect limited to about 100 W/m^2

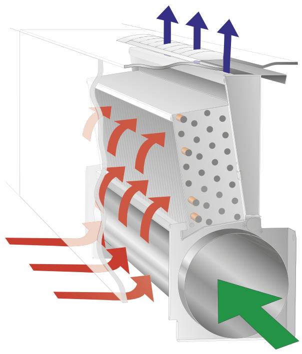


Principle for cooling panels

Induction units

A unit via which the room can either be heated or cooled.

When the induction unit is used, ventilation air is supplied to the room via the induction unit. The ventilation air flows through a nozzle at high speed, which results in the room air being "drawn along" through a combined heating and cooling coil with two separate water circuits. In this way it is possible to heat or cool the room through a single unit without another fan other than the one in the central air handling unit.



Principle for induction units

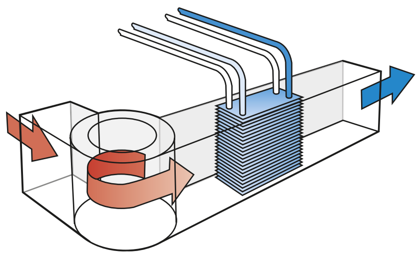
Distinguishing characteristics for induction units

- Relative high cooling capacity
- Contains functionality for supply air
- It is important that the supply air is dehumidified so that condensation precipitation and reduced capacity are avoided.
- Low operating and maintenance costs
- Requires no filter replacement
- Low sound level

Fan convectors

A unit via which the room can either be heated or cooled.

A fan convector is equipped with a fan that circulates the room air through the unit. The air is heated or cooled in the unit in a combined heating and cooling coil with two separate water circuits. Hot or cold water is fed to the coil from a central facility in the building. Fan convectors are the room chillers that has the greatest cooling capacity, but also the highest sound level, and a significant service requirement.



Principle for fan convector

Distinguishing characteristics for fan convectors

- High cooling/heating capacity
- Can manage wet cooling if a drainage system is fitted
- Relatively high noise level (at high output power)
- High operating and maintenance costs (for example, filter and fan replacement and cleaning of the drainage system)
- Requires a separate system for supply air

Ventilation systems

Traditional grouping of ventilation systems

The choice of suitable technical solution is an important step in the design process. System selection should be made with regard to the following main factors:

Efficiency
The technical solution's ability to meet the set requirements.

Operating reliability
The technical solution's ability to provide satisfactory operation over a long period

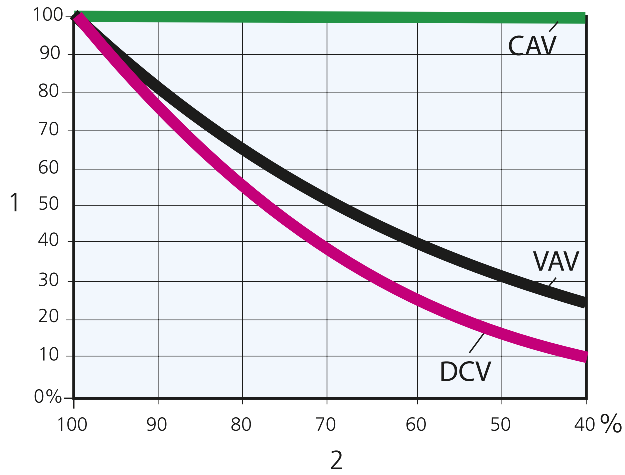
Resource efficiency
The technical solution's energy efficiency, cost effectiveness, etc.

Simplicity and tolerance
When choosing the technical solution, you should always strive for simplicity, comprehension and tolerance to fluctuations in operating conditions. Avoid technical solutions that do not permit the application area of the premises to be changed, windows to be opened, or is otherwise sensitive to external disturbances.

Basic principles and characteristics
There are different ventilation engineering solutions that can meet the requirements of right airflow to all parts of a system. The main categories spoken about are:

- CAV system (Constant Air Volume), system with a constant airflow. The simplest and generally “least expensive” option.
- VAV system (Variable Air Volume), system with variable airflow, which is usually controlled by some type of room temperature sensor. The fan is equipped with some form of pressure control.
- DCV system (Demand Controlled Ventilation), a VAV system but with DCV refers to extensive demand control of airflows and temperature, as a rule, through air quality or occupancy sensors.
- All system solutions can of course be implemented with either mixing or thermally controlled ventilation (displacement ventilation).

Both CAV and DCV systems can be combined with alternative heating and cooling units for control of the room temperature.



Relative energy use depending on type of system.
1. Relative energy use fan (%)
2. Relative airflow requirement (%)
CAV = Constant Air Volume
VAV = Variable Air Volume
DCV = Demand Controlled Ventilation

CAV system

CAV systems (Constant Air Volume) are characterised by the airflow being constant. The rooms with the greatest cooling requirement normally determine the size of the supply air temperature that is prepared in the central air handling unit. In some rooms, such as conference rooms, the supply air may need to reheat. This is done because a room must not be perceived as cold when no one has occupied it for some time.

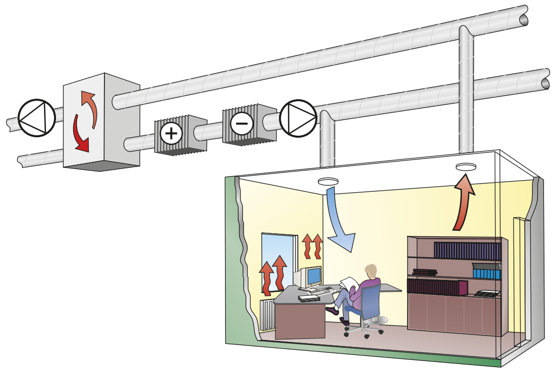
The supply air temperature in a CAV system can be constant or varied in relation to the outdoor temperature. When temperature control occurs centrally or with a constant supply air temperature a correction is made during the winter to the correct room temperature in individual rooms, for example, with radiators.

CAV systems are used where both the generation of heat and contamination are low and fairly constant. The supply air flow is primarily determined by the air quality requirements. If the hygienic air flow is insufficient remove the generated heat, you can supplement with products for waterborne cooling. The CAV systems are usually built up according to the branching principle with adjustment dampers in each branch. The pressure drop across devices is selected so that together with the pressure drop across the adjustment dampers they provide the correct flow distribution.

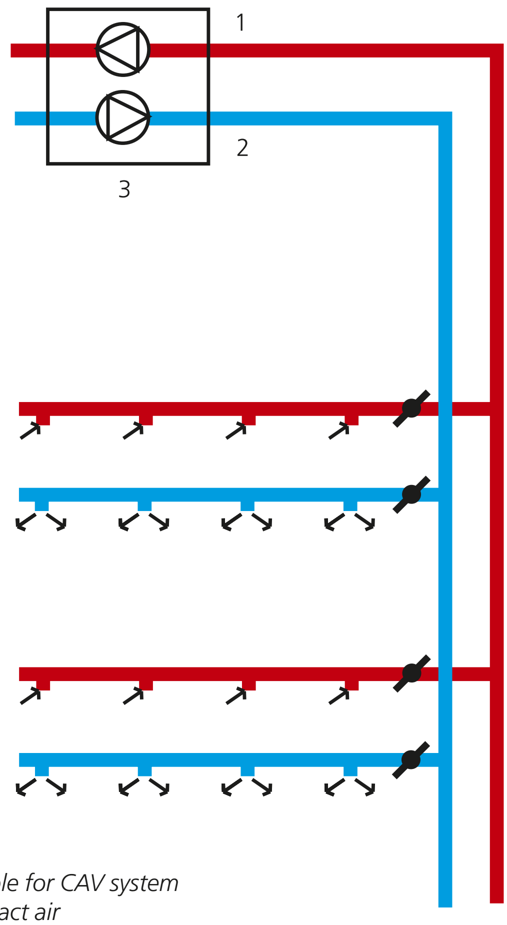
Although a CAV system supplies air at a constant flow, sometimes speed controlled fan motors are used, on which the speed is regulated down when the cooling load in the building permit. The air flow then drops in proportion to the speed.

The disadvantage of this principle is that the system can easily become unbalanced due to disruptions from thermal lifting forces, changes in damper positions, etc.

Another disadvantage is the relatively high pressure drop across dampers and terminals that is necessary to ensure that flow variations will not be too great. This in turn causes sound problems that can be troublesome while energy consumption is unnecessarily high. A lowering of the fan speed, to reduce energy usage during certain periods, means flow distribution cannot be maintained, due to the pressure drop across the terminals and dampers decreases.



Principle for CAV system



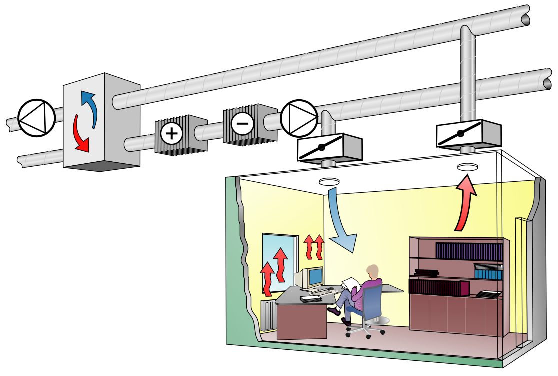
Principle for CAV system
1. Extract air
2. Supply air
3.Air handling unit

VAV system

In VAV systems (Variable Air Volume) the airflow supplied to each room varies as required, but the temperature of the supply air is kept constant, i.e., the supply air temperature does not change with a change in load. However, a seasonal control normally occurs of the supply air temperature as a function of the outside temperature.

The airflow to each room is controlled by dampers in the form of terminal devices in direct connection to the room, while the central supply and extract air fans are controlled by means of guide vane control or variable speed controlled fan motors, usually frequency controlled.

Control normally occurs by maintaining a constant static pressure with sensors in the supply air system's farthest branch ducts. The flow varies from max. on the warmest day down to about 20% of max. during the coldest days, when the air only has the task of meeting demands on air quality.



Principle for VAV system

DCV system

The DCV system is basically the same as the VAV system, but the DCV concept concerns more stringent control of the air flow and temperature.

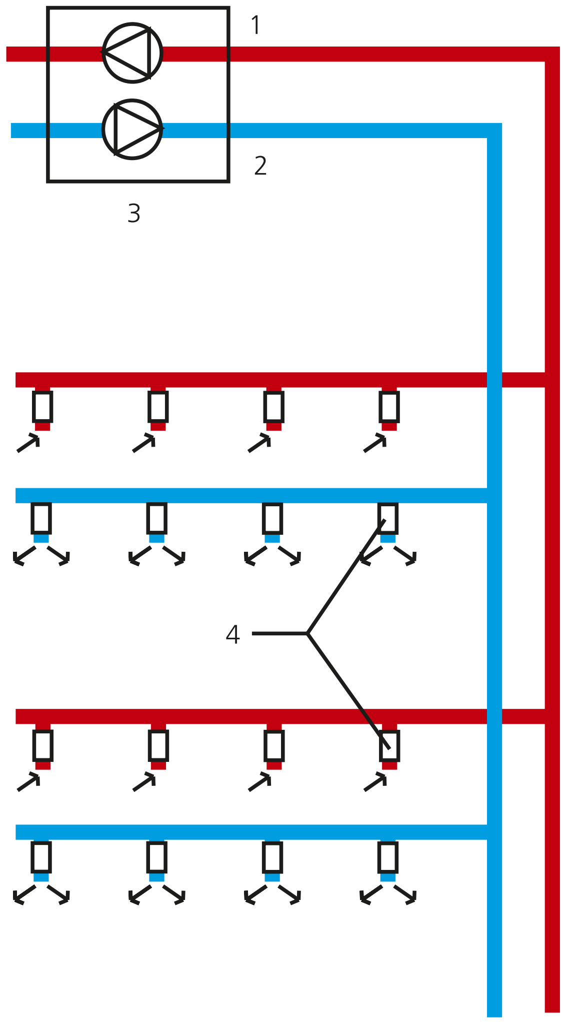
The DCV system is used when the person load varies. Heating is best performed with radiators. The room's cooling load is controlled by a varied airflow. Under certain conditions, heating from the ceiling with the help of comfort modules is also a viable alternative.

The DCV systems differ from the CAV systems as there is pressure control in the main ducts for the supply air and extract air. This is necessary from both energy and sound standpoints.

Another difference is that in the immediate vicinity of the room units (supply air diffusers, comfort modules, etc.) are control units that control the airflow through them. A fundamental problem with this is that when the flows decrease, the pressure drops. This is solved by also having controlled zone dampers.

Demand-controlled ventilation

It is generally accepted that if we, as users of an installation, can simply influence its setting, then we perceive the installation as significantly better. For example in homes, this flexibility means residents in a very simple way are able to adapt their airflows. Being able to control the airflow according to the need in different room units has previously been uncommon in traditional FTX systems. Instead it has been attempted to try to keep the airflow as constant as possible. Yet there is a clear advantage if the airflow within reasonable limits can be adjusted as required in individual rooms. This must be possible without the need to reduce airflows in other rooms. The minimum airflow rates in the different room units must also always be guaranteed.



Principle for DCV system
1. Extract air
2. Supply air
3. Air handling unit
4. DCV unit

Air conduction in premises

Mixing ventilation

Mixing ventilation can be generally used in conjunction with comfort ventilation, i.e., irrespective of whether the ventilation air is used for cooling or heating. Supply air is supplied here in such a way that the concentration of impurities is equal in all parts of the premises. Likewise, the temperature differences are very small in the building, which benefits comfort. This is the most common flow principle in our premises on account of the comfort benefits. Supply air is generally supplied at ceiling level or below the windows with a relatively high impulse. In order to prevent draughts, supply air diffusers must be chosen carefully so that the throw length, spread, etc are well balanced in relation to the size of the room.

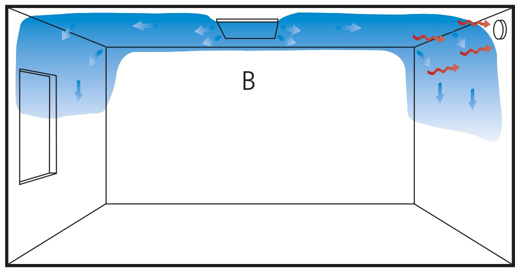
The design work must take into account the following:

- Activity level/type of premises
- Room dimensions
- Airflow, etc.
- Possible cooling requirement
- Resulting air velocity in the room
- Resulting sound level

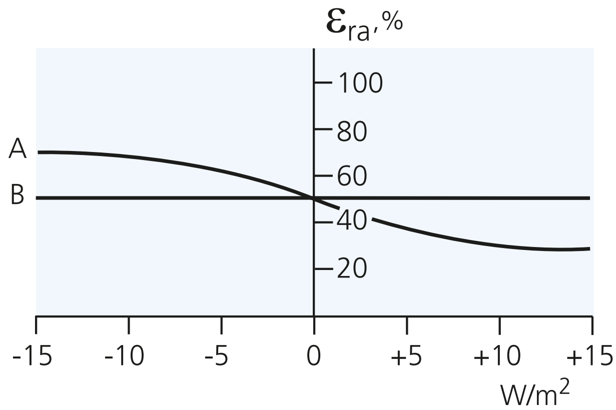
1. It is important that the level of activity can be determined in order to assess the comfort limits that shall apply.
2. Room dimensions affect the flow pattern and thus the comfort in the premises. It is therefore important that, at the design stage, to correct throw data in accordance with applicable design rules.
3. The lowest possible airflow is determined according to the hygiene requirements.
4. A calculation, where both internal and external loads such as energy accumulation in the building are taken into account, must form the basis for the calculation of the required cooling. Together with the comfort requirements this gives the basis for choosing the system solution and an appropriate supply air flow.
5. The terminals are specified with a throw length with an end velocity of 0.20 m/s. This end velocity can be corrected for different operating conditions, to ensure correct flow is achieved without problems with draughts in the room. This guide includes information how this work can be performed, see section Designing for mixing ventilation.
6. The calculation of the resulting sound level from air terminals and duct system in relation to the current sound absorption in the room should always be performed.

Other factors that must be considered are e.g. for vertical air supply:

When heated and cooled air is to be supplied to a room vertically, this affects the throw length. For heated air it is shortened while it is extended for cooled air. This relation can be calculated using calculation software where the flow, temperature variations between the supply air and room air temperatures, and supply angle are stated.



Supply air, ceiling - Extract air, ceiling



The diagram shows which air exchange efficiencies (ε_{ra}) can be obtained as a function of the supplied power with air.
A = Supply air diffuser by the floor and extract air diffuser in the ceiling
B = Supply air diffuser in the ceiling and extract air diffuser in the ceiling

Summary for mixing air ventilation

- Common flow forms
- Schools, offices
- Low ceiling heights
- Supply air usually at ceiling level
- High comfort due to the small temperature differences in the room
- Withstands large temperature differences on room temperature when choosing the right terminal (cooling)
- Low risk of draughts

Thermally controlled ventilation

The thermally controlled ventilation is characterised by the supply air, which is below room temperature, being supplied at a low speed at floor level. The supply air spread more or less out across the floor surface and is affected in the room where the heat sources occur. The supply air flows with the help of these heat sources upwards and is evacuated. The thermally controlled ventilation is therefore split into:

a) displacement ventilation

b) equalising ventilation

One of the conditions in order for the principle to work without comfort problems is that the air distribution across the supply air diffusers can change, if necessary. By changing the air distribution depending on how the occupied zone is used prevents comfort problems. By adapting the supply air flow and the number of supply air diffusers to the heat emitting sources (machines, people, etc.), a ventilation system with high efficiency and good comfort can be produced.

Displacement ventilation

The displacement ventilation is characterized by:

1. Supply air is supplied at floor level
2. No appreciable mixing of the room air in the supply air is attempted
3. The supply air is supplied at low speed
4. Supply air is below room temperature
5. The extract air is evacuated at ceiling level

Equalising ventilation

The equalising ventilation is named because attempts are made to equalise the temperature distribution within primarily the occupied zone. The principle is characterized by:

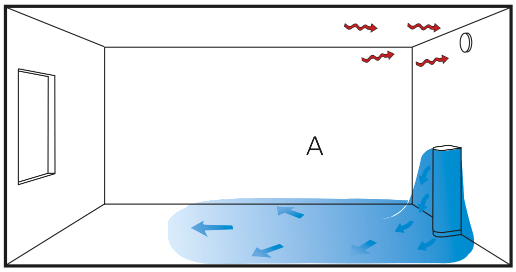
1. Mixing of the room air in the supply air is attempted
2. The supply air is supplied at low speed
3. Supply air is below room temperature
4. The extract air is evacuated at ceiling level

Mixing of the room air in the supply air is achieved e.g. by

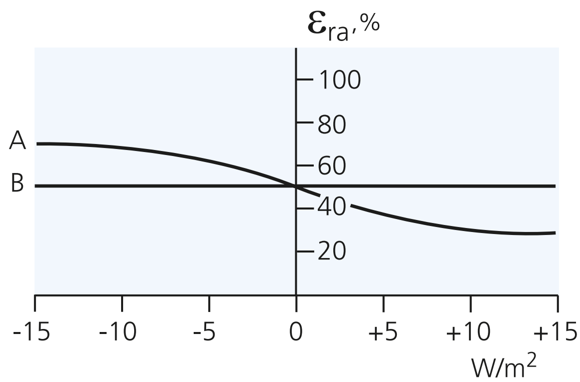
1. Place the supply air diffusers high in the room
2. Create co-ejection of the room air in or adjacent to the supply air diffuser

Summary for thermally controlled ventilation

- Large air flows
- Large areas
- Heat source in the room
- High ceilings
- Limited cooling capacity
- Large near zone – risk of draughts



Supply air, floor - Extract air, ceiling



The diagram shows which air exchange efficiencies (ε_{ra}) can be obtained as a function of the supplied power with air.
A = Supply air diffuser by the floor and extract air diffuser in the ceiling
B = Supply air diffuser in the ceiling and extract air diffuser in the ceiling

Displacement ventilation

The illustration below shows a displacement ventilation system. The ventilation air is supplied below room temperature at floor level and is drawn out at ceiling level. The ventilation air is distributed along the floor and starts to rise upwards when it comes into contact with hot bodies, which generates upward convection currents.

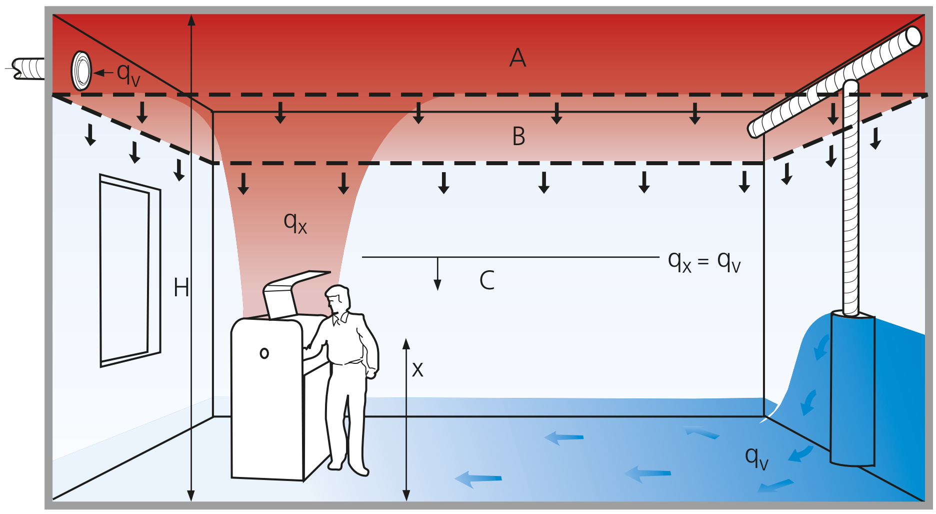
The warm point source in the illustration produces a impurity, for example, heated air which is lighter than the ambient air. The impurity rises towards the ceiling and more air is drawn into the plume-like shape.

If the volume flow of air into the impurity plume, when it reaches the ceiling, is greater than the ventilation airflow, the whole contaminated flow cannot be directly evacuated by the ventilation air. A part of the impurities will therefore be recirculated downwards in the room. We get a front with contaminated air that begins to move downwards in the room.

The front stops at the level where the volume flow of air in the upwardly rising impurity plume is equal to the ventilation airflow.

Two zones are formed in the room, an upper zone with polluted air and a lower zone of "clean" air. In rooms with mobile work and a high level of activity, it is desirable to get the clean zone as high as possible and preferably above the breathing zone.

The requisite airflow is determined on the basis of applicable standards and regulations about hygienic threshold levels.



The air is supplied below room temperature at floor level and is drawn out at ceiling level. The supply air (q_v) is distributed along the floor and rises up when it comes in contact with hot bodies.
q_v = ventilation air flow l/s
q_x = convection airflow in the impurity plume at level x (l/s)

A = Contaminated zone
B = Variable front with polluted air
C = Clean Zone

Piston flow

Piston flow involves ventilation air being distributed in such a way that the airflow direction is uniquely determined and only moves in one direction. The air can be said to go like a piston through the room.

Relatively high air speeds are required for piston flow to work. Velocities ≥ 0.35 to 0.40 m/s are required to get a stable piston flow in the room. On account of these high velocities, piston flow never comes into question in connection with comfort ventilation. The main use is instead associated with clean rooms where very high demands on air quality in the room.

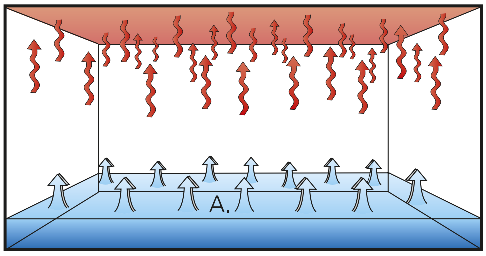
Short-circuit flow

This is a type of flow, which as far as possible, must be avoided.

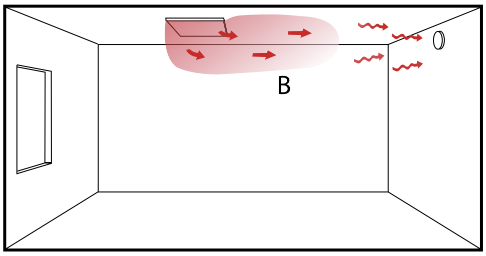
The flow means that a part of the supply air goes directly out with the extract air without benefiting the occupied zone, i.e. the supply air is short-circuited.

The condition, in order for the flow to form, is that both the supply air and extract air diffusers are located at ceiling level and that the supply air velocity is too low and the supply air temperature is higher than the room temperature.

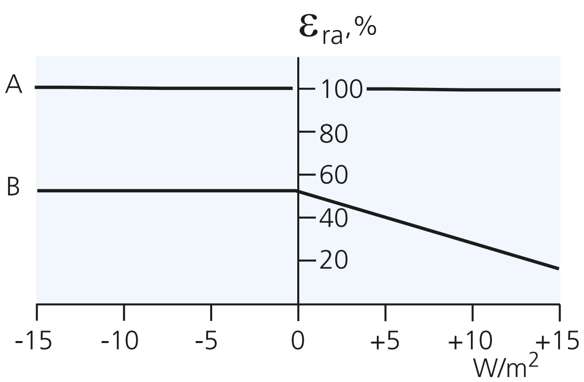
The short-circuit flow can also be obtained in cases when the supply air diffusers are located at floor level (low velocity) and the supply air (which is below room temperature) escapes through open doors or low positioned extract air diffusers.



Supply air, floor - Extract air, ceiling
In the event of pure piston flow the air exchange efficiency is equal to 100%.



Supply air, ceiling - Extract air, ceiling (heated air)



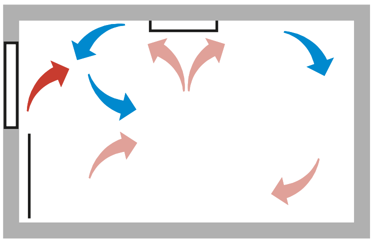
The diagram shows which air exchange efficiencies (ε_{ra}) can be obtained with different diffuser placements as a function of the supplied power with supply air.
A = Supply air diffuser by the floor and extract air diffuser in the ceiling
B = Supply air diffuser in the ceiling and extract air diffuser in the ceiling

Practical guidelines

The principal supply air systems normally used in premises in connection with comfort ventilation are specified below. The maximum cooling loads that are specified refer to airborne cooling capacity, and take into account accumulation in the building structure.

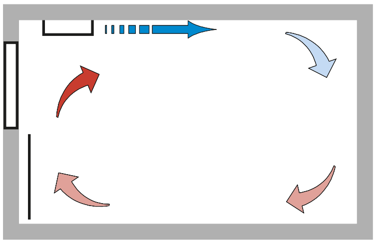
Ceiling placement

Be careful with the definition of the occupied zone. In cases where downward air currents can be accepted at the walls, the principle is advantageous. Look out for convection currents from the window wall.



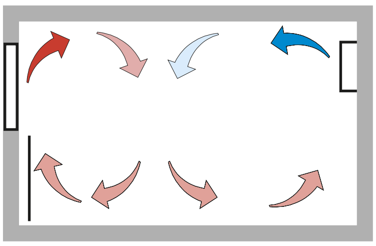
Ceiling placement, front edge

For the supply of air below room temperature, a satisfactory result is usually obtained if the supply air diffuser is designed for a throw length, that is slightly longer (1 to 2 m) than the room depth.



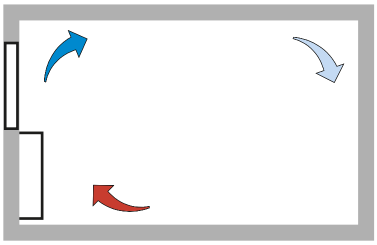
Rear edge placement

Selection of the supply air diffuser is extremely significant as the throw length is very important. For example, for an office where the workstation is located near the window wall, max $l_{0.2}$ should be equal to the depth of the room multiplied by 0.7 when air below room temperature is supplied. Longer throw lengths can ideally be used if the occupied zone does not go closer to the window than 0.75 m.



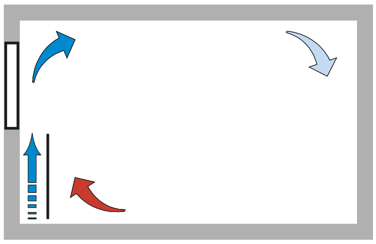
Window seat placement

Be aware of the supply air and window temperature. In cases where there is a risk where both of these temperatures may be lower than the room air temperature, there is a risk of air plunges in the zone closest to (1 to 2 m) the window.



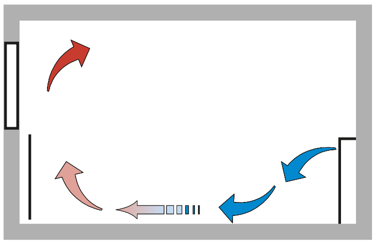
Placement behind radiators

This placement requires a high outlet velocity from the supply air diffuser to prevent plunging in the occupied zone. It is important that the windowsill if fitted is designed so as to not to direct the air into the room.



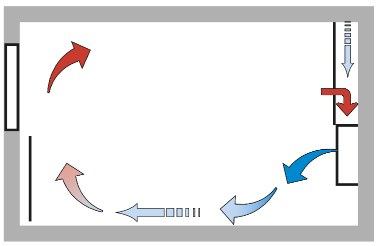
Floor placement, low speed

If diffusers with a flexible distribution pattern are used, a temperature difference between the room air and supply air of max 6°C can be applied. Maximal cooling load about 35 W/m^2 at ceiling heights of approximately 2.8 m.



Floor placement, low speed with ejector

With a special ejector section as an add-on to the low speed diffuser equalizes the temperature distribution in the room's occupied zone. Temperature differences between 6 and 9°C between the room and supply air can therefore be applied.



Basic facts

Exchange time and air exchange efficiency
The exchange time and air exchange efficiency of the air in the room at different air flow conditions

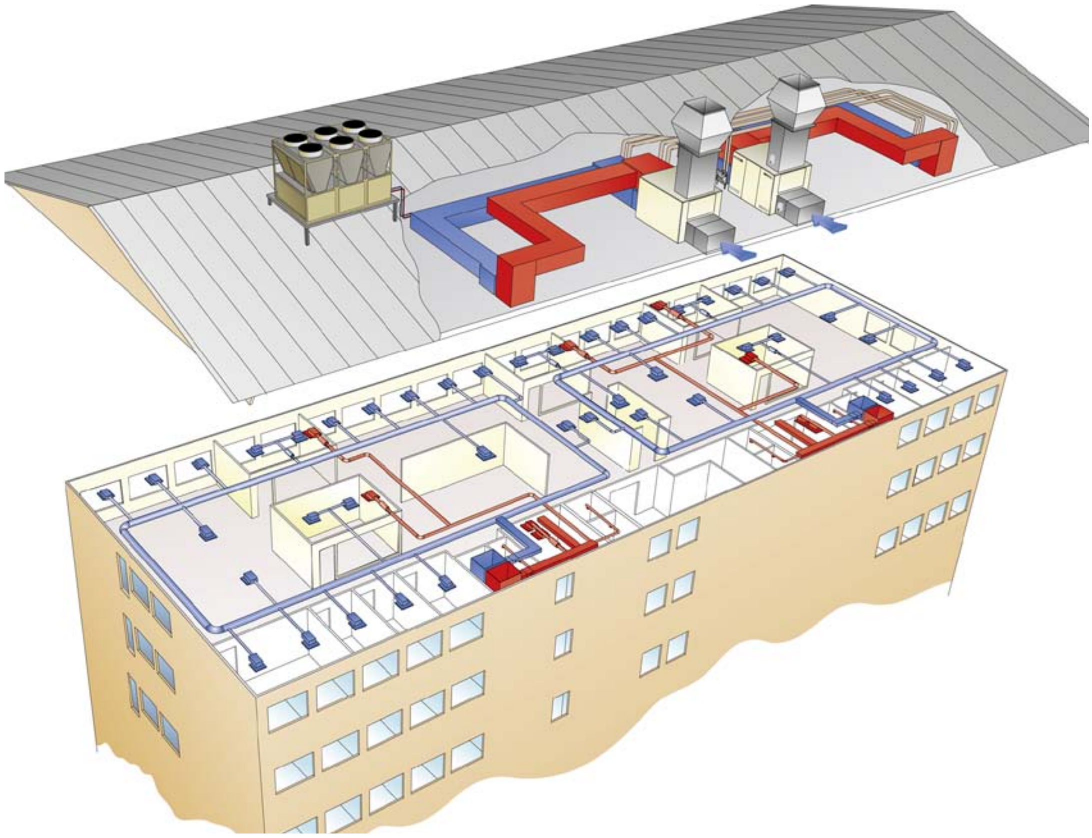
Airflow	Exchange time of air in the room $2 \cdot \tau_n$	Air exchange efficiency ε_{ra}
Displacing and equalising	$> \tau_n$ $< 2\ \tau_n$	<100 % >50 %
Mixing	$2\ \tau_n$	50 %
Piston flow	τ_n	100 %
Short-circuit	$> 2\ \tau_n$	<50 %

System solutions in different building types

In this chapter a number of different system solutions are presented for different types of buildings. The solutions must be seen as typical examples of solutions with high energy efficiency and comfort.

Offices, airborne systems

The aim of a system designed for offices is to create the perfect indoor climate, at the lowest possible energy cost, so that the client can feel secure as regards both the investment and when operating the system.



The illustration shows a system application for office buildings comprising several storeys. Nowadays, many offices are sized for the maximum number of people present. The level of occupancy in an office varies according to the business in question. In many cases the office is far from fully occupied, which means that the installed ventilation system is oversized. This costs money and energy. The solution is a system that is demand-controlled with regard to occupancy and load in the office.

The office storeys are supplied by a system divided into two centrally located air handling units and externally produced electricity, cooling and heating media. The air handling system demand-controls the office building's airflow requirement.

The offices are air conditioned and ventilated with the supply air diffusers and radiators.

The conference rooms are air conditioned with variable flow dampers, ceiling terminals and radiators.

Public areas, such as the reception, kitchenette and cafeteria, are air conditioned with air terminals.

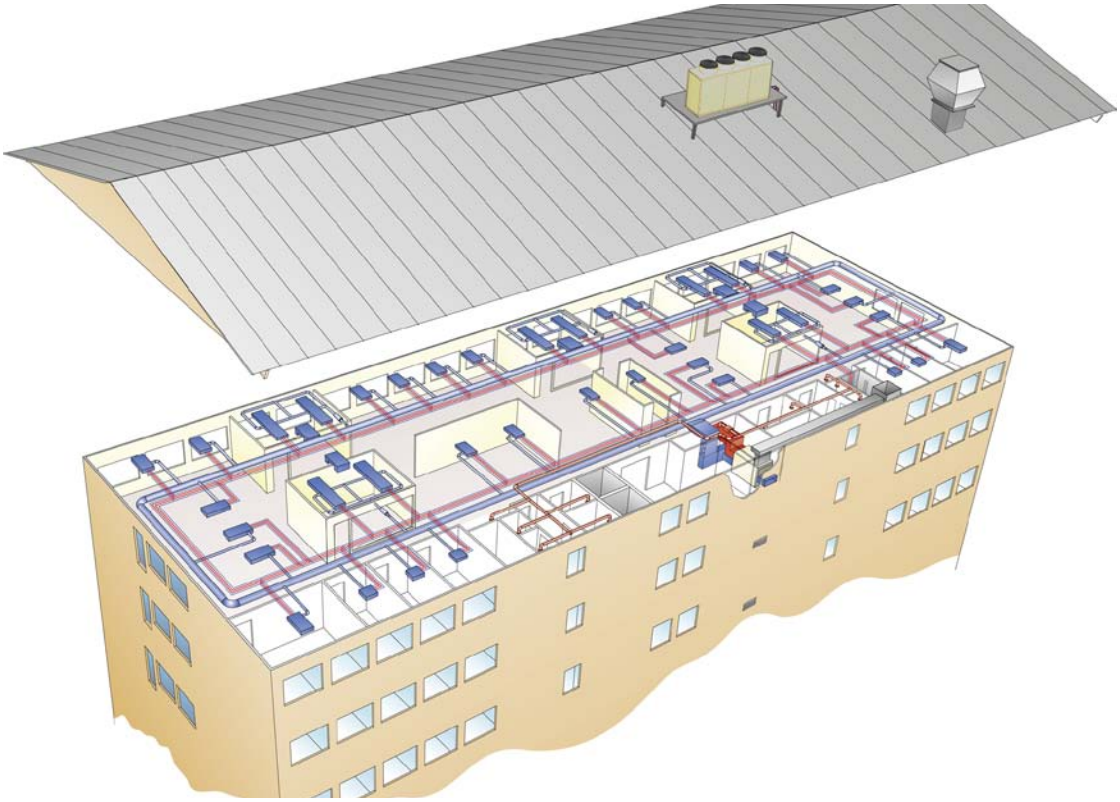
A communication unit connects the system with the air handling units.

Advantages of airborne systems

- Energy efficient free cooling can be utilised
- Low installation cost
- Requires little space.
- Best air quality

Office, waterborne systems

The aim of a system designed for offices is to create the perfect indoor climate, at the lowest possible energy cost, so that the client can feel secure as regards both the investment and when operating the system.



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Each office storey has its own climate system that meets air quality and room climate needs.

Room-controlled comfort modules provide exactly the desired temperature in each room.

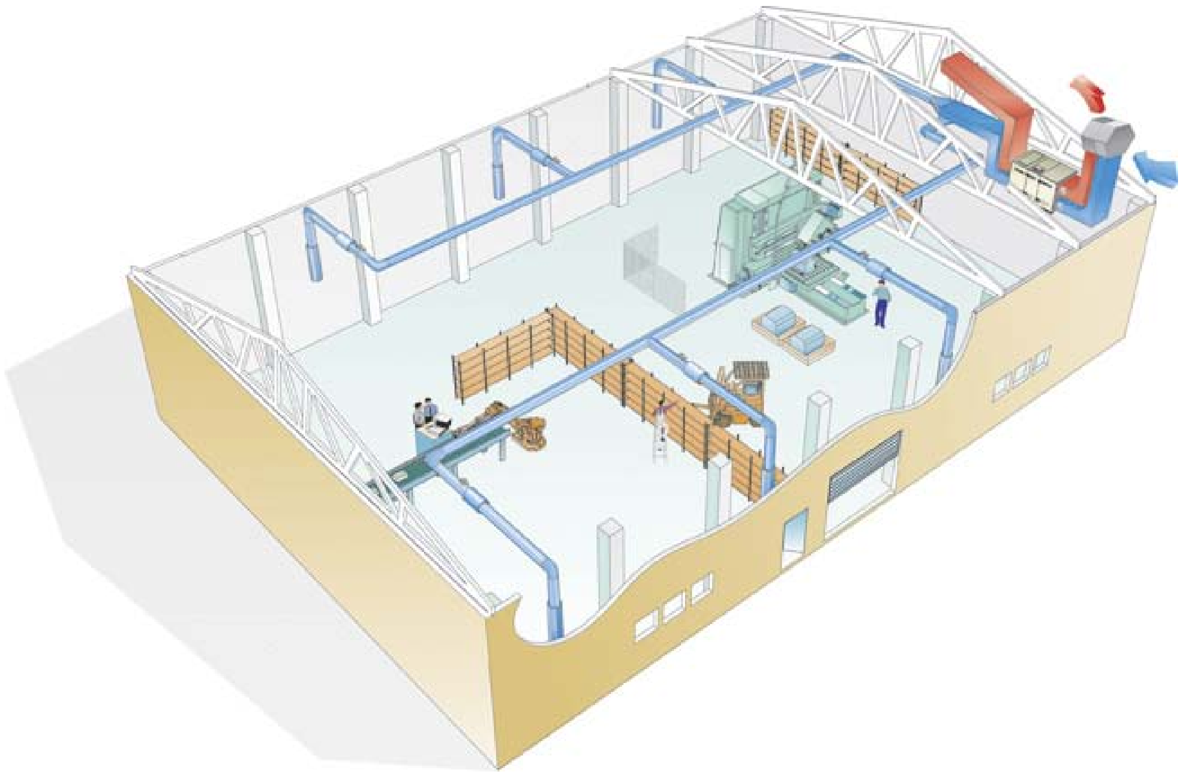
The air handling unit provides exactly the right amount of air via pressure regulation with the communication unit, as well as optimum temperature on the cooling and heating water via a function that controls the primary water circuit.

Advantages of the waterborne system

- Individual climate regulation
- Smaller duct dimensions
- Ventilation, heating and cooling in one system
- High cooling and heating capacities
- Maximum thermal comfort

Industrial buildings

The indoor climate in an industrial building must meet special needs. Naturally as a workplace it requires ventilation. From a temperature standpoint, it has different needs during different seasons but usually a heating load prevails at night and in the morning and a cooling load during the day. The premises usually have large areas and ceiling heights.



High quality air handling units and supply air diffusers can meet these requirements in a very efficient manner.

A suitable air handling unit delivers air, heating and cooling, and has built-in control equipment. The unit should also be equipped with return air dampers, allowing for energy-saving and efficient heating at night and early morning.

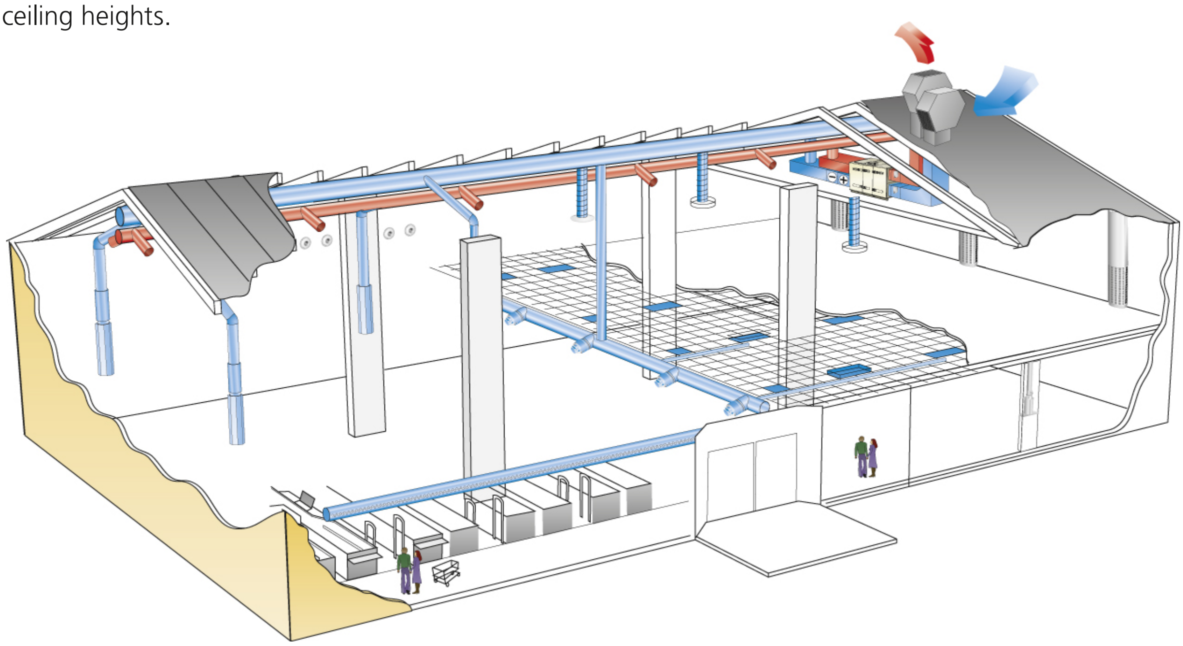
The supply air diffusers should be adapted to provide both heating and cooling. In the event of heating requirement, the supply air passes through the upper, nozzle-equipped, section. The air is forced at a high velocity, without causing draughts or noise, down towards the floor. In the event of a cooling requirement, the supply air passes through the lower perforated section. The air is delivered at low speed and naturally moves towards the floor.

Functions

- Unoccupied premises: 100% return air, heating if necessary.
- Before the shift: 100% return air, morning adjustment where the temperature is raised to the desired value.
- During the shift: Designed water flow.
- In the event of a heating load, forced heating
- In the event of a cooling load, forced cooling.
- The supply air diffusers are designed to deliver either heated or cooled air.

Commercial buildings

The indoor climate in large commercial buildings must meet special needs. The temperature must be uniform throughout the year and throughout the day. During large parts of the day there is excess heat and usually there are also large variations in different parts of the premises. Large glazed areas give a heating need. The premises usually have large areas and ceiling heights.



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Functions

- Air handling units with built-in control equipment
- The supply air diffusers are designed to deliver either heated or cooled air.

Public buildings

In public buildings, such as schools and libraries, the load caused by occupants varies substantially as the day progresses. This normally requires complicated solutions to achieve correct airflow and temperature at every individual point in time and for every individual load condition.

A simple and cost-effective solution permits mixing several types of zones, both with variable airflow and constant airflow. The concept is based on an air handling unit supplying the rooms with preheated/cooled air at a constant supply air temperature of 15-18°C. At room level, an intelligent room controller controls temperatures and air quality in response to signals from temperature sensors and CO_2 sensors, or occupant detection sensors. This solution provides a demand-controlled and substantially energy-saving ventilation system.

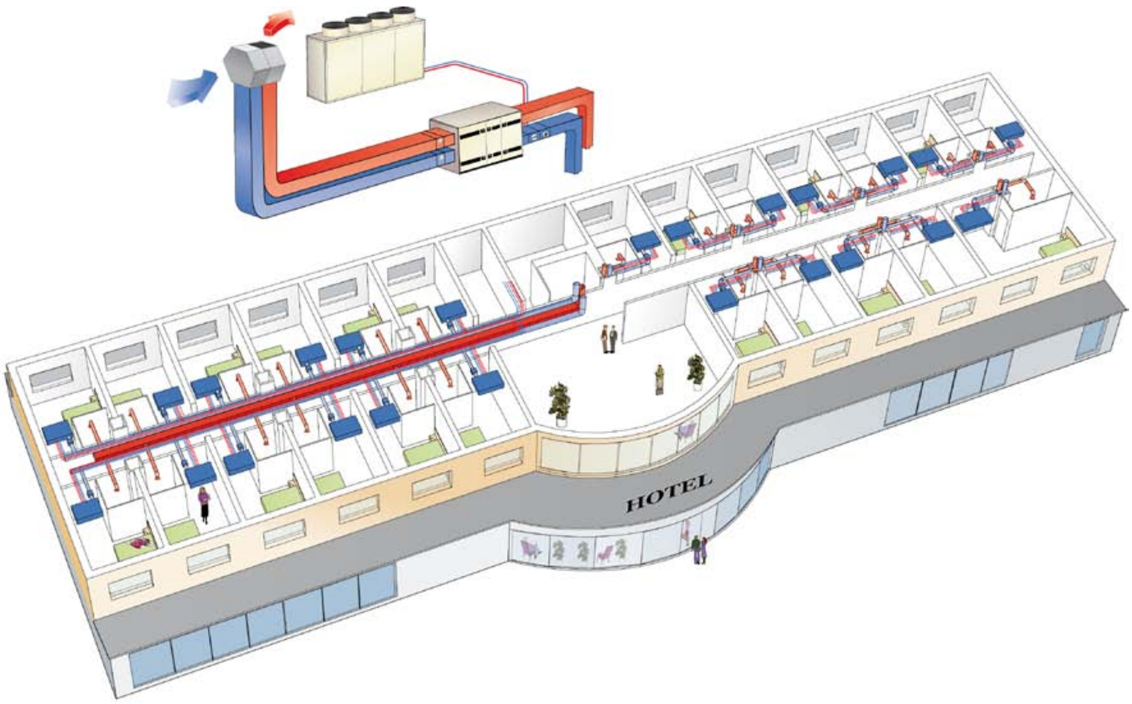
The air handling unit together with the active or passive air terminals and regulators constitute a flexible package solution. Ventilation is then demand-controlled and several different control options can be utilized.

Functions

- Constant temperature on the supply air temperature is set at the preset value and has provision for winter and summer compensation.
- Constant pressure in the supply air duct.
- The extract air is evacuated as common transfer air and is slave controlled to the same flow as the supply air.
- Rooms with active air terminals.
- Rooms with passive air terminals.

Hotels

The most important economic key figure within the hotel sector is the occupancy level. The occupancy level is directly dependent on the well-being of the hotel guests, satisfied guests always return. One of the most important requirements that hotel guests request, is that the room should be perceived as "fresh, quiet and comfortable". Most also want to have an opportunity to influence the temperature in the room. The requirement of the system solution is thus to create an individual demand-controlled indoor climate with optimal operating economy.



A proposal for a principle solution for hotels with a demand-controlled ventilation system must be characterized by a holistic approach and complete with all necessary components. The example shows a system, based on a typical hotel with five floors, divided into two subsystems, which together create opportunities of substantial savings compared to the today's standard hotel solutions.

Cooling and heating production is provided by a multifunctional unit. The individual hotel rooms are air-conditioned and ventilated with compact comfort modules and the control systems are connected to a centrally located air handling unit.

Public areas such as the restaurant, lobby and conference room area are air conditioned with comfort modules, which, together with the air handling system, demand-controls the airflows in these areas.

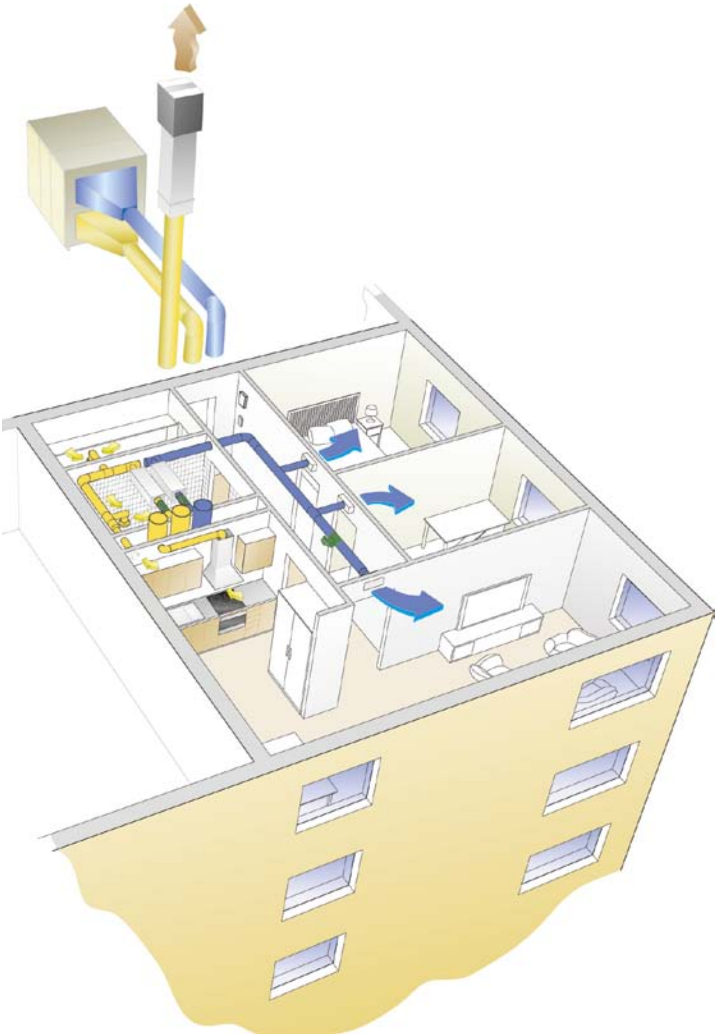
In order to optimise fan operations and ensure the room climate in all rooms, flow control is linked together with the air handling unit.

Savings potential

- Demand-controlled ventilation gives approximately 30% lower energy consumption
- Zonal supply air temperature
- Room heating with maintenance-free comfort modules.

Apartment buildings, central solutions

A good indoor climate in apartments is a prerequisite for people to feel good. Increasingly stringent energy requirements, demand new and energy efficient solutions. At the same time, the installation must handle cooking fumes and other odours without transferring these to other apartments or stairwells.



The example shows an air handling system for demand-controlled indoor climate, where each apartment features damper control via a control panel. Energy consumption is below 20 kWh/m^2, year, which by a wide margin is within the framework of the EU energy directives for buildings' energy use.

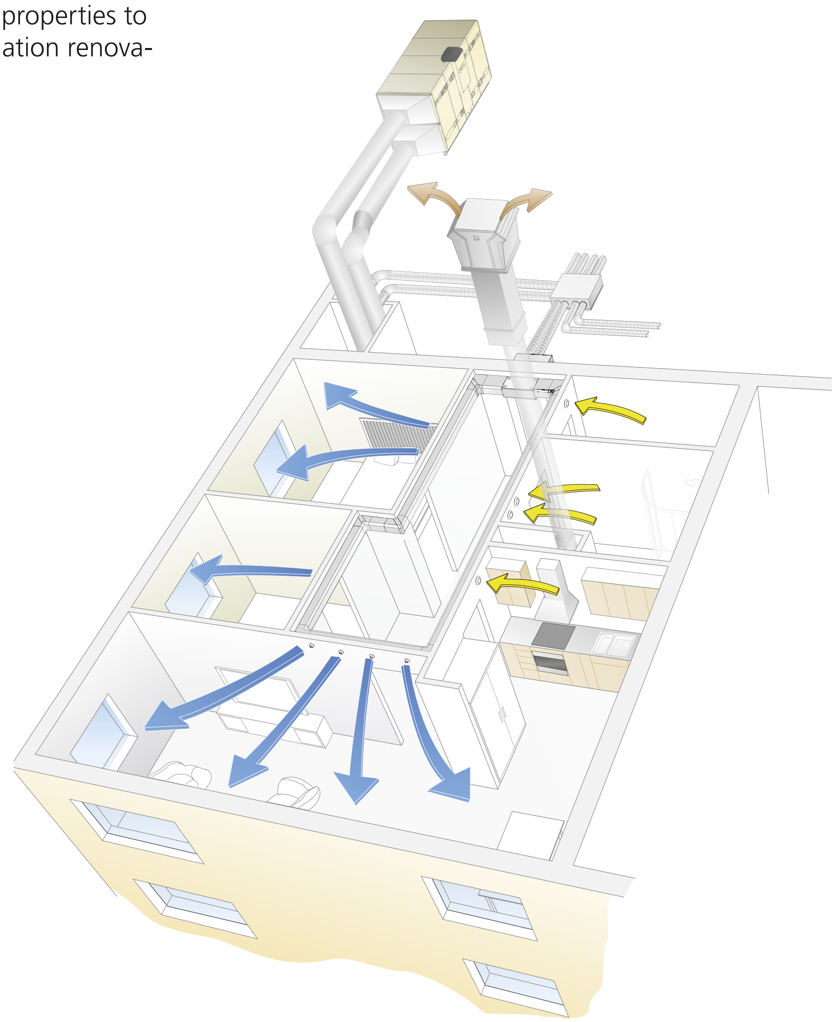
An air handling unit manages the supply and extract air to an appropriate number of apartments via pressure control. The airflow is controlled in each apartment as needed via a control panel and a timer. The system controls to an economical low flow when no one is at home. At other times, the system operates at normal flow and when using the cooker hood with forced ventilation. An economic special function is that the supply air is principally directed to the living room during the day and to the bedroom at night. Thus, a lower airflow per apartment can be used and still satisfy the indoor climate where you are in the apartment. The solution is energy efficient because the ventilation is adjusted to where the people are in the apartment and what they do.

Advantages

- Good air quality.
- Simplicity for the user and installer.
- Small energy use.

Apartment buildings, central solutions for renovation

It is a challenge both time and cost-wise for owners of older residential properties to implement the necessary ventilation renovation.



A proposal for a solution that is relatively easy and cost effective is to install a system that automatically controls the need in each individual apartment.

A central ventilation unit is connected via main ducts or shaft to a distribution box for each landing. A technical box, containing motorised dampers, air quality, humidity and temperature sensors, is placed in the stairwell, and above the door of each apartment. It is docked to a fire cartridge with fire dampers.

From the technical box/fire cartridge holes are made in the apartment's hallway and a distribution box. A loop with ducts for the supply and extract air is installed against the angle between the ceiling and wall in the hallway, which is subsequently covered with prefabricated duct casing. Holes are drilled in the walls for the rooms that are to have supply or extract air.

The central ventilation unit is pressure controlled. Each apartment is regulated independently of the neighbouring apartments through the technical box, where the airflow is automatically adapted to the signal from air quality and humidity sensors.

At the same time as this involves maximum comfort for both individuals and the building, it also means energy efficient operations, where you ventilate just as much as needed.

Advantages

- Demand-controlled ventilation with automatic regulation of the air quality and humidity per apartment.
- High energy efficiency
- Integral mechanical fire solution
- Fast and cost-effective installation

Apartment buildings, decentralised solution

With a decentralised solution for ventilation in apartment buildings, each apartment has its own ventilation system.

Unit placement
The simplest solution is usually to place the ventilation unit in bathrooms or other suitable spaces. The downside is that service personnel must enter the apartment to replace filters and other maintenance.

It is also possible to build the ventilation unit into the stairwell. It may involve more complicated ducting, but the advantage is that the service can be performed without access to the apartment.

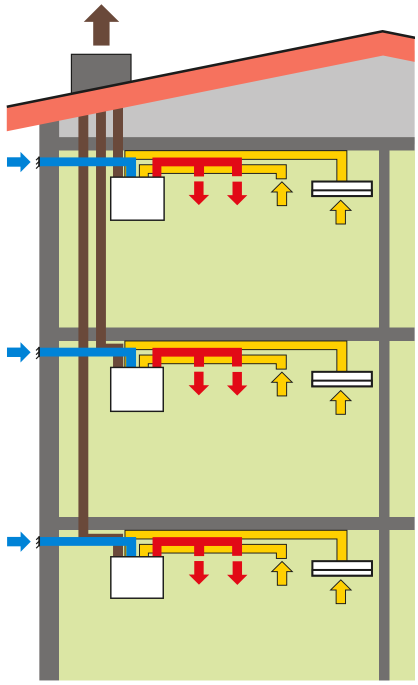
Outdoor air and exhaust air
The simplest solution is usually that both outdoor air and exhaust air for each unit is taken via the facade. It is also possible to utilise the shaft or main ducts.

Cooker hoods
Installation and investment costs will be lower if the ventilation unit's extract air fan is also used for the cooker hood's airflow.

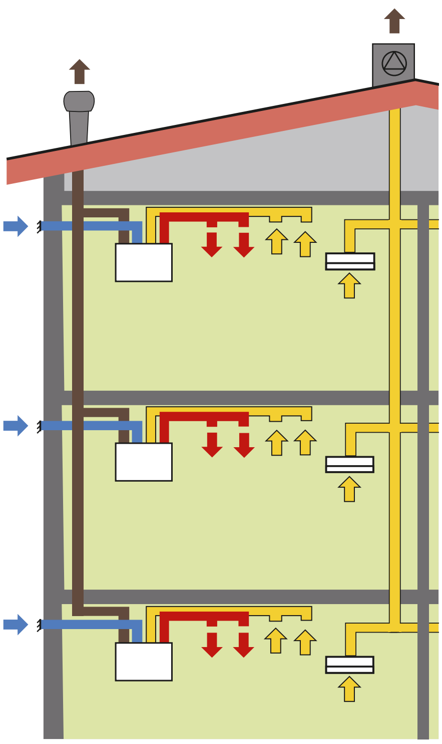
Another solution is for cooker hoods to be connected via a shaft or main duct to the central ceiling fan.

Miscellaneous
Local restrictions may prevent leading exhaust air to the facade. Local restrictions may also prevent leading the cooker hood's exhaust air through the unit.

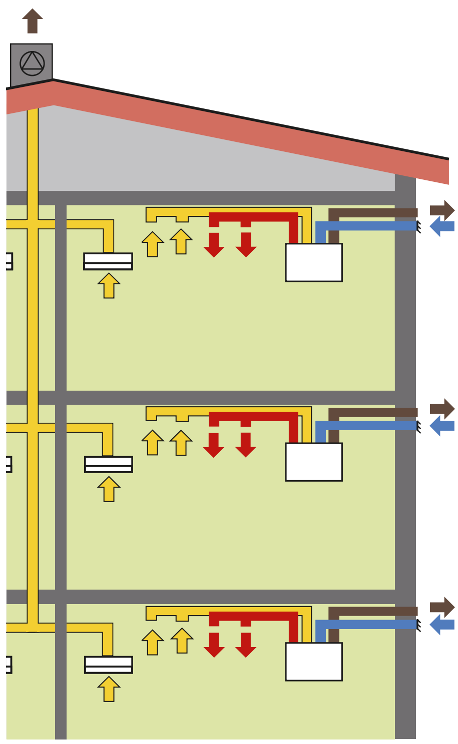
Irrespective of the solution, separate measures may be necessary to prevent air transfer, crosstalk and for fire protection.



The solution with outside air via the facade, cooker hood attached to a ventilation unit and separate exhaust ducts to a central exhaust hood.



Solution with outdoor air and exhaust air via the facade, the cooker hood connected via main ducts to the central ceiling fan.



Solution with outdoor air and exhaust air via the facade, the cooker hood connected via main ducts to the central ceiling fan.

Detached and terraced houses

In principle, detached and terraced houses always have a separate ventilation system.

Outdoor air and exhaust air

The most common solution is that outside air is taken via the facade and exhaust air is led to the roof hood.

Cooker hoods

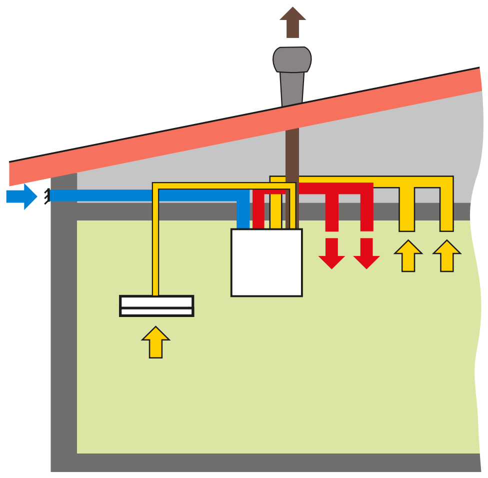
Installation and investment costs will be lower if the ventilation unit's extract air fan is also used for the cooker hood's airflow.

Another solution is to connect the cooker hood to its own roof fan, alternatively, that the cooker hood has built-in fan.

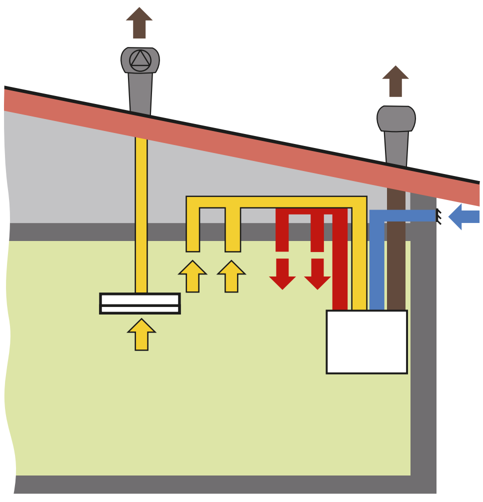
Miscellaneous

Local restrictions may prevent leading the cooker hood's exhaust air through the unit.

No specific actions are normally needed in detached and terraced houses against air transfer, crosstalk or fire protection.



The solution with outside air via the facade, exhaust air to the exhaust hood, cooker hood attached to a ventilation unit.



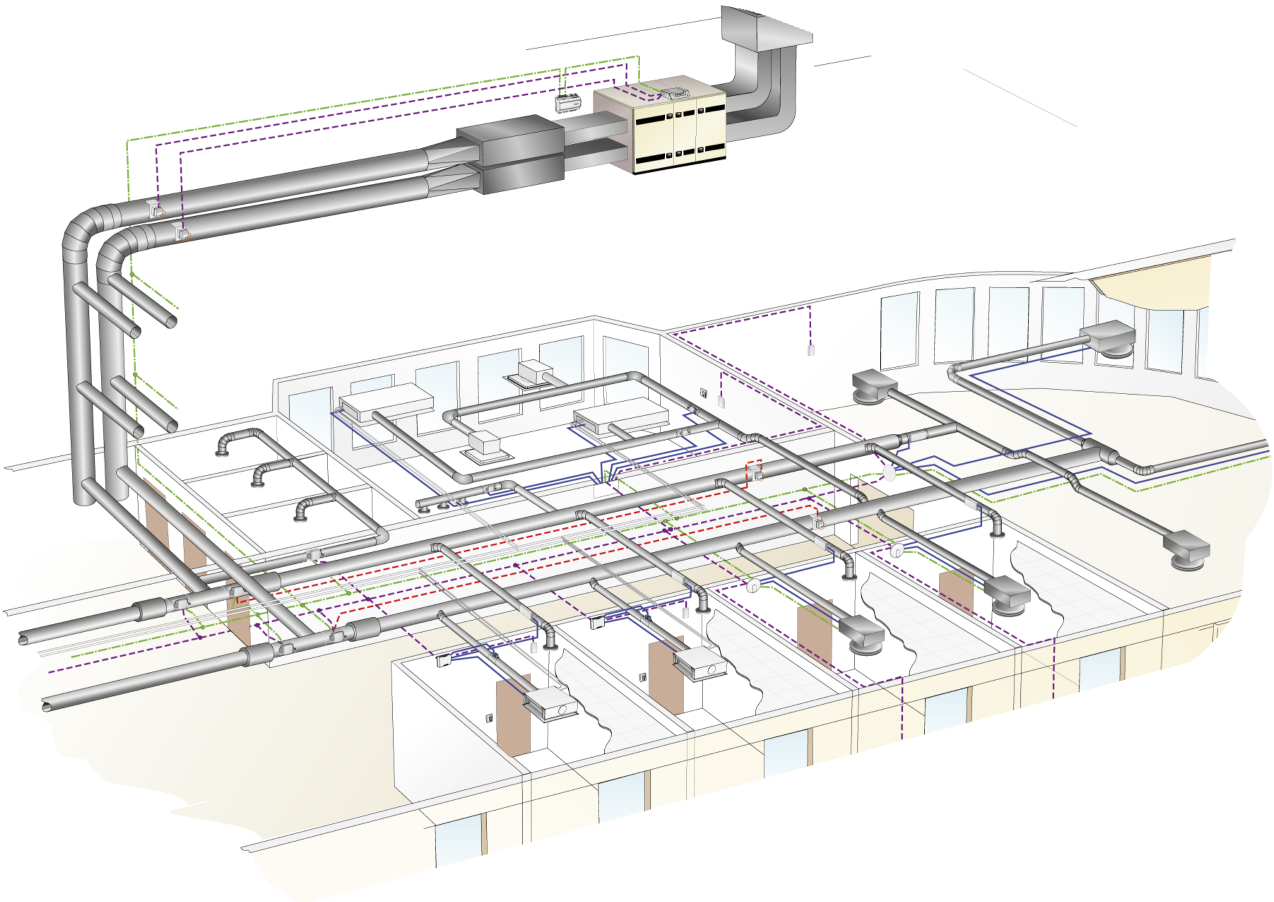
The solution with outside air via the facade, exhaust air to the exhaust hood, cooker hood attached to a ceiling fan (or fan of its own in the cooker hood).

Project planning

Introduction

The planning of an installation for indoor climate must take into account a number of factors. In addition to the specific requirements demanded by the client/user, the requirements as discussed in the previous chapter, "From requirements to a technical solution" must be met.

This planning chapter provides instructions and guidance on how the different systems and components in systems can be designed.



Project design for a good acoustic environment

Ventilation systems impact on the acoustic environment

A ventilation system in a building affects the acoustics through sound generation, crosstalk in the duct system, leakage in cut-outs and increased room attenuation of diffuser openings in the room.

In addition, there are the effects of vibration forces caused by fans. All these aspects must be considered during the design stage.

Sound generation

Sound is generated by several factors such as fans, dampers, diffusers and air currents.

Fans

Sound power levels in dB are usually stated by the manufacturer in octave bands and as the total sound power level.

Dampers

Sound power levels in dB are usually stated by the manufacturer in octave bands. The levels are often specified in the octave band between 63 Hz and 8000 Hz.

Air diffuser

Inherent sound to rooms is usually stated as the noise level in dB(A) related to the room attenuation 10 m^2Sabine, which means that the sound level applies at a distance from the diffuser in a room with 10 m^2 of sound absorption area. If the room's occupied zone reaches up to the diffuser the fact that the noise level in the diffuser's near field is much stronger than the noise level at a distance in the middle of the room must be taken into consideration.

Airflow

The air flow in ducts creates turbulence with unevenness, branches, connections, etc., and therefore sound. In most cases quality diffusers are designed with sufficient attenuation in order to meet the sound generation from an air velocity of 8 m/s in the main ducts and at most 4 m/s in the branch ducts. Variations occur, especially in rooms with demands for low background levels.

Working method

This working method follows the traditional design management with idea sketches, system documents and construction documents. The method consists of:

- In the early stage - sketch phase, determine the measures necessary to attenuate the sound from fans to rooms and to the surroundings in terms of size and fundamentally design the system.
- Detail design is performed at the end of the stage and generally proposed measures are defined.

During the first sketch phase you should, together with the architect:

- Determine the geometry and volume of the main body of the house.
- Suggest the number of units and machine room with respect to the building's form and activities.
- Determine the size of the machine room.
  NOTE! The size has more dimensions than the length, width and height. Also state here the noise level and dynamic disturbing forces on floor structures, power requirements, etc.

The natural steps

A

Knowing the ventilated volume, the size of the unit can be specified. Thus, the general sound power from the unit can be determined and the measures required in order for the unit's sound to be accepted in the building and the surroundings.
The unit room's dimensions are specified, i.e.:

- the required area in m^2 and
- the requisite noise level in the unit room and the
- need of vibration isolation with the demanded floor structure..

The size of unit rooms is determined by whether the attenuation measures have space to limit the noise emission to the surroundings and that the sound power level to the shaft does not exceed 60 dB(C) in the duct. All low frequency attenuation shall be achieved in the unit room.

All sound attenuators that contain porous sound absorbing materials must be accessible. Sound attenuators must be possible to open or be accessible by other means and the porous material to be cleaned or replaced.

The vibration isolation of unit makes demands on the floor structure and the height in the unit room. This must be determined at an early stage so that the right floor structure or supplementary measures can be designed.

B

All sound requirements are compiled and shall include:

- maximum design noise level dB(A)/dB(C) in the unit room, stated by the ventilation consultant
- maximum design noise level dB(A)/dB(C) in the shaft, stated by the ventilation consultant
- maximum design noise level dB(A)/dB(C) above the corridor ceiling, etc., stated by the ventilation consultant
- noise level in the room dB(A)/dB(C), stated by an acoustic engineer or architect
- sound insulation in partition walls and floor structure R_WdB, stated by an acoustic engineer or architect
- noise emission to surroundings, L_W sound power in outdoor air and exhaust openings, stated by an acoustic engineer or ventilation consultant.

C

Principle documents are drawn up where the size of the low frequency attenuators is specified in the machine room and sound attenuators in the duct system are distributed as needed next to the room.

Sound attenuators after throttle dampers to rooms in office buildings and the like. In school buildings and adjacent to meeting rooms with higher demands on crosstalk attenuation this usually means 1200 mm sound attenuators between the throttle dampers and rooms. The different sound attenuators generally give class LD 1, 2, 3, etc.. and are designed in detail in the construction document phase. Duct routing should be designed so that you do not have to go through separating structures with high demands on sound insulation. Perform crosstalk calculations using calculation software. Here the R_W values for duct routing are specified.

D

All duct routing must be designed so that it can be implemented with good accuracy and air-tightness. It is important that the construction documents show the design of lead-through openings that can be performed and sealed in a satisfactory manner. The responsibility for the detailed design is the planning engineer. The design must also be such that assembly can take place without the risk of injury to the fitters, which is also planning engineer's responsibility.

At the end of the project planning, the sound conditions are computed in some critical system branches. The whole system is rarely computed. In connection with the calculation, sound attenuators in the unit room are specified in detail as well as the sound attenuators in the duct system. Previously stated requirements are interpreted into sound attenuator proposals of an adequate size.

The sound level must be determined with a 5 dB margin for each system in the detail design. This means that when the supply and exhaust air systems are run simultaneously, the margin will be 2 dB, which is a necessary margin in terms of the differences in performance and uncertainty in measurements. If the building contains additional systems such as cooling, heating systems, etc. which are likely to affect the noise level in the room, the margin must be extended by an additional 2 dB per additional systems. In the event of three such noise producing systems, these must be calculated against a demand that is 7 dB below the current requirement. If they contribute equally to the sound level in the room, this means together they answer for a noise level that is 5 dB over the design of each system, i.e. the margin will be 2 dB to requirement. This means that the supply air and exhaust air systems in a house with three systems incl. heating, should each be designed to handle 23 dB(A) in e.g. classrooms and residential rooms.

E

The construction documents should include, a document drawn up to show hole-making, lead-through openings and the sealing method and subsequent repair in terms of acoustic and fire protection requirements and flexibility. The sealing tightness is critical in order for sound requirements to be met in the finished house. Products that are initially designed to be easily recessed in walls, floor structures, etc., can provide a satisfactory sealing tightness.

Construction phase

Normal design flaws that lead to unwanted noise are constantly monitored during the construction phase. This includes, among others:

Changing material
Equivalence in terms of sound must be checked. The presentation of equivalence from all of the following aspects should be demanded.

Fan installation
Fans are normally vibration isolated in the unit. It is important to remember that there should be a flexible connection between the fan and duct system.

Duct design, air-tightness, pressure drop
Ducts must be joined without sharp interior edges. Leakage causes noise. Unnecessary pressure drops cause noise and an inferior working state for the fan. Make sure duct routing is smooth with gradual bends and transitions.

Refinishing
All seals around ducting in lead-through openings must be checked. Usually only the visible part of the hole has been sealed. Higher requirements on sound insulation should be measured. Measurements can be made early in production, and act as a guide to the contractor.

Final pressure drop
It is wrong to accept excessive pressure drops across diffusers with sound generation as a result. Most diffusers offer an adjustment option that must be used to equalize the flow between rooms in a branch duct, but not to throttle the entire system's flow distribution.

Commissioning
Must be done so that the diffuser does not give too loud inherent sound to the room.

Checks
Sound levels must be checked during commissioning. Sound to the surroundings must be measured.

Noise from fans

Normally the generated sound power level is specified for fans.

The data is presented in eight octave bands and for different sound paths. The value in each octave band is obtained by reading the total sound power level, $L_{w,\,tot}$, in the fan diagram and correct with the current correction factor, K_{ok} according to the table in fan diagram.

Measurements are normally made according to ISO 3741 or ISO 5136.

ISO 3741 is used for measuring the sound power level emitted to the fans' or units' surroundings and ISO 5136 is used for measuring the sound power level emitted to the duct.

In order to obtain the most realistic values, the sound measurements should be made according to the ISO methods and with the fan in its casing.

If the measurement is done with free standing fans, the result will be a lower sound level. The trade association ASHRAE in the USA states in the Application of Manufacturers Sound Data:

> "For sound measurements, free standing fans have a 5-10 dB lower noise level in octave bands from 250 Hz and lower than fans in the unit casing."

Measurement accuracy

ISO has, when designing the measurement method for sound power level to the duct, also investigated inaccuracies in different octave bands (90% certainty)

Octave band (Hz)	63	125	250	500
Inaccuracy (dB)	±5.0	±3.4	±2.6	±2.6

Octave band (Hz)	1000	2000	4000	8000
Inaccuracy (dB)	±2.6	±2.9	±3.6	±5.0

Acoustic calculations

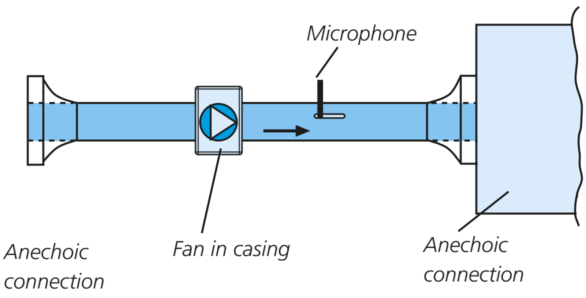
The right column shows a typical fan diagram. The fan's working area is within the hatched area.

The total sound power level $L_{W,\,tot}$ to the outlet ducting can be read (blue colour) from the diagram. The following formula can be used for breaking down the sound power level into octave bands: $L_{W,\,ok} = L_{W,\,tot} + K_{ok}$.

Correction factors for different sound paths, K_{ok} can be obtained from the table.

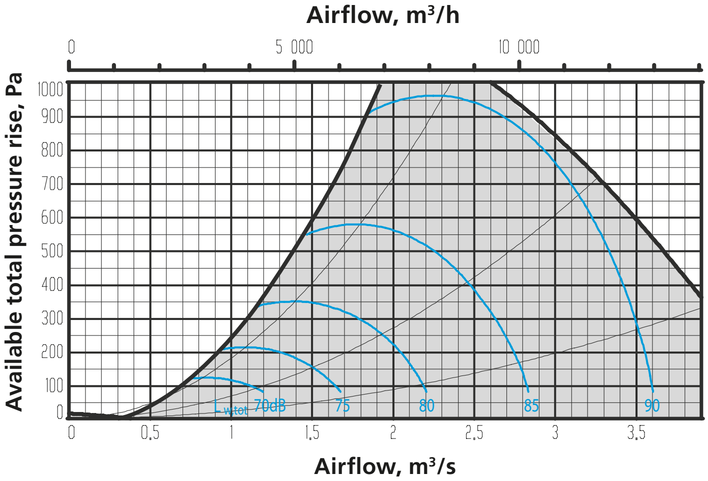
The calculation program can usually produce accurate values for a specific operating instance.

ISO method



Measurements are made inside a duct with a specified design and anechoic connection. Measurements and calculations are made on 1/3 octave band.

Example of the fan diagram



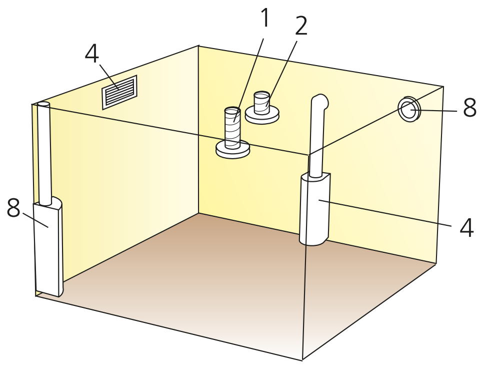
Example of correction factors, K_{OK}, dB

Sound path	Octave band, no./mid-frequency, Hz							
	1	2	3	4	5	6	7	8
	63	125	250	500	1000	2000	4000	8000
To the outlet duct	-4	-9	-7	-5	-8	-9	-11	-11
To the inlet duct*	-9	-10	-10	-21	-29	-32	-36	-33
To air handling unit surroundings**	-15	-23	-30	-26	-41	-42	-45	-42

* The integral attenuation of filters and heat exchanger has been taken into account.
** Total sound power level emitted to the surroundings is calculated as the sum of the levels in the supply air and the extract air.

Sound power level and sound pressure level

Using the diagram below, you can easily determine the difference between the sound power level and the sound pressure level at different distances, r, and equivalent room absorption, A.



Direction factors for different diffuser placements.
Q = 1 Middle of the room, free blowing
Q = 2 Wall or ceiling
Q = 4 Wall near the ceiling
Q = 8 Corner placement

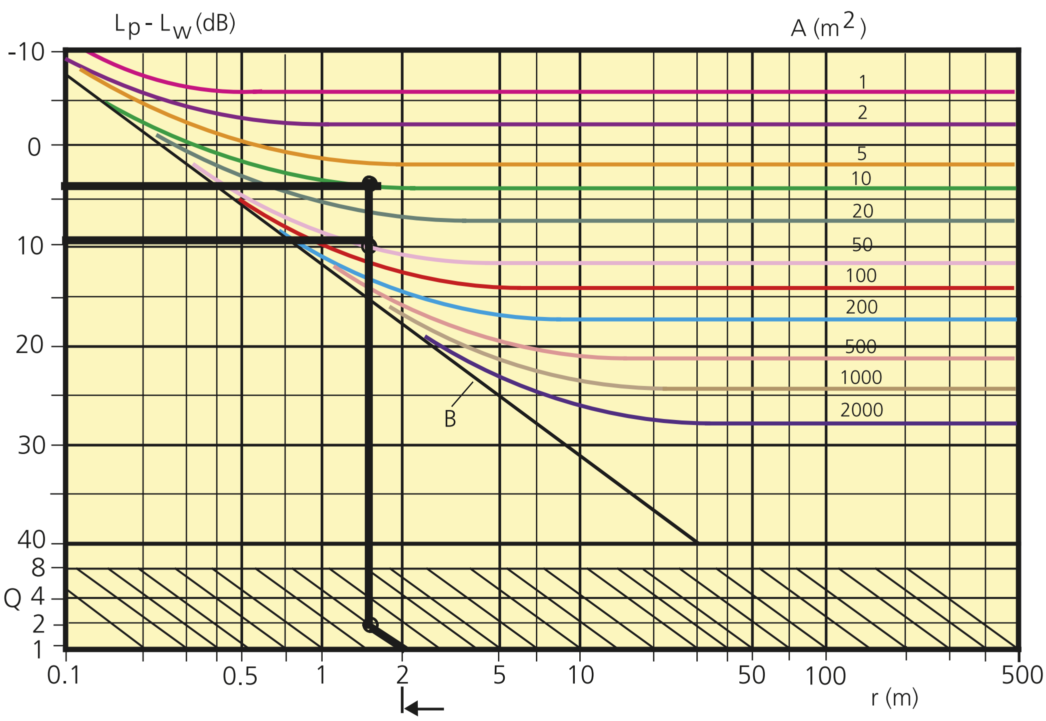
Example
In a room with 50 m² of equivalent absorption area, the distance between the occupied zone and a supply air diffuser is 2 m. The air terminal is mounted in the ceiling (Q = 2). The noise level from the supply air diffuser and the duct system is according to catalogue information 43 dB(A). L_p - L_w will according to the diagram be (9 dB - 4 dB) 5 dB.
The sound pressure level in the room is then L_p = 43 - 5 i.e. 38 dB(A).

Calculating the difference between the sound power level (L_w) and sound pressure level (L_p) as a function of the equivalent sound absorption area (A) and the distance to the sound source (r) at different air terminal placements (Q) is performed as follows:

$$L_p - L_w = 10 \cdot \text{Log}\left(\frac{Q}{4\pi r^2} + \frac{4(1-\alpha_m)}{A}\right)$$

where

Q = direction factor
r = distance to sound source (m)
A = equivalent sound absorption area
α_m = mean absorption factor for the total limitation surface



Difference between the sound power level and sound pressure level
$L_p - L_w$ = Difference between the sound power level and sound pressure level (dB).

A = Equivalent sound absorption area (m²)
B = Free Field
r = Distance from sound source (m)

Sound generation in straight ducts

The sound power level for sound generated in a straight duct is obtained according to the equation:

$L_{wtot} = 10 + 50 \log v + 10 \log s$

v = air velocity in the duct, m/s

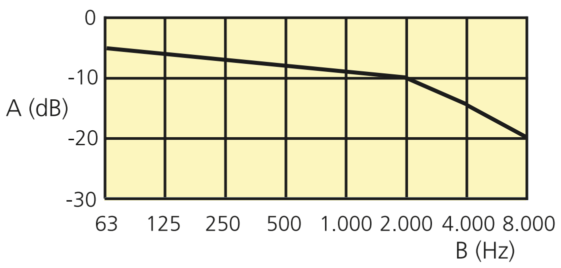
s = the duct's cross-section area, m^2

Example
Air velocity, v = 10 m/s
Duct area, s = 0.5 m^2
The sound power level, L_{wtot}, then becomes 57 dB

The total sound power level is divided on the octave bands as follows:

	Mid-frequency (octave band)							
	63	125	250	500	1000	2000	4000	8000
Output level dB	57	57	57	57	57	57	57	57
Correction acc. figure below	-5	-6	-7	-8	-9	-10	-15	-20
Octave band level	52	51	50	49	48	47	42	37

The octave band distribution of the total sound power level is obtained approximately according to the following figure:

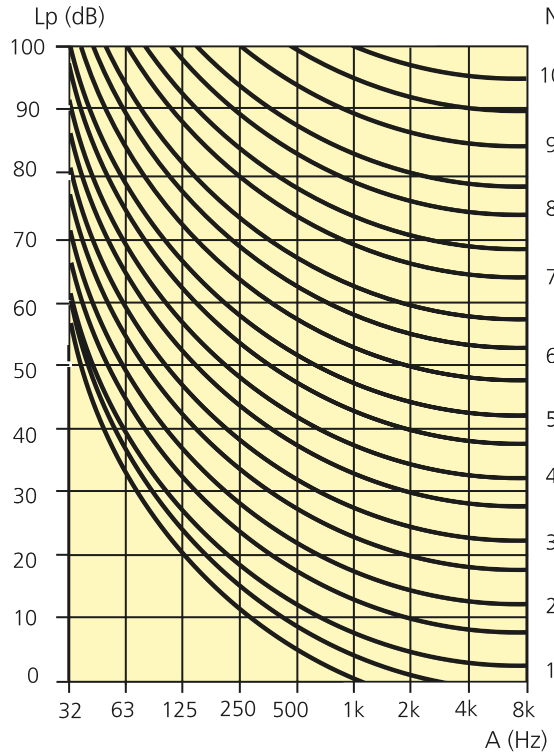


Octave band distribution of total sound power level in straight ducts
A = Relative sound power level dB/octave (dB above 1 pW)
B = Mean frequency octave band (Hz)

Comparisons

In order to get an idea of how disruptive a sound can be, you can compare the frequency analysis of the sound with the normalized noise criteria curves so-called NR curves. The NR value is stated with the number of the highest NR curve, which is touched by the curve for the frequency analysis.

A direct recalculation of a dB(A) value to an NR value cannot be done. As a guide value, the dB(A) value is usually considered to be five units higher than the NR value. However, the difference is dependent on the sound's frequency distribution.



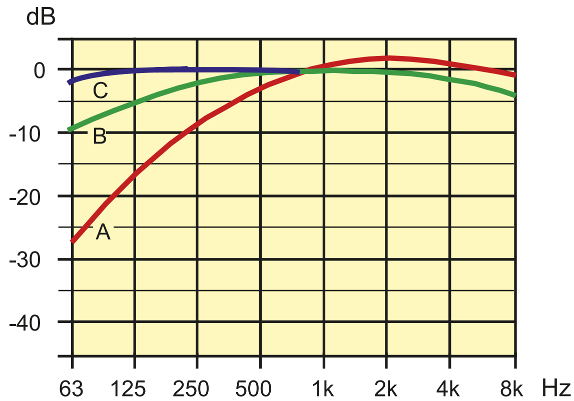
NR curves
Lp (dB) = Sound pressure level
A = Mid-frequency (Hz)

Octave band	Mid-frequency	Band limits	Wavelength
No	Hz	Hz	m
2	125	88-177	2.720
3	250	177-354	1.360
4	500	354-707	0.680
5	1000	707-1410	0.340
6	2000	1410-2830	0.170
7	4000	2830-5660	0.085
8	8000	5660-11300	0.043

Recommended octave bands according to ISO

Mid-frequency	Filter A	Filter B	Filter C
Octave band	(dB)	(dB)	(dB)
Hz			
63	-26.2	-9.3	-0.8
125	-16.1	-4.2	-0.2
250	-8.6	-1.3	0
500	-3.2	-0.3	0
1000	0	0	0
2000	+1.2	-0.1	-0.2
4000	+1.0	-0.7	-0.8
8000	-1.1	-2.9	-3.0
16000	-6.6	8.4	-8.5

Weighting filters for sound measurements



Attenuation for different weighting filters

When the dB(A) value is calculated, the measured values are corrected with the A-filter's approximate values as set out in the table. The resulting sound levels by octave band are then added logarithmically.

Example

A sound measurement gives the sound pressure levels 45 dB (125 Hz), 40 dB (250 Hz), 36 dB (500 Hz), 37 dB (1000 Hz), 34 dB (2000 Hz) and 25 dB (4000 Hz). The resulting sound level expressed in dB(A) will then be 41 dB(A).

Frequency	125	250	500	1000	2000	4000
Measured values	45	40	36	37	34	25
Corr. for filter A	-16	-9	-3	0	+1	+1
Result	29	31	33	37	35	26

Correction for A-filter

Sound attenuation

Unit attenuators
Unit sound attenuators are usually used on air handling units.

Sound attenuators resistive
The most common type is an absorption attenuator where the air flows along the sound absorbing material; the longer the sound attenuator, the greater the attenuation. Resistive sound attenuators provide better attenuation at higher frequencies. The attenuation is stated in dB in octave bands, which corresponds to the attenuation that is obtained if the corresponding duct is replaced with the attenuator. Sound attenuators in elbows give efficient damping.

Sound attenuators reactive
A reactive sound attenuator can give good attenuation at low frequencies if the volume is large enough. An example of a reactive sound attenuator is a pressure box, lined inside with sound absorbing material. The sound energy is assumed to be uniformly distributed over the surface and attenuation corresponds proportionately to the size of the opening relative to the entire interior surface. Preferably the inlets and outlets must not be placed opposite each other as high frequency sound can be short-circuited through the sound attenuator.

Branches
Normally, you can distribute the sound energy in the duct system's different branches in proportion to the area size. An approximate method is to distribute the sound energy in proportion to the air distribution in the system. There are deviations, for example, duct branches in straight connections close to the fans that can give more sound at certain frequencies than what this approximate method states. However, with care, the method gives a good estimate of the system's sound attenuation properties.

Room absorption

The room's volume, condition of the surfaces and interior furnishings significantly affect the resulting sound level. The table with reference values for the absorption factor α and the diagram can be used to calculate a room's equivalent sound absorption area.

It generally applies that the equivalent sound absorption area (A) is calculated as follows:

$$A = S \cdot \alpha_m$$

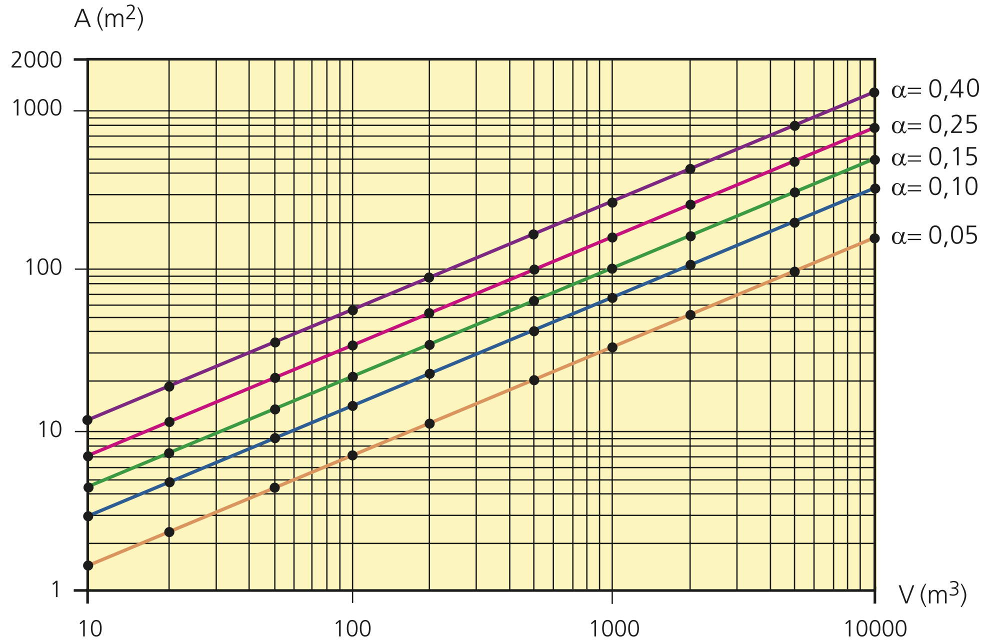
where $S \cdot \alpha m = S \cdot \alpha 1 + S2 \cdot \alpha 2 + \ldots + Sn \cdot \alpha n$

S= the room's total limitation area (m^2)
S1...Sn= area of sub-surfaces (m^2)
α1...αn= absorption factors of sub-surfaces
α_m = mean absorption factor for the total limitation surface

Type of room	Mean absorption factor α_m
Radio studio, music room	0.30 - 0.45
TV studio, store, reading room	0.15 - 0.25
Homes, offices, hotel rooms, conference rooms, theatres	0.10 - 0.15
Classrooms, nursing homes, small churches	0.05 - 0.10
Factory halls, swimming pools, large churches	0.03 - 0.05

The reference values for different premises' mean absorption factor.

Example
A local shop for clothes with dimensions of 20 x 30 x 4.5 m (i.e. 2700 m^3) has a mean absorption factor of α_m = 0.40. The premises' equivalent room absorption will be 500 m^2.



Equivalent sound absorption area
A = Equivalent sound absorption area (m^2)
V = Room volume (m^3)

Sound attenuation in duct openings

The sound level at the inlet from the duct to the room is reduced through the sound's propagation, which normally occurs radially. This reduction is called orifice attenuation.

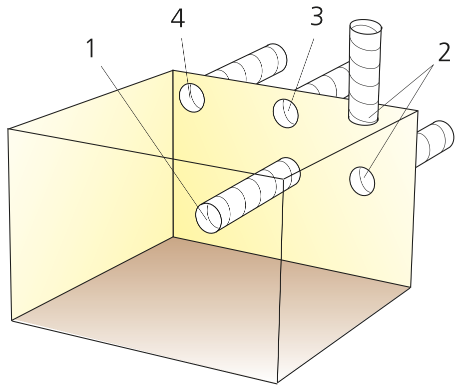
The attenuation (ΔL) for each air terminal is specified in octave bands. The orifice attenuation is calculated in these values. The diagram below is used to calculate orifice attenuation for a freely ended duct.

Example

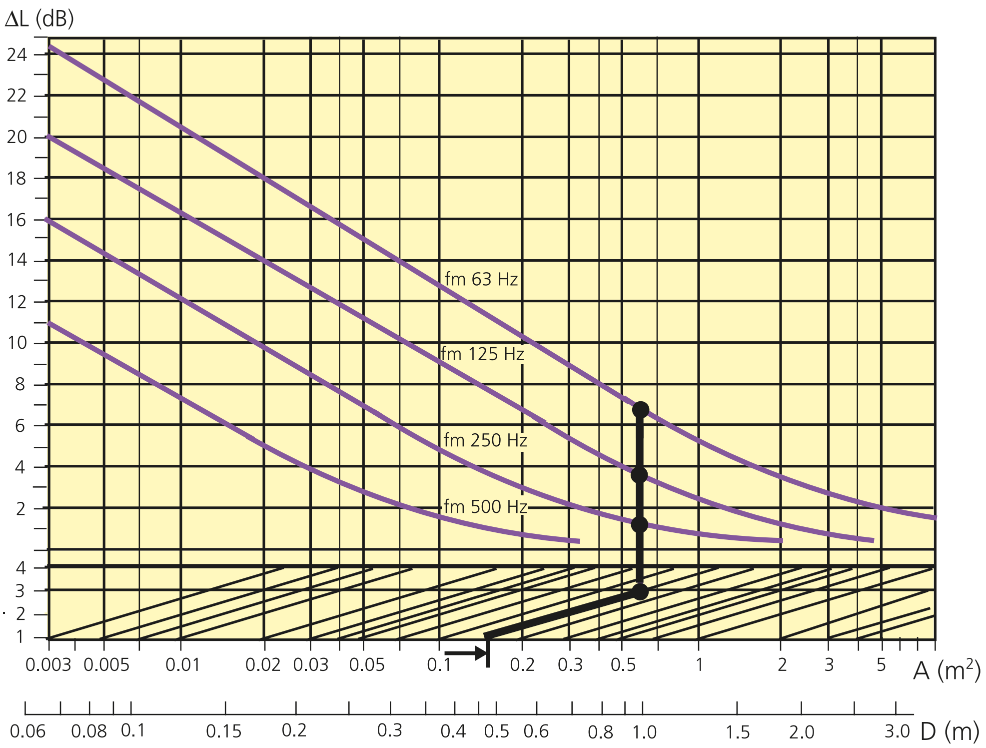
A rectangular duct opens into the room according to placement option 3 in the illustration to the right, and its cross-sectional area is 0.15 m^2.

According to the diagram, the following orifice attenuation is obtained.

Octave band (Hz)	63	125	250	500
Orifice attenuation dB	7	4	1	0



Placement of the duct opening
1. Middle of the room
2. Wall or ceiling
3. Wall near the ceiling
4. Corner

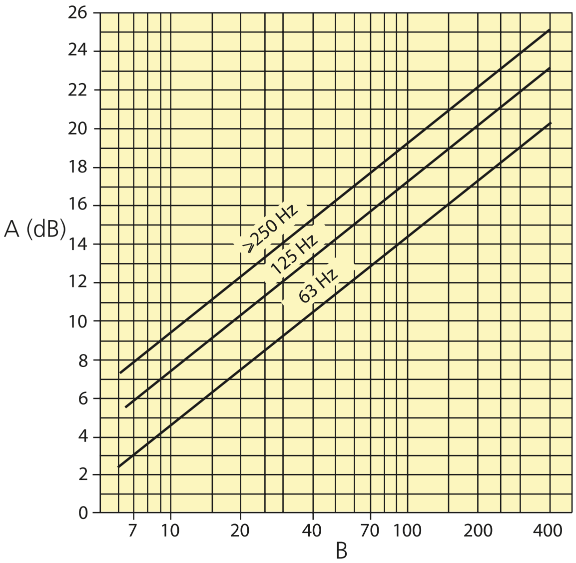


Sound attenuation
ΔL = Attenuation (dB)
B = Placement of the duct opening (see illustration above to the right)
A = Cross-sectional area of the duct with rectangular cross section (m^2)
D = Opening diameter for ducts with circular cross section (m)

Attenuation in interior suction or pressure chambers

If the sound on its way from the fan to the room passes an interior lined chamber, the sound will be attenuated in proportion to the difficulty the sound has in leaving the chamber.

The following figure can be used to determine the attenuation in a chamber that is internally lined with 100 mm mineral wool.

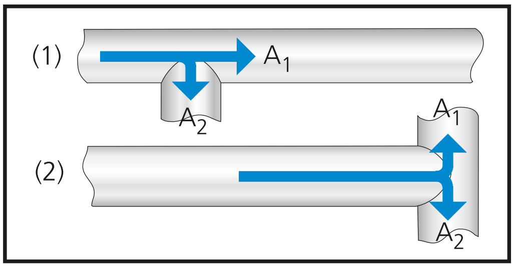


Attenuation in pressure and suction chambers internally lined with 100 mm thick mineral wool. (Note! The inlet and outlet must not be placed opposite to each other.)
A = Attenuation (dB)
B = Lined area divided by the outlet area.

Branch attenuation

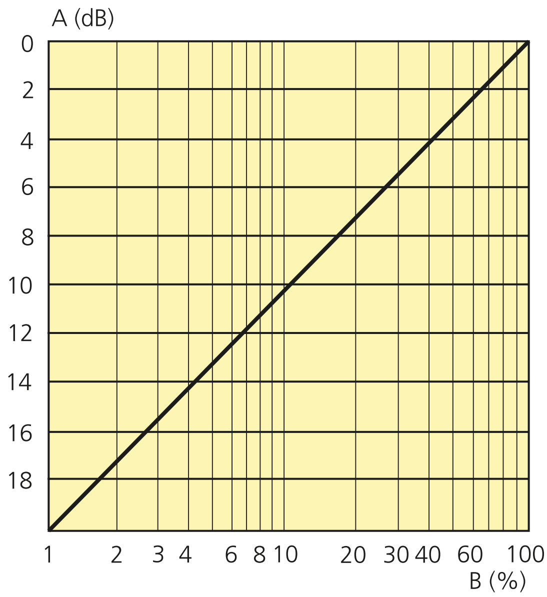
At a branch the sound effect separates in relation to the duct areas, i.e. (A1/A2 as shown below). In cases where the air velocity in all ducts is relatively similar, the sound effect will divide in the same way as the air volume. A branch duct that transports 10% of the total air volume will also contain 10% of the sound effect.

As high frequencies can be compared to rays of light, in the illustration below we see that only a small part of the high frequency sound propagates in the branch. Thus, in this case, we underdimension the attenuation at high frequencies (> 500 Hz). However, in a T-pipe, the sound energy divides itself according to the same relationship as the duct areas.



Sound propagation in branches

The diagram below can be used both when looking at the relationship between air volume and area.



The relationship between the % branched air and attenuation
A = Attenuation (dB)
B = Air volume to room/terminal (%)

Duct attenuation

Only small attenuation is obtained in the duct system apart from elbows, bends and changes in size where so-called reactive attenuation is obtained.

Integral attenuation

The attenuation of terminals is specified in dB in the octave bands and usually refers to the attenuation from the sound effect in the duct to the sound level in the room at 10 m^2 Sabine room attenuation. There are other presentation, so be wary of misunderstandings! Also note that some terminals act as amplifiers, i.e. the integral attenuation can be negative.

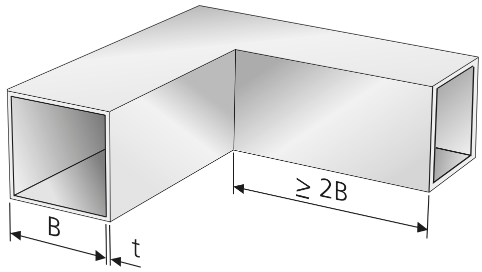
The attenuation for larger terminals designed for industrial applications is specified for 150 m^2 room attenuation.

Attenuation in elbows

When the sound in a duct hits an elbow, some of the sound is reflected back. The amount reflected depends on the duct's dimensions and the wavelength.

The attenuation in rectangular elbows is significantly greater than for circular elbows. Normally attenuation is not calculated in circular elbows. Attenuation for rectangular elbows are shown in the table below.

The lining must have a length corresponding to the minimum duct width; thickness of the lining material, t, must be at least 0.1 × B.



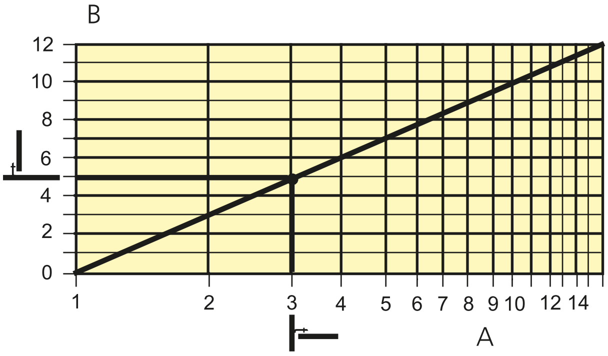
Internally lined elbows with damping

Attenuation in rectangular elbows with and without absorption linings

	Duct width	Attenuation in dB at octave mean frequency (Hz)						
	mm	125	250	500	1000	2000	4000	8000
Without absorption lining	125			1	5	8	4	3
	250		1	5	8	4	3	3
	500	1	5	8	4	3	3	3
	1000	5	8	4	3	3	3	3
Lining before the elbow	125			1	5	8	6	8
	250		1	5	8	6	8	11
	500	1	5	8	6	8	11	11
	1000	5	8	6	11	11	11	11
Lining after the elbow	125			1	7	11	10	10
	250		1	7	11	10	10	10
	500	1	7	11	10	10	10	10
	1000	7	11	10	10	10	10	10
Lining before and after the elbow	125			1	7	12	14	16
	250		1	7	12	14	16	18
	500	1	7	12	14	16	18	18
	1000	7	12	14	16	18	18	18

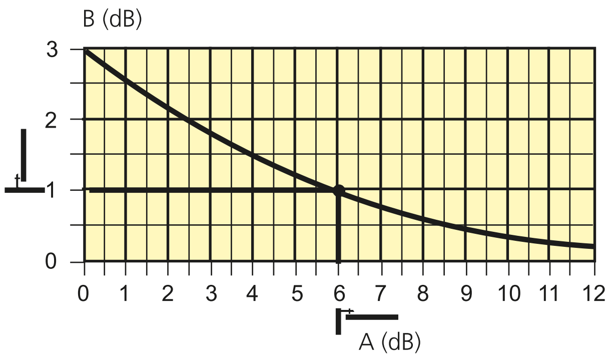
Addition of sound levels

All sound sources in the room in question are added logarithmically. Addition can be done by using the diagram below for the addition of a number of equal or different sound sources.



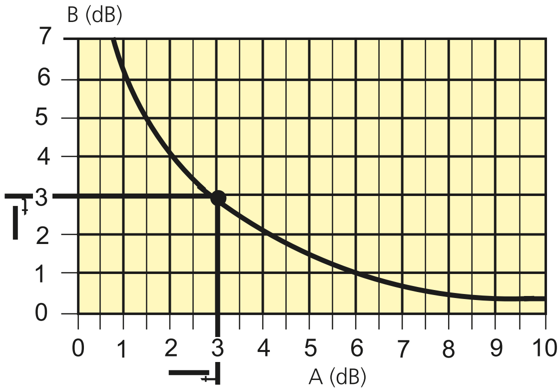
Logarithmic addition of several equal levels
A = Number of sound sources
B = Increase to be added to the level of the sound source, dB

Example
In a room there are three extract air diffusers which each give 25 dB(A).
Together these give a sound pressure level of 25 + 5 = 30 dB(A).



Logarithmic addition of two different levels
A = Difference between levels to be added (dB)
B = Increase to be added to the higher level (dB)

Example
Total of 30 dB(A) and 36 dB(A) is 37 dB (A).



Logarithmic subtraction of two different levels
A = Difference between total level and the level from sound source 1 (dB)
B = Reduction to be subtracted from the total noise level (dB)

Example
In a room with both supply and extract air systems the total noise level is 35 dB (A). Only the supply air system gives 32 dB(A). The difference is 3 dB(A), which means that the extract system gives 35 - 3 = 32dB(A).

Basic facts
All sound sources in the room in question are added logarithmically.

Mathematically layout for logarithmic addition and subtraction if character change occurs:

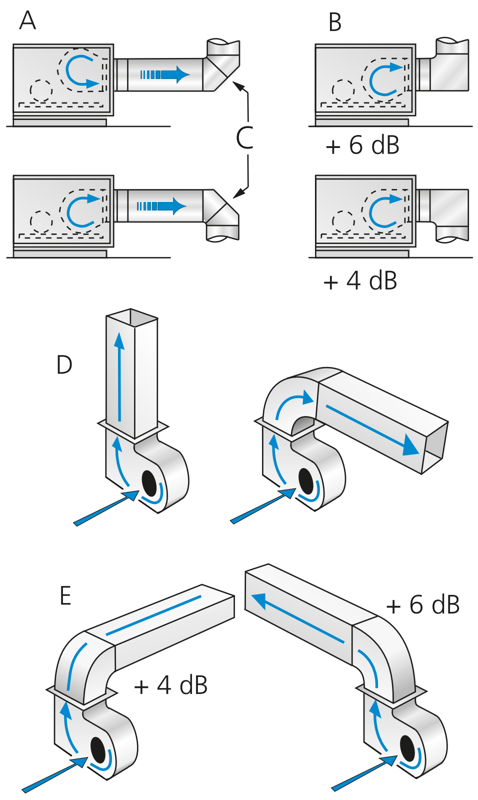
$L_{Atot} = 10 \text{ Log } (10^{(LA1/10)} + 10^{(LA2/10)} + ...)$

Design tips for sound

Duct connections to fan outlets

The duct's connection to the fan is the first place that may generate unnecessarily high pressure drops and thus unnecessarily high sound levels. When deflecting the air via an elbow, consideration must be taken to the velocity distribution in the duct before the bend.
An incorrect design with a cross elbow directly next to a traditional radial fan with scroll-shaped housing will increase the total sound power level by 4 dB. In addition, if the fan is turned “upside and down” the sound level will be about 6 dB higher than with a correct design. So-called plenum fans are much more forgiving.

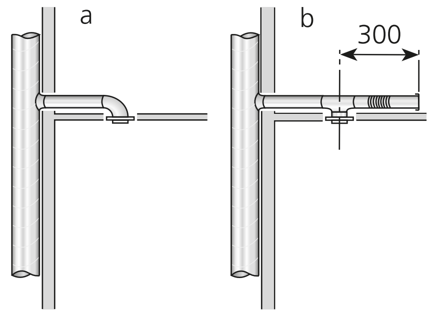
Some examples of right and wrong design and examples of noise increases that can result are given below.



Examples of the right and wrong duct design after a fan
A = Correct design
B = Wrong design
C = Elbow angled 45 degrees on the outside
D = Correct design - Dust elbow should be aligned with the fan's direction of rotation.
E = Wrong design

Sound attenuators for extract air diffusers

One way to counteract crosstalk via a common ventilation duct is to fit a sound attenuator between the main duct and the terminal. Placing the sound attenuator as shown in the figure, can remove the resonance effects and thus significantly improve the sound insulation.



Counteracting crosstalk
a) Low sound insulation
b) High sound insulation. High attenuation can be obtained especially at duct lengths of 1-3 m.

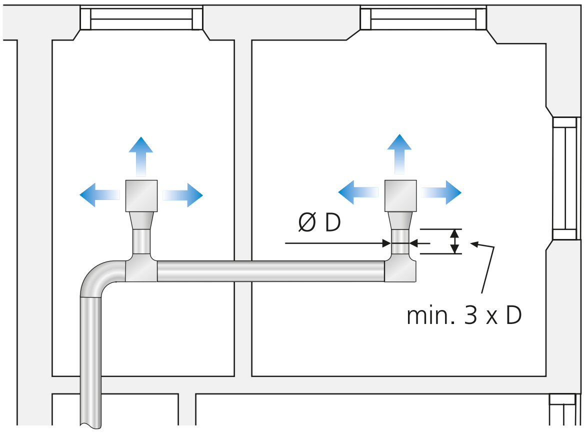
The same procedure can be used for supply air diffusers to prevent crosstalk and attenuate duct sound.

Selection of air terminal

Terminal selection should be done so that the terminal’s sound generation is 5 dB below the requirement specified for the room in question.

Distance between the duct and terminal

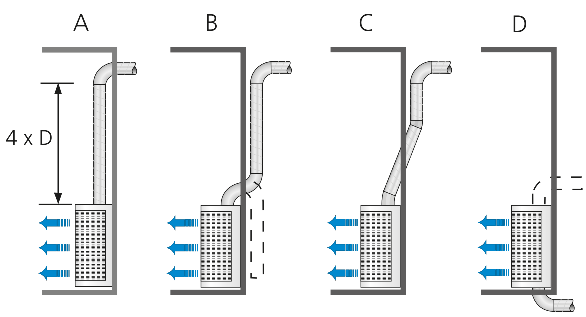
The specified values in different types of documentation regarding pressure drop and sound generation apply on the condition that the velocity distribution in the terminal connection is balanced. A common error is that the terminal is placed too close to the branch duct with accompanying sound problems. One recommendation is that the branch of the duct is at least 3 times the diameter of the duct. See the example below.



The branch from duct to the terminal should be at least 3 x D.

Duct routing with low speed terminals

Duct routing is extremely important for the size of the sound generation. Duct elbows directly before the terminal can give substantial increases in sound generation. See the example in the illustration and table below.



Example of how different connections to the terminal affect sound generation

Sound effect at duct connections

	Duct connections			
v m/s	A	B	C	D
4-5 m/s	+ 2	+ 6	+ 3	+ 3
6-8 m/s	+ 4	+ 10	+ 6	+ 6

Sound effect (dB) for different duct connections and different air velocities in the connecting duct

Calculation model dB(A) to dB(C)

The model shows an example with a supply air diffuser for displacement ventilation, which at a flow of 170 l/s with open damper gives 30 dB (A) according to the sizing diagram.

Sound power level
Correction factor, K_{OK}

	Mid-frequency (octave band) Hz								
	63	125	250	500	1000	2000	4000	8000	
dB(A)	30	30	30	30	30	30	30	30	
K_{OK}	12	8	4	2	0	-10	-22	-23	
L_W	42	38	34	32	30	20	8	7	
Plus C-filter	-0.8	-0.2	0	0	0	-0.2	-0.8	-3	
Result B(C)	41.2	37.8	34	32	30	19.8	7.2	4	44
	Room attenuation at 10 m² Sabine.								4
	Result								40

Result:
At 170 l/s with the damper in the open position gives the same as the supply air diffuser
40 dB (C).

Designing for fans

A fan is intended to provide flow transport of air or other gas.

For such a flow to take place, for example, in a duct system, requires a pressure increase of the gas in a suitable location in the system. The requisite pressure rise can be accomplished with a fan or - in instances when a particularly large pressure increase is required - with a compressor.

Explanation of terms used in the equations below:

q = volume flow at the fan inlet .... m^3/s (m^3/h)

Δp_t = total pressure increase between the fan's connections .... Pa (mm vp)

n = fan speed .... rpm

P_u = theoretical power .... kW

P_r = fan impeller's power load................................. kW

P_e = consumed active power effect from the network kW

η_r = fan impeller's efficiency..................................... %

η_e = total efficiency for the fan................................. %

Definition of efficiencies for fans

Fan impeller efficiency:

$$\eta_r = \frac{P_u}{P_r} \times 100\%$$

The fan's total efficiency

$$\eta_r = \frac{P_u}{P_e} \times 100\%$$

where P_u is the theoretical effect according to

$$P_u = \frac{q \times \Delta p_t}{1000} \quad \text{kW}$$

The speed's effect on the fan power

In the event of unchanged load conditions (unchanged throttling) changes:

1. The air volume in direct proportion to the speed

$$\frac{q}{q_1} = \frac{n}{n_1}$$

2. Static, dynamic and total pressure in direct proportion to the square of the speed change

$$\frac{p}{p_1} = \left(\frac{n}{n_1}\right)^2$$

3. The power load in direct proportion to the cube of the speed change.

$$\frac{p}{p_1} = \left(\frac{n}{n_1}\right)^3$$

Working method

Flowing gaseous mass is supplied energy in a fan through one or more vaned-equipped fan impellers. When passing through the fan impeller(s) both the gas's dynamic and static pressure usually increase.

The outlet velocity from the impeller is transformed usually to static pressure during the passage from the impeller outlet to the fan outlet.

In radial fans this transformation from velocity energy into static pressure takes place in the spiral casing. Fans that are connected to the duct system generally have the same connection area on the inlet and outlet. In such instances, the gas velocity and thus the dynamic pressure is equal in the fan connection and the fan's total pressure increase is only perceived as an increase of static pressure between the fan's connection flanges.

However, a free-drawing fan draws air from a room, where both the static pressure and the velocity is 0, and delivers the air in the fan outlet at a specific velocity and increased static pressure. Accordingly, in this case the fan's total pressure increase is perceived as an increase of both the static and dynamic pressure.

Fan data at differing density

The diagram and data for fans specified by manufacturers of fans and air handling units usually apply for a density of 1.2 kg/m³ at the fan inlet.

The density is 1.2 kg/m³ for air with a temperature of 20°C at relative humidity of 50% and at sea level (1013 mbar). The following relation applies for recalculation of fan data to another density.

1. The air flow in m³/s does not vary with density.
2. Static, dynamic and total pressure is taken from:
   $p = p_{1.2} \times K_1 \times K_2$.
3. Power load is taken from:
   $P = P_{1.2} \times K_1 \times K_2$
4. The density is taken from:
   $\rho = 1.2 \times K_1 \times K_2$
   where K_1 and K_2 is taken from the diagram opposite.

In many contexts normal cubic metres, nm³, or normal cubic metres per second, nm³/s are used.

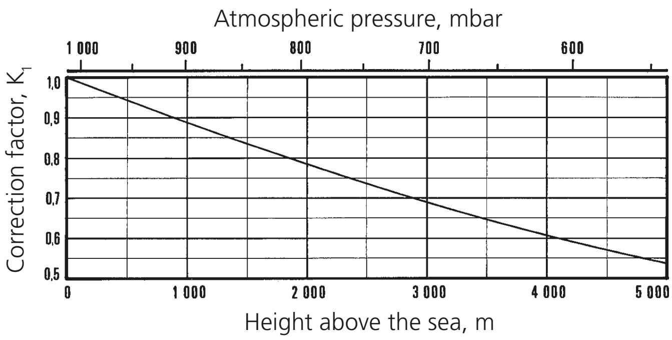
Normal cubic metres, nm³, refers to the quantity of gas which at a pressure of 1 bar and a temperature of 0°C, has the volume 1 m³.

Accordingly, airflow expressed in nm³/s is constant independent of whether the air is cooled or heated. Conversion from airflow expressed in nm³/s to actual airflow in m³/s is as follows:

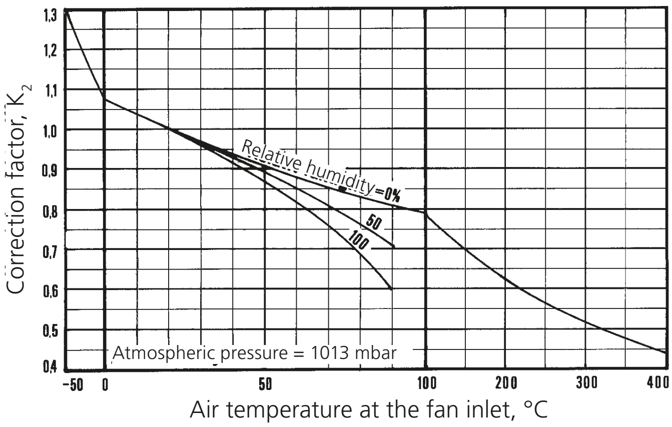
$$q = q_n \times \frac{1.06}{K_1 \times K_2}$$

where q_n is the airflow in nm³/s.

Correction factor, K_1



Correction factor, K_2

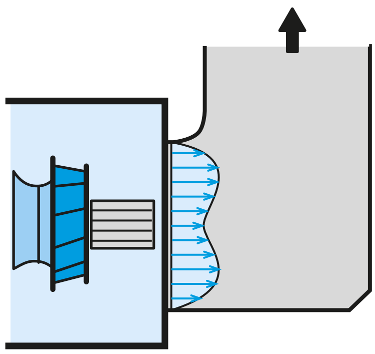


System losses fans

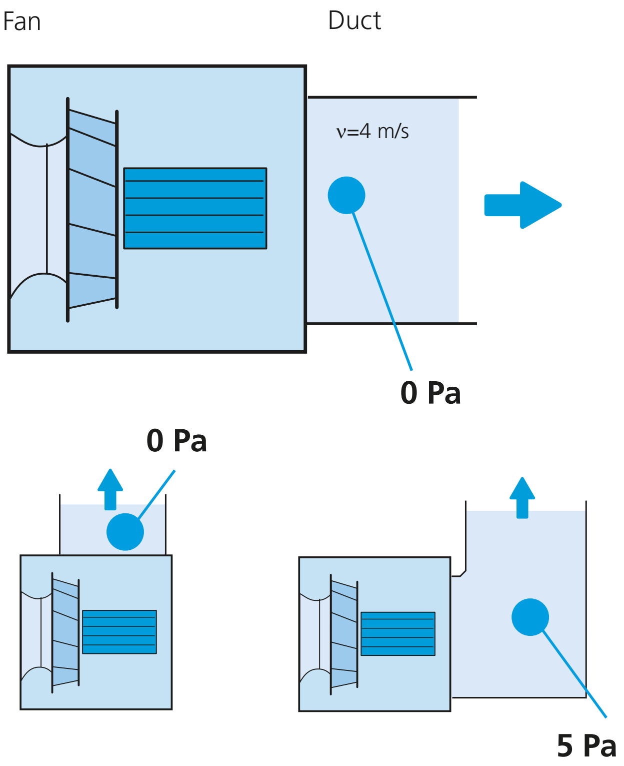
System losses plenum fans

Fans with an even spread pattern and low outlet speed minimises system losses at the duct connection.

A sharp duct elbow fitted directly to the outlet only gives rise to the elbow's normal pressure loss of approx. 5 Pa. You can even choose outlets upwards without extra pressure loss.



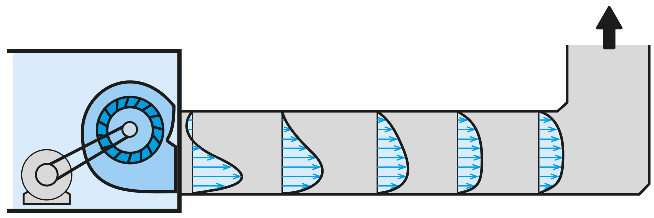
A plenum fan has a low air velocity and gives an even distribution pattern immediately after the outlet.



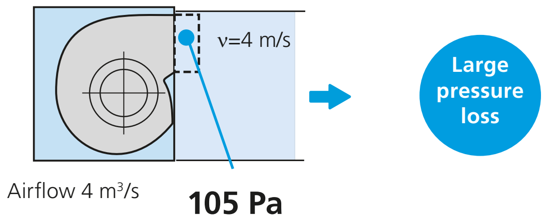
System losses for radial fans

Radial fans work at higher air velocities. The air is led out through a "scroll-shaped housing", which means that there will be an uneven distribution pattern.

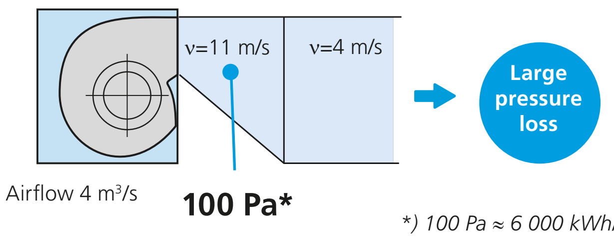
This means that sharp duct elbows directly on the outlet give large pressure losses. The only way to reduce the pressure losses is to have a long straight duct before the elbow.



Conventional radial fan with air distributor

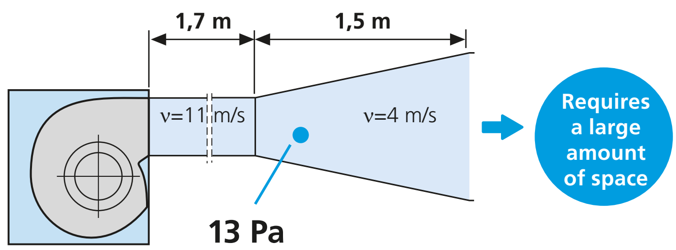


Conventional radial fan with straight duct and diffuser



**) 100 Pa ≈ 6 000 kWh/year.*

Conventional radial fan with a transition



Airflow 4 m³/s

Project design for duct systems

For mechanical ventilation the system is usually divided up:

- Extract air system
- Supply air and Extract air systems
- Supply air and Extract air systems with energy recovery

Today, as the energy recovery systems is virtually a matter of course, henceforth only this will be described and specified in illustrations etc.

Design

As it costs energy to power a ventilation system, it is desirable that the system is as energy efficient as possible. The system must therefore have certain characteristics. A good design is necessary to produce a duct system that is energy efficient, has low sound levels and minimal leakage.

The duct system should be designed for the lowest possible frictional pressure drop, which means that it must be as aerodynamic as possible and cause a minimum of air vortices. In this way, the sound levels will also be low.

As energy costs, you ensure that air is not lost through leakage. Consequently, you should strive to produce an as airtight system as possible. You should therefore as far as possible use type approved ducts with sealing components, primarily circular ducts and duct fittings.

In order to achieve good overall economy, the system should also be installation-friendly and easy to adjust.

The optimal duct system

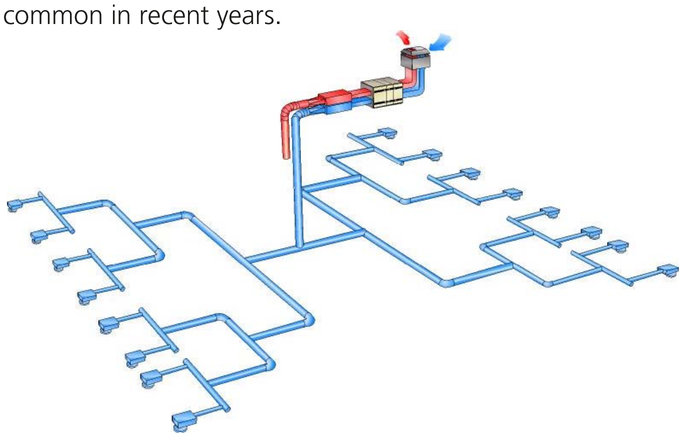
The ideal duct system should basically be symmetrical and divided into sub-systems with short ducts. This design simplifies commissioning. The conditions for building a symmetrical duct vary from case to case, but if you consider symmetry during the design phase, it usually produces a better result.

Reality in many cases

It is quite common that the duct system is designed with long ducts with many branches, which means that there will be large pressure drop difference between the first and last terminals on the main pipe work, which in turn gives rise to sound problems. The smaller the pressure that needs to be throttled in the terminal, the quieter the system. A system with long duct routing and many branches are also very difficult to trim. Preferably use branch dampers in the system for improved air distribution.

Ring feed system - good option

In order to balance out the pressure conditions you can design the duct system for ring feeding. It is then important to remember that the manifold duct should be the same size all the way round, as it will then be easier to maintain the same pressure throughout the manifold duct. If you also have terminals connected with a relatively high pressure drop, this is a simple solution. Ring feed systems in the ventilation industry have always existed, but has become increasingly common in recent years.



An optimal duct system is symmetrical and divided into sub-systems.

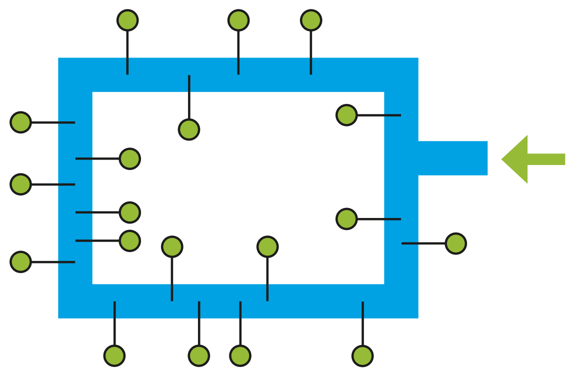


Illustration of a ring feed system

Sizing

How should we design a duct system? What should you start with?

Airflow

You always start out from the different room functions in the building and determine the supply and extract air flows for each room. Adding together these flows produces the total air flow that the air handling unit must be able to produce in order to provide the building with the right amount of air. The air flow is stated in m^3/s, l/s or m^3/h.

In order to size the duct system correctly you need to take into account the air velocity, duct area and the frictional pressure drop.

Air velocity, duct area and frictional pressure drop

Using the air flows in the ducts and the duct area, you can calculate the velocity at which the air moves through the ducts. However, you must also take into account the frictional pressure drop as the air is enclosed in a duct system. This makes it harder for the air to reach the room, but on the other hand, it ensures that it finds the right room. The frictional pressure drop is measured in the unit P, Pascal.

The frictional pressure drop is in ducts with its elbows and branches, but also in the unit, sound attenuators, dampers, air terminals, etc.

The design conditions for the ventilation system's heart, the air handling unit, are taken from the combination of the total air volume and the total pressure loss. The total pressure drop is often divided into unit + duct system.

We must therefore make a calculation for the supply air system and one for the extract air system.

Today there is widespread access to computerized calculation software, which is why a detailed description of the calculation process is not given here. Basically it is a question of locating the heaviest distance between the unit and the end terminal, which is not always, but usually is the longest route. All partial pressure drops along this stretch are added - pressure drop in the duct, in elbows, branches and other duct components, pressure drop in air terminals, internal pressure drop in the unit and in other duct-mounted components, for example, sound attenuators. You can also possibly add a factor for leakage. This factor can vary between 5 and 15% depending on the air-tightness class.

Basic facts

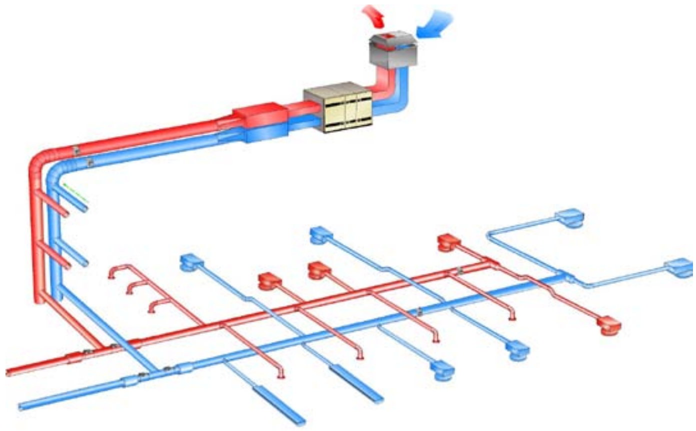
A fan's additional energy is a total pressure P_t, which is composed of a static pressure P_s (potential energy) and a dynamic pressure P_d, (kinetic energy).

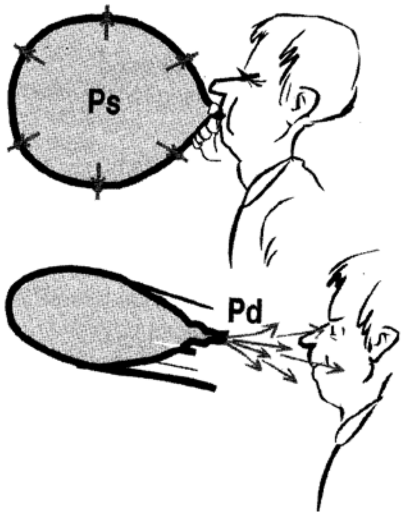
Airflow q (m^3/s)
Duct area a (m^2)
Velocity v (m/s)

$q = v \cdot A$ m^3/s
$v = q / A$ m/s

Pressure drop (Pa)

Total pressure $p_t = p_s + p_d$





P_s = Static pressure

P_d = Dynamic pressure

Recommended air velocities

What velocities must be maintained to produce an energy efficient system?

The Swedish Indoor Climate Institute recommends the velocities below to produce such a system.

The recommendation says 0.8-1.2 Pa/m.

Since the circular ducts usually have a braced inside with reinforcement grooves from size 250, the table shows the consequences of this bracing.

Recommendations of this kind can of course vary between countries, but this is nevertheless a strong guideline in terms of just electrically efficient systems.

Ø d mm	Smooth inside	Braced inside
	v m/s q l/s	v m/s q l/s
100	2.8 22	
125	3.2 40	
160	3.7 75	
200	4.3 135	
250	5.0 250	4.4 220
315	6.0 470	5.1 400
400	7.0 880	5.6 700
500		7.7 2.4
630		11.4 14.0

Project design for mixing air ventilation

Ventilate with the right type of terminal

Characteristics that are normally important for supply air diffusers designed for mixing ventilation are:

1. High co-ejection of the room air so that the low supply air temperatures can be utilized.
2. Short throw lengths for ceiling and wall diffusers without air dropping from ceiling and entering the occupied zone to early resulting in draught problems.
3. Ability to supply large airflows without the throw lengths becoming too long.

One way of satisfying the characteristics 1 and 2 above is to ensure that:

- The outlet velocity of the air from the diffuser is high, which, means that the terminal's outlet area (A_0) must be small.
- The terminal constant (k) must be low.
  The same principles apply to request 3 above, however, with the limitation that there is a conflict between small A_0 and large airflow. As the throw length is proportional to the expression: $k/\sqrt{A_0}$ the terminal constant must be small so that the throw length is short. Simultaneously, the outlet area must be as large as possible. Conflicts arise then with the need of high co-ejection, which is proportional to the expression:

$x/k \cdot \sqrt{A_0}$ where x = the distance from the terminal.

The so-called rotation diffuser (swirl diffuser) is designed specifically to give a low terminal constant and relatively high outlet velocity. Characteristic for typical rotation diffusers is the capacity restriction compared to e.g. perforated supply diffusers.

In order to produce the lowest possible terminal constant, requires the supply air to be supplied via ceiling diffusers and that it is distributed evenly over the terminal's whole outlet area. The distribution angle must be 360° for the lowest terminal constant.

The traditional rotation diffuser is designed to supply the air via a number of long, rectangular slots, which are usually arranged radially in a circular shape. The principle of supplying air via a number of slots allows the outlet area to be limited and a high velocity can be maintained. One disadvantage of slots is that flexibility is limited in terms of the ability to provide variations in the distribution pattern.

Nozzle diffusers give flexibility

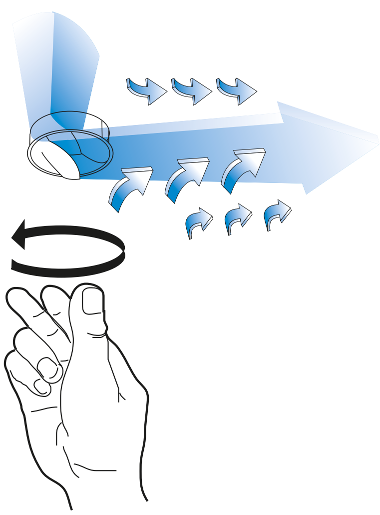
The method of continuously adjustable round nozzles, provides in this respect much greater flexibility. Different characteristics, and opportunities, are obtained with different numbers of nozzles. Even the size of the nozzles affects the ventilation options. The more and the smaller the nozzles in a diffuser, the greater the variation.

The distribution patterns that can be easily produced with the nozzles can be varied infinitely. The following variants can be easily be produced:

- All-round distribution
- 1-, 2-. 3 and 4 way distribution.
- Tangential distribution
- Vertical distribution
- Simultaneous vertical and horizontal distribution

The tangential distribution can be carried out in different ways. The most common is that all the nozzles are directed in the same way, i.e. either clockwise or anticlockwise. This setting gives the highest co-ejection. If you prioritise a short throw length the nozzles can be adjusted so that an impulse loss is obtained. If the nozzles are arranged in rings, every other nozzle ring can be directed anticlockwise and every other clockwise. This gives rise to a sharp impulse loss and with that a shorter throw length.

Other factors of great importance, which are rarely discussed, include the supply air's outlet direction just as the air leaves the diffuser (nozzle). In order for the impulse loss not to become too large, the air must have an outlet direction, which is parallel to the ceiling surface, as shown below. This is important in cases where large temperatures below room temperature occur. Use of rotation diffusers is motivated precisely because they suit large for temperatures below room temperature. If the nozzle or the slot does not meet this requirement, for large temperatures below room temperature, the thermal forces may become dominant compared to the Coanda effect. Undesirable flow patterns can then easily ensue.



Principle for adjustable, round nozzles.

Characteristic properties of different terminal types

The following table shows a basic overview of the most common terminal characteristics:

Terminal type	Characteristics
Nozzle diffuser, ceiling mounting	Installation flexibility, short and long throw lengths can be obtained. Both vertical and horizontal distribution patterns can be obtained. Large temperatures below room temperature can be utilized. Large variation range for airflow without air “dropping”. Manages large temperatures below room temperature.
Nozzle diffuser, wall mounting	Flexible installation Distribution to the side and/or forwards. Short or long throw lengths.
Guide vane perforated ceiling diffuser	Large temperatures below room temperature can be obtained. Fixed distribution patterns. Flush mounting in ceilings. Air flow capacity is generally higher than in nozzle diffusers.
Guide vane perforated wall diffuser	Large temperatures below room temperature can be obtained. Fixed distribution patterns. Air flow capacity is generally higher than in nozzle diffusers.
Perforated ceiling diffuser	Relatively large temperatures below room temperature can be obtained, however, less than for nozzle diffusers. Short throw lengths on account of the impulse loss obtained by the perforation. Different distribution patterns possible through mechanical devices. Suitable for large airflows.
Perforated wall diffuser	Relatively large temperatures below room temperature can be obtained, however, less than for nozzle diffusers.
Linear slot diffuser in the ceiling	Limited flexibility in the distribution pattern. Long throw lengths despite high co-ejection.
Circular slot diffuser in the ceiling (conical diffuser)	High co-ejection at narrow slots due to favourable conditions between terminal constant and outlet area. Relatively large temperatures below room temperature can be utilized. Limited flexibility in the distribution pattern.

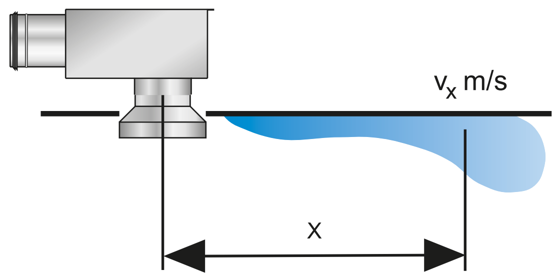
Throw lengths

General

According to the Swedish VVS-AMA (General gudielines) the throw length should be specified with a final velocity in the air jet of 0.2 m/s ($l_{0.2)}$. To calculate other end velocities, refer to the manufacturer's calculation program.

Conversion

Higher air velocity can be accepted, for various reasons, when a supply air jet reaches the occupied zone or hits an obstacle, for example, a wall. The air velocity can, within a limited area in the air jet, be calculated as shown below.



$$x = l_{0,2} \cdot \frac{0,2}{V_x}$$

The calculation of the air velocity at distance x from the diffuser

x = distance in metres from the diffuser to the point in the air jet where the air velocity is V_x m/s
V_x = the air velocity at the distance x from the diffuser

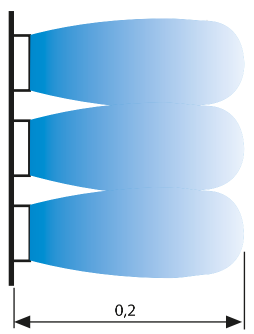
Example

An air diffuser has a throw length $l_{0.2}$ = 3 m. The throw length $l_{0.3}$ then becomes:

$l_{0.3}$ = · 02/03 = 2 m

Combining supply air jets

When two or more supply air diffusers are placed so close together that the jets combine the throw length is extended. For the calculation of this extension, use a calculation program.



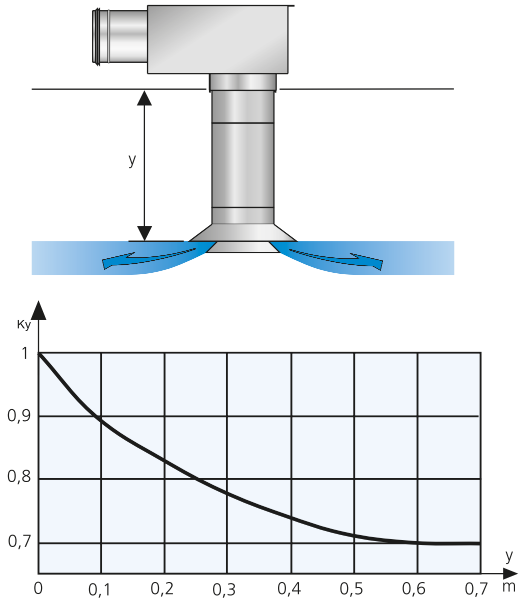
Combining supply air jets

Deviating installation

The specified throw lengths for slot air diffusers, conical diffusers and perforated diffuser apply for ceiling mounting. If the supply air diffusers are suspended, and the jet is directed so it does not adhere to the ceiling, the throw length is reduced due to the co-ejection that can occur on both sides of the supply air jet.
The following conditions apply with suspended installation of supply air diffusers where the jet does not adhere to the ceiling:

$l_{0.2}$ suspended = $k_y \cdot l_{0.2}$

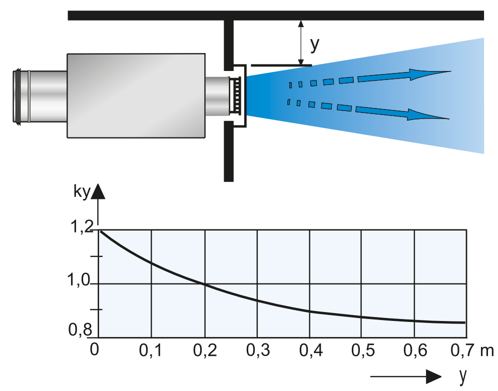
where k_y = correction factor depending on the distance, y, between the diffuser and ceiling



Correction factor k

For grilles, the specification applies to wall jets, i.e. with adhesion to the ceiling. If the grille is installed more than 0.2 m from the ceiling the throw length decreases.

The following conditions apply when wall mounting the grille when the grille is more than 0.2 m from the ceiling:
$l_{0.2}$ close to the ceiling = $k_y \cdot l_{0.2}$ where k_y = correction factor depending on the distance, y, between the diffuser and ceiling.



For wall mounted grilles, where the throw length is measured for diffusers mounted at a distance of 0.2 m from the ceiling, the diagram above applies ($l_{0.2}$) for other distances between the grille and ceiling.

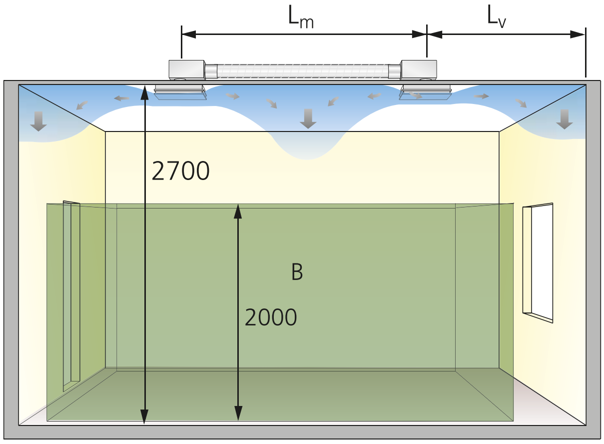
Minimum distance between supply air diffusers

The minimum distance between two supply air diffusers, which have air jets directed towards each other, may be shortened due to the core jets' final velocity can be permitted to be higher in the mixing point, without the total jet velocity in the occupied zone exceeding 0.2 m/s. This is due to a strong mixture of the two air jets taking place and their velocities are slowed down. The following relationship applies:

$L_m = k_v (l_{0.2}$ diffuser 1 $+ l_{0.2}$ diffuser 2)

L_m = Minimum distance between supply air diffusers.
k_v = Correction factor, according to the diagram on the right.

Example
Two supply air diffusers, each with a throw length $l_{0.2} = 5.0$ m get a minimum distance at a temperature below room temperature of 6 °C on $L_m = 0.72\ (5.0 + 5.0) = 7.2$ m.



Minimum distance L_m between supply air diffusers.
B = Occupied zone

Minimum distance between supply air diffusers and the wall

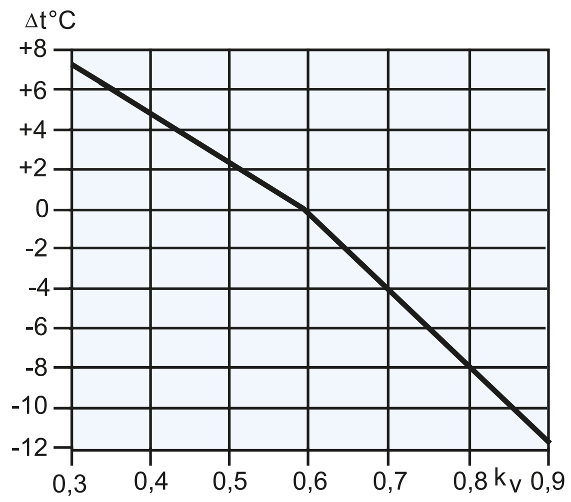
An air jet, that hits a wall, is permitted to have a higher speed than 0.2 m/s because of the deceleration and deflection, which then occurs. The following relationship applies:

$L_v = k_v \cdot l_{0.2}$

L_v = Minimum distance between supply air diffuser and wall.
k_v = Correction factor, according to the diagram on the right.

k_v is obtained from diagram to the right. Note that the above formula does not apply generally to outer walls, where convection currents or cold draughts may occur.

Example
A supply air diffuser with a throw length of $l_{0.2} = 5.0$ m and $\Delta t = 4$ °C can be placed $L_v = 0.67 . 5 = 3.35$ m from the wall.



The relationship between the correction factor kv and the temperature difference Δt °C ($t_{supply\ air} - t_{extract\ air}$).

Minimum distance between supply air diffusers with high ceiling heights

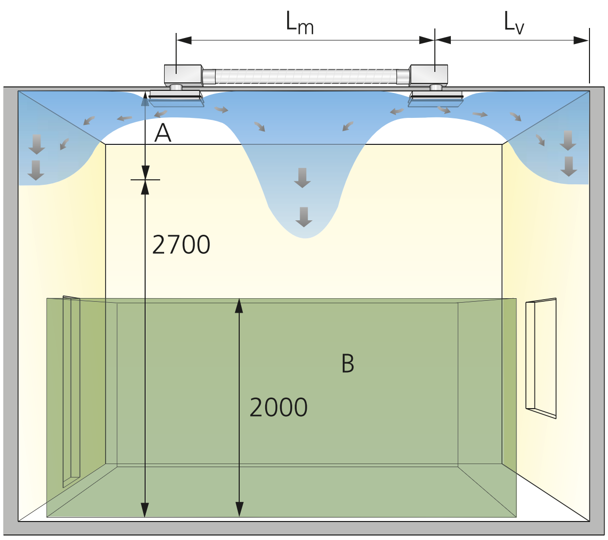
The specified throw length applies for a normal ceiling height of 2.7 m. For higher ceiling heights, the distance between the ceiling and occupied zone is counted as the deceleration distance for the air jet. The illustration in the top right corner shows the relationship for the distance between the two supply air diffusers and the distance to the occupied zone.

$L_{mA} = L_m - A$

Example
Two supply air diffusers, each with a throw length $l_{0.2} = 5.0$ and $\Delta t = -6$ °C mounted in the ceiling with the ceiling height 4.5 m gets a distance $L_m = (5.0 + 5.0) . 0.72 = 7.2$ m Calculating the distance $L_{mA} = 7.2 - (4.5 - 2.7) = 5.4$ m, i.e. the diffusers can be installed with spacing of 5.4 m

The illustration below to the right shows the relationship for the distance between the supply air diffuser and the wall, which can also be corrected due to the jet's longer deceleration distance.

$L_{VA} = L_V - A$

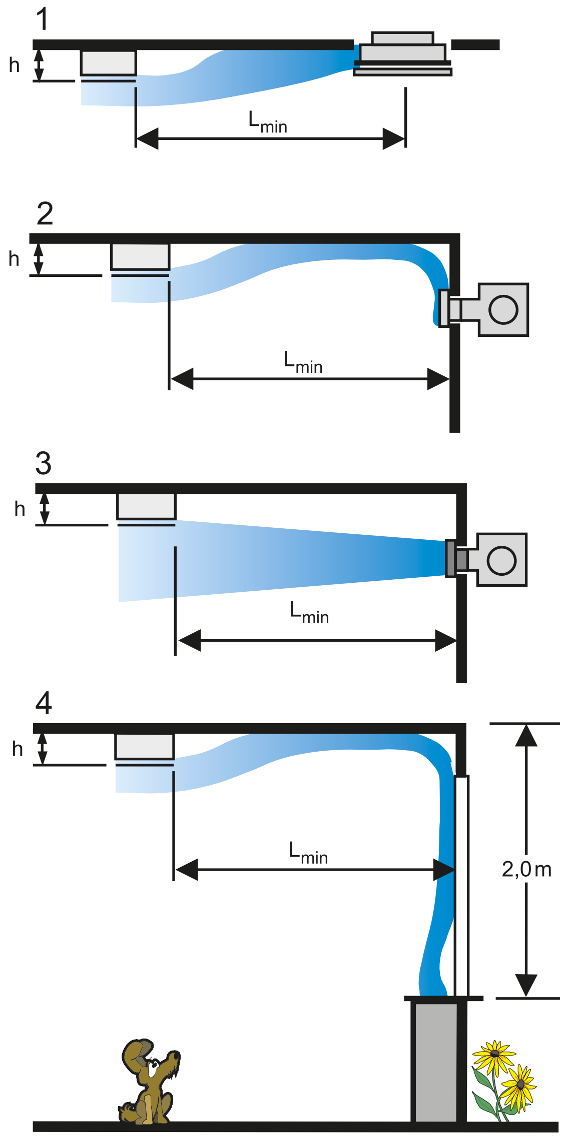


Minimum distance between supply air diffusers at the greater distance A + 2700
B = Occupied zone

Minimum distance between supply air diffusers and obstacles

Lighting fixtures and other items that may obstruct air distribution should not be placed too close to the supply air diffusers.

Different options are possible:



Different options for air supply with obstacles on the ceiling.
1 = Ceiling diffusers
2 = Rear edge diffusers
3 = Grilles in the wall
4 = Supply via window sill

For each option it applies that the minimum distance, L_{min}, is dependent on the distribution pattern from the diffuser and the supply air temperature. The shape of the obstacle is also very significant. Obstacles with rounded or angled edges are a minor disturbance compared to those with straight edges.

The following guideline values can be specified:

Option 1 and 2.
It applies to air at most 6 °C below room temperature:

$$L_{min} \geq 25 \cdot h$$

50% higher values are recommended for larger temperatures below room temperature.

Option 3
For grilles, these must be mounted at a distance from the ceiling that is ≥ 2h.

In addition, if the grille’s hydraulic diameter is greater than 1.4 x h there is no risk of plunging due to the obstruction. That is

$$\frac{2\,ab}{a + b} > 1.4 \cdot h$$

where a is the width of the grille grating width and b its height.

Option 4
For this option, the obstacle's height is limited for different L_{min} according to the diagram below.

Critical heights for obstacles on the ceiling with air supply via the window sill, as a function of distance from the window sill.

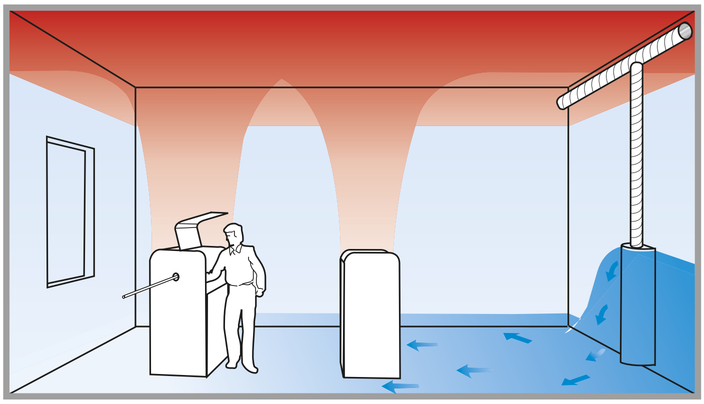
Project design for displacement ventilation

For ventilation according to the principle displacement flow, air is supplied at a very low air velocity. This means that the flow pattern in the room is controlled by the density differences, in the air. You can say the flow pattern is thermally controlled. This means that other factors must be taken into consideration than what is normal with mixing ventilation.

It is therefore important to carefully analyse the component conditions associated with the design planning. The following sub-elements can be distinguished: process analysis and calculation.

Process analysis
- Type of business/premises
- Degree of activity
- Convection currents
- Room dimensions
- Room layout

Calculations
- Airflow
- Energy balance calculation
- Convection air flow
- Resulting sound level
- Near zone



Displacement ventilation

Process analysis

Type of business/premises
The use of low velocity diffusers in rooms with contaminated production are appropriate as the diffusers have a very small co-ejection of room air.

Degree of activity
In rooms where consideration must be given to comfort, it is important to determine the comfort requirements based on the applicable activity level.

Convection currents
Convection currents arising in the premises determine together with the supply air the flow pattern in the premises. Both the supply air flow and the supply air diffusers therefore have a very large impact on the final result. If the design is to be based on the generation of contamination and the maximum contaminant concentration in the occupied zone, the airflow must be designed according to local standards and regulations regarding hygienic threshold values.

Room dimensions
Room height has a great impact on the air exchange and ventilation efficiency that will be obtained.

A high ceiling allows more space for storage of contaminated air while low ceilings limit these possibilities.

Room layout / General
As the position of heat sources and their size are extremely important for the end result, it is important to know their placement.

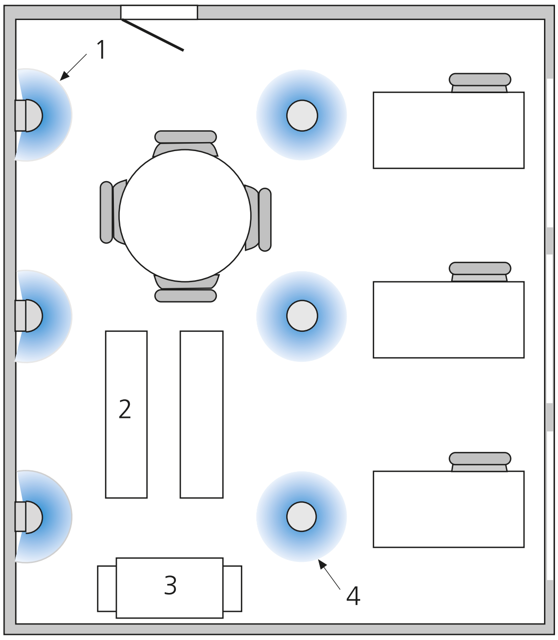
Accurate analysis of e.g. the placement of workshop machines is therefore a prerequisite for the design planning work.

In comfort systems, it is necessary to know how the rooms are to be furnished in order to determine the best possible placement of supply air diffusers.

Room layout / Open-plan office
Try to find places where people are not sitting permanently. Usually there are special "walkways" in whose vicinity the diffusers can be placed.

Flat or semi-circular diffusers can be placed by pillars, or perhaps round diffusers in the form of "fake" pillars.

Other areas that may be appropriate for diffuser placement in offices can be, for example the "central literature collection" or by the printer/copier.



Diffuser placement in an open-plan office
1 = Concentration of supply air diffusers far from the workplace
2 = Book shelves
3 = Copying
4 = Alternative supply air diffuser in the form of "fake" pillars

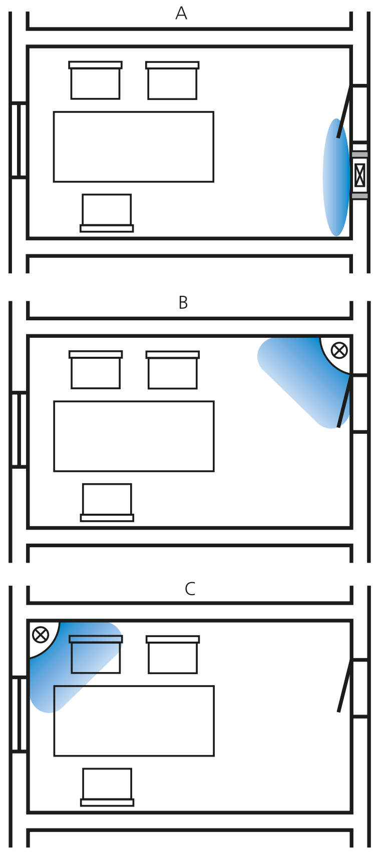
Room layout / Individual office
In normal office rooms the depth is often greater than the width. Try to place the diffuser in or on the wall to the corridor. This gives a great distance from the diffuser to the person sitting at the desk.

A
Often the diffuser can be installed in the stud compartment closest to the door as this surface is often unfurnished.

B
Diffuser placement behind the door panel is a less appropriate solution. The near zone is affected and increased air velocities occur along the wall, where in this case the visitors sit.

C
If the diffusers are placed by the facade, they should be displaced to one side depending on where the desk is positioned.

Occasional visitors that end up in the near zone are not as sensitive to draughts as the person sitting in the office all day.



Diffuser placement - individual office

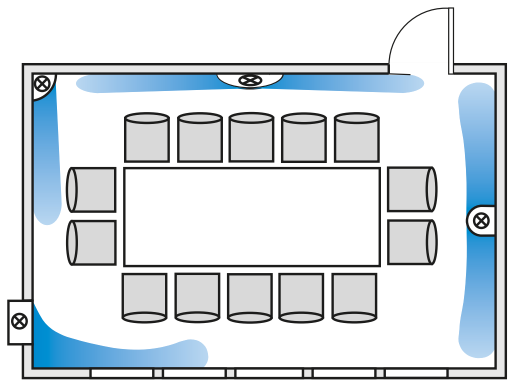
Flexible distribution patterns, near zones
In most cases it is a clear advantage if supply air diffuser's distribution pattern can be altered to prevent seating coming inside the air diffuser's near zone.

Each air diffuser will then have a number of circular rotatable air deflectors behind the perforated front panel. The distribution pattern is influenced by turning these. As the room's layout or function may change over time, flexible distribution patterns are a distinct advantage.

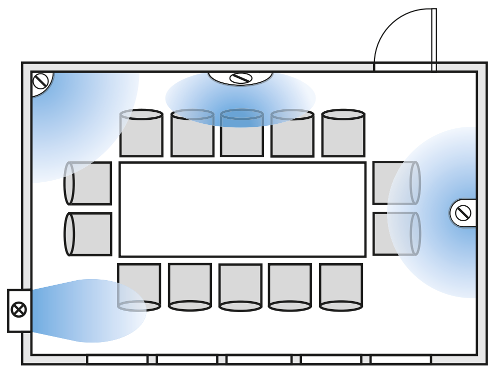
A rule of thumb is that the specified near zone's distribution above the floor area (m^2), can be transformed besides the default setting to:

- Right-hand directed near zone
- Right-hand directed near zone
- Long and narrow near zone

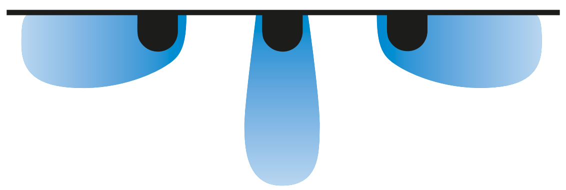
Common for these options is that the distribution area is the same as for the default settings.



The supply air diffuser where the shape of the near zone has been adapted according to the room's furnishings.



The supply air diffuser, where no redistribution of the flow pattern has been made.



Examples of optional settings.

Calculations

Airflow

For industrial installations, where the ventilation system must be designed for a certain maximum contaminant concentration in the occupied zone, it is important that the supply airflows are determined based on the amount of emitted contamination m (mg/s) in the premises and the permitted contamination concentration c_{till} (mg/m^3).

The airflow q is obtained as:

$$q = m/C_{till} \text{ m}^3\text{/s}$$

The concentration, c_{till} must always be less than the hygienic threshold value according to applicable regulations. If the contamination in question can also be found in the supply air, the supply airflow q is calculated according to the equation:

$$q = m/C_{till}-C_t) \text{ m}^3\text{/s}$$

where c_t = the contamination concentration in the supply air in mg/m^3.

In comfort systems the lowest air flow is generally 0.35 l/s per m^2 floor area. However, for individual offices, the outdoor airflow should not be less than 12 to 15 l/s per person.

Energy balance calculation

A calculation of internal and external heating loads that take into account the heat accumulation in the building must form the basis of a calculation of the requisite cooling load. This gives, together with comfort requirements, an appropriate system selection and supply airflow. Calculation of the displacement system's ability to provide cooling capacities is evident from this planning guide.

Convection airflow

The convection airflow for different heat sources in industrial installations does not need to be calculated if the airflows are determined as described above. The size of the convection airflows can also be disregarded in comfort systems when selecting the supply airflows.

Resulting sound level

The supply air diffusers typically have very low sound generation. Inherent sound attenuation is also often low, which is why you should always make an accurate sound calculation for duct system. Correction for the current room absorption must be carried out.

Near zone

It should be noted that during the design work it is necessary to take into account the size of the near zone. One cannot count on anyone needing to permanently occupy the near zone. This is especially important in premises with a large number of people. Therefore choose first and foremost diffusers according to the shape and size of the near zone, and secondly according to diffusers' sound generation.

The correlation between airflow, temperature gradients and thermal loads for displacement ventilation
A method to calculate the requisite airflow in order to limit the vertical temperature gradients for different heat loads are specified below. The method is taken from: Notice No. 16 of Building Services Engineering KTH, March 1991.

The temperature gradient must be limited and not exceed the limit values set out in "requirements on the indoor climate".

The requisite minimum ventilation airflow for a given maximum temperature gradient is obtained from diagram 1.

The temperature difference between level 1.1 m in the room and the supply air temperature is determined from diagram 2.

A check of the air temperature at floor level (t_{lg}) is made with the help of diagram 3. This check is important to avoid problems with comfort. t_{lg} must not fall below 20 °C.

A practical guide is also that the supply air temperature should not fall below 18 °C.

Designations

t_{lg}	= air temperature at floor level
t_t	= supply air temperature
t_f	= extract air temperature
s	= vertical temperature gradient (°C/M)
h	= height of room, m
$\Delta t_{t,1}$	= difference between the temperature at level 1.1 m and the supply air temperature

Methods for calculating the requisite airflow to limit vertical temperature gradients for different heating loads
An office room with a ceiling height of 2.7 m has a cooling load of 25 W/m². The vertical temperature gradient must be limited to 1.7 °C/m. Calculate the requisite airflow and temperature 1.1 m above the floor and on the floor.

Solution:
s x h = 1.7 x 2.7 = 4.6 °C

Diagram 1 gives q/A = 2.8 l/s,m²
Diagram 2 gives $\Delta t_{1.1}$ - 1.1 x s = 3.6
which gives $\Delta t_{1.1}$ = 3.6 + 1.1 x 1.7 = 5.5°C

Set the supply air temperature t_t = 18°C
which gives $t_{1.1}$ = 18 + 5.5 = 23.5°C

Diagram 3 gives
(q/A = 2.8 l/s,m² or 10 m³/h,m²

As:
$t_f - t_t$ = 25/2.8 · 1.2 = 7.5 °C

gives t_{tg} = 0.4 x 7.5 + 18 = 21°C

Result:
- requisite airflow = 2.8 l/s,m²
- air temperature at the level of 1.1 m = 23.5 °C
- air temperature on floor = 21.0 °C

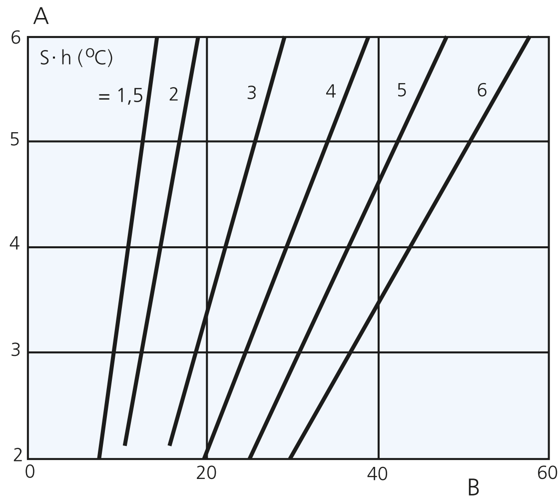


Diagram 1. The required ventilation flow as a function of the cooling capacity for different products of gradient and room height.
A = Airflow (l/s,m²) B = Cooling capacity (W/m²)

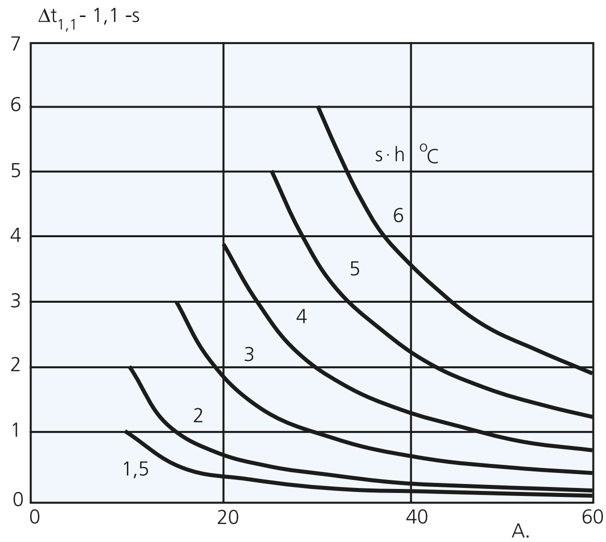


Diagram 2. The temperature difference between the air at floor level and the supply air as a function of the cooling capacity for different products of gradient and room height. A = Cooling capacity (W/m²)

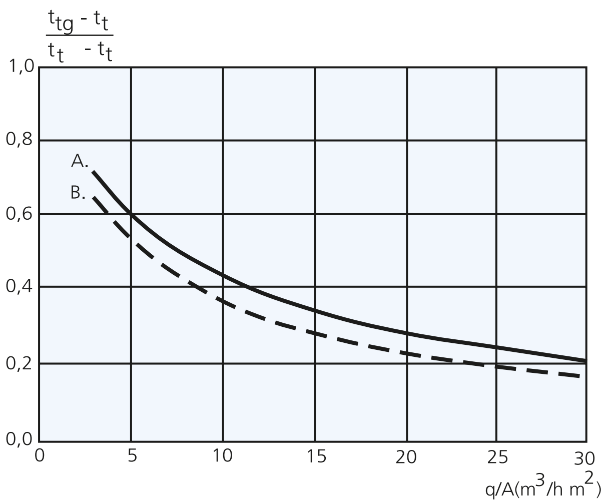


Diagram 3. Sizeless temperature difference in the floor zone at different airflows.
A = α_{kg} = 5 W/(m² · K) (thermal exchange constant due to convection on the floor surface). B = α_{kg} = 3 W/(m² · K)

Project design for waterborne climate systems

Calculation procedure

1) Start from with the current use of the building, the user's requirements and applicable energy requirements.

2) Determine the comfort requirements that apply to the building with regard to applicable building regulations. Guidance is given in the Swedish Indoor Climate Institute's "R1 Classified indoor climate systems - Guidelines and specifications".

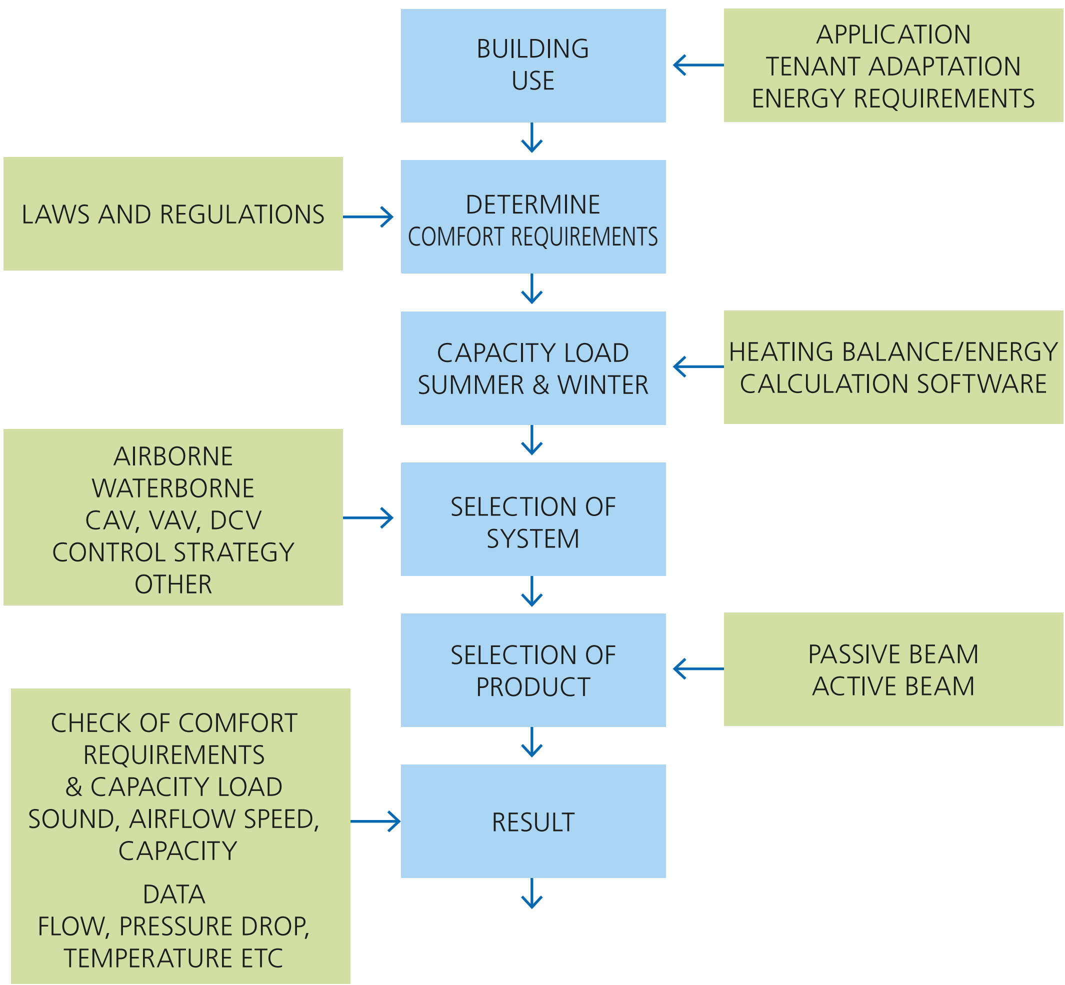
3) Determine the installation's capacity needs for winter and summer using the heating balance/energy calculation software.

4) Select a system solution based on the obtained calculation results (Airborne or waterborne; CAV, VAV or DCV; Control strategy, etc.)

5) Select the appropriate terminal products and the number of units (passive chilled beams, active chilled/climate beams, comfort modules or induction units).

6) Check the selected solution against the capacity, comfort and sound requirements, and throw length requirements. Correct the solution if necessary.

7) Proceed with the design of the unit and duct system based on calculated values for airflow, pressure drop and temperature.



Sizing key

System design

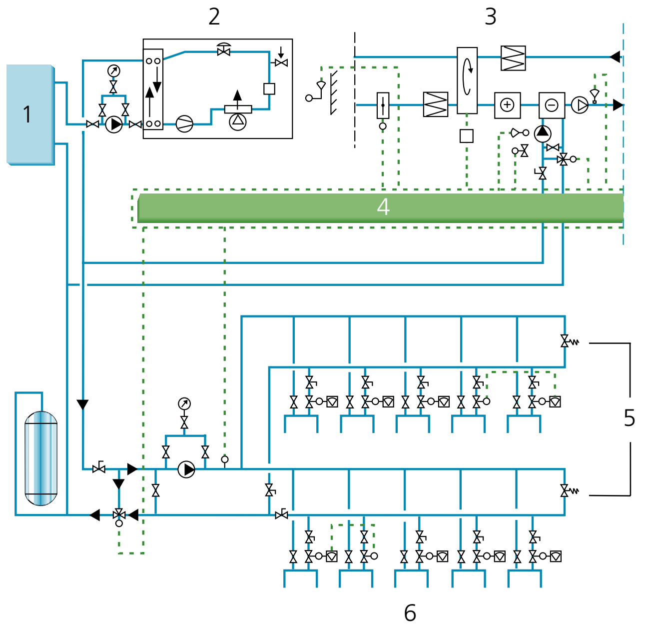
Cooling system

In a Nordic climate the cooling system should be designed with the evaporator placed indoors. Excess heat is either led off with a condenser placed outdoors or via the glycol system and dry coolers placed outdoors.

If in the Nordic climate you choose outdoor chillers instead, i.e. the evaporator is located outdoors, it is recommended that the intermediate heat is located indoors. This to avoid freezing point lowering agents (usually glycol) in the cooling water circuit. There are two factors why glycol should be avoided in the cooling water circuit. The pressure drop increases by 15-25% depending on the composition of the brine solution. Furthermore, the cooling capacity is reduced by approximately 15%, depending on the thermal exchange constant being lower on the water side.

Regulation of end devices

Chilled beams and perimeter units are almost exclusively fitted with 2-way valves. The advantages compared to the three-way couplings is a lower cost and simpler design and adjustment. In order to prevent high pressure at low loads, overflow valves are positioned in several places in the system. The fact that pressure controlled pumps can now be installed at a reasonable cost, also promotes the choice of a 2-way system.



System proposal
1 = Water tank
2 = Water chiller
3 = Air handling unit
4 = Control unit
5 = Differential pressure valve
6 = Chilled beams

Condensation protection

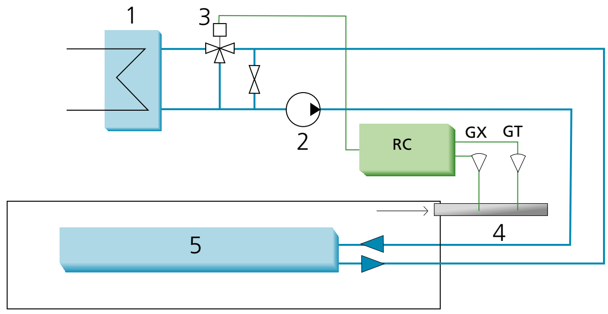
The air humidity can be high on warm late summer days. The higher the moisture content in the air, the higher the limit temperature (dew point temperature) for condensation on surfaces. You can read this relationship from a Mollier diagram (moisture-enthalpy diagram).

For example, at 25 °C and 50% relative humidity, the dew point is 14 °C (applies at normal atmospheric pressure 101 kPa), i.e. it starts to condense on a surface whose temperature is 14 °C or lower. During late summer days, the dew point temperature can sometimes rise to 15 °C, and in extreme cases, after a shower of rain, up 17 °C.

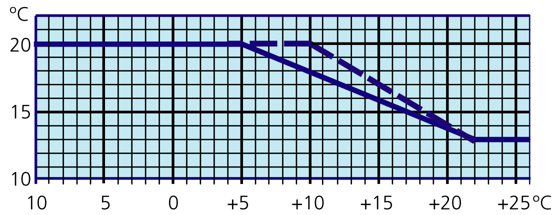
In order to avoid problems with condensation, you should always ensure that the system prevents condensation on end devices. The best way to do this is to always chill the supply air to ensure condensation is removed in the supply air unit's cooling coil.

Another possibility is to use a sensor that measures the relative humidity of the extract air according to the illustration to the right. The shunt group valve is controlled to keep the water temperature above the dew point temperature.

Outdoor compensation is applied to the supply air according to the diagram on the right to ensure drying of the air at high outdoor temperatures and high relative humidity. The engagement point +5 °C may vary slightly from system to system. See the dashed alternative curve. However, it is essential that when outdoor temperatures is around +22 °C and above, to have dehumidification after the air handling unit so that the supply air's dew point temperature is lower or equal to the supply temperature on the chilled beam's cooling agent.



Condensation protection control via the shunt group.
1 = Evaporator/condenser
2 = Circulation pump
3 = Shunt
4 = Extract air duct
5 = Chilled beam
RC = Regulator
GX = Humidity sensor
GT = Temperature sensor



Supply air compensation with regard to the outdoor air temperature.

Project design of the condensation protection

Proposals are given here for a suitable pipe system design with associated control and balancing valves for the interaction of between the air handling unit's drying of the supply air through condensation and the cooling agent temperature to the chilled ceiling side to prevent condensation on the chilled ceiling.

The system has been selected for the design outdoor temperature DUT = +25 °C and relative humidity RH = 50% which corresponds to a dew point of +14 °C. The cooling agent side's design temperature to the chilled beams is set to +13 °C supply and +17 °C on the return, see the diagram to the right.

On air treatment side, the cooling coil is designed for +8 °C flow temperature and +13 °C return temperature. These are temperatures that provide good conditions even for district cooling. Here an installation of 1000 m² has been assumed with a supply airflow of 1.5 l/s m².

For district cooling, you should note that suppliers often require a return temperature of at least +16 °C. Consideration should also be taken when designing to the loss of at least one degree in the heat exchanger for district cooling. This means that the return temperature from the end device must be at least +17 °C.

As shown in the Mollier diagram's curve for above the design data, an enthalpy difference Δi of 16 kJ/kg is obtained.

$P_{TL} = q_{TL} \cdot \rho_{TL} \cdot \Delta i$ [kW]

$P_{TL} = 1.5 \cdot 1.2 \cdot 16 = 28.8$ kW

P_{TL} = requisite power for cooling of the supply air with associated condensation precipitation at the design outdoor temperature DUT
ρ_{TL} = supply air's density in kg/m³
q_{TL} = supply air's flow in m³/s

The above gives a design cooling agent flow q_w at Δt_w = 5 K (+8°C to +13°C), and P_{TL} = 28.8 kW.

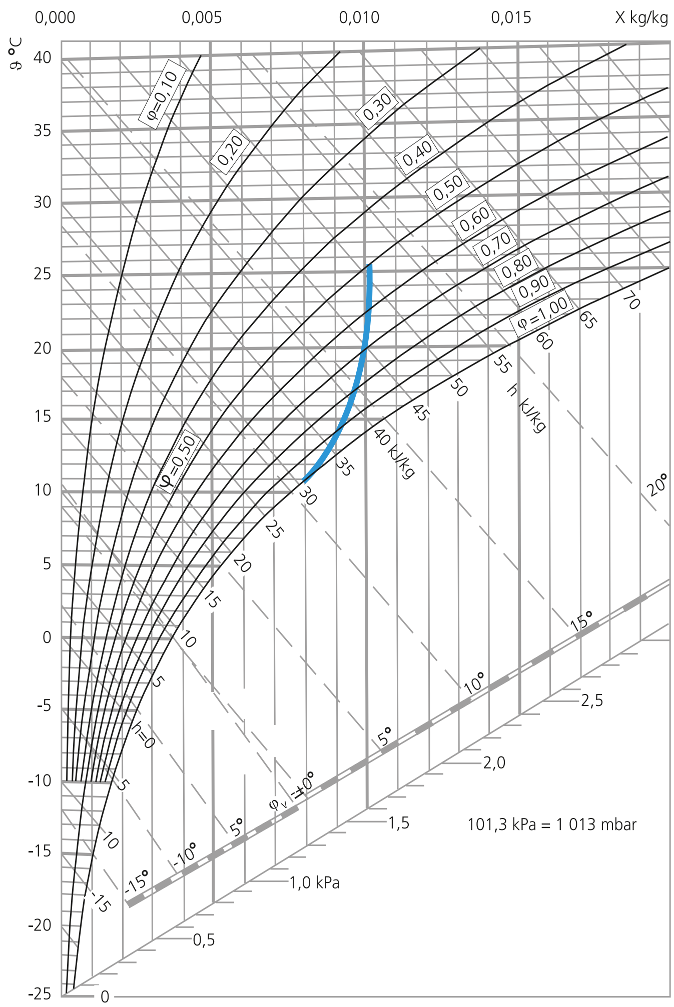
$qw = P_{TL} / (\Delta t_w \cdot c_p) = 1.72$ l/s

$qw = 28.8 / (5 \cdot 4.187) = 1.72$ l/s

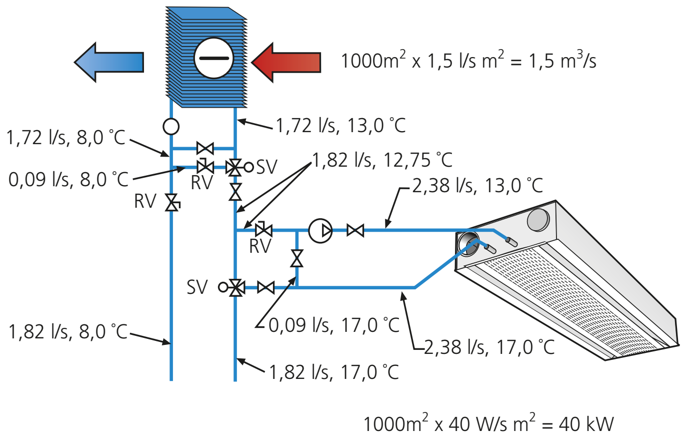
ρ_w = water's density in kg/m³
c_p = specific heat of water in kJ/kg °C
$4.187 = \rho w \cdot cp / 1000$

An equivalent procedure gives for the chilled beam side 1000 m² · 40 W/m² = 40 kW in requisite cooling load.
$q_{wk} = 40 / (4 \cdot 4.187) = 1.38$ l/s

Try the bypass flow of 0.09 l/s to the three-way valve SV1 past the air cooler. Now using the separate flows and their temperatures, calculated the mixing temperatures in different parts of the pipe system. From the calculated mixing temperatures it is evident that they interact well in design instance. The balancing valves must be measured with the control valve's port fully open against each balancing valve when the established flow for the RV valve is adjusted to the calculated value. The illustration to the right shows the resulting flows and temperatures.



Condensation protection. Change of state for the supply air.



Condensation protection. System principle with flows and temperatures.

Temperatures
The following temperatures should be seen as a recommendation. Discrepancies may naturally occur.

Recommended temperatures
Supply flow temperature cooling:
>13 °C (see also the previous section "Condensation protection"

Temperature increase cooling:
2-4K

Supply air temperature in cooling mode:
See the diagram Supply air compensation in the previous section "Condensation protection"

Room control
Control of the room temperature is normally done individually in each room via a room temperature setting. The room units control cooling and (where appropriate) heating valves in sequence so that heating and cooling are not supplied to the room simultaneously if the building is new.

In older buildings with poor insulation, it should not be assumed that the cooling and heating are supplied in sequence. Here it is recommended that the targeted operating temperature is controlled. It may happen that you need heating at the perimeter wall while there a cooling demand in the internal climate zones.

Water quality
If air is permitted to enter the pipe system, all the steel will begin to rust and disperse corrosion products in the system, with the result that heat exchanger performance will be affected and copper tubes risk to corroding. It is therefore important that the pipe system where the cooled water passes is completely closed.

Process water must be pure and clear and you should take samples of the water initially so you are sure that the system was leak-free from the start. Otherwise it takes only a few years before it starts to leak and then the damage is already done as system by then is contaminated by rust.

Recommended limit values
Recommended limits for the product are given in its documentation. On the water side, limits are given for the operating pressure and for pressure testing the completed installation. Stated pressures refer to the coils' working and test pressures. In most cases, it is the fittings and valves that limit the system's total pressure.

The minimum water flow is stated for the cooling and heating water circuits. These refer to the minimum flow required for safe evacuation of air in the circuit. The lowest supply flow temperature is determined by the dew point temperature and should always be dimensioned so that the system works without condensation.

The highest supply flow temperature denotes the highest temperature recommended for continuous operation. Note that large and rapid temperature fluctuations, especially in radiation ceilings, but also to some degree in climate beams with flange coils, can cause clicks due to the linear expansion in components. High supply flow temperatures (60-80 °C) may occur only when the heating load is great and must always follow the outdoor temperature so that the supply flow temperature drops as the heating load becomes less. This is also preferable from an energy standpoint as losses in the pipe system will then be lower.

High temperature systems where control of the room temperature is based on the products being continuously heated up and cooled down should be avoided.

Nozzle configuration

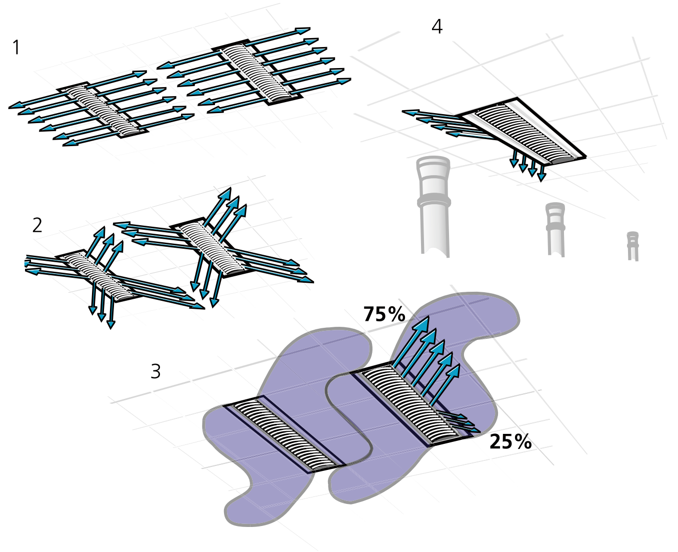
Capacity data for ceiling units is usually specified for different nozzle configurations. The configurations give each product unique K-factors so that you can optimise the airflows, nozzle pressure and flow direction for each type of room and product placement. In most cases symmetrical two-way supply (50-50%) is used for climate beams, but can also be configured for asymmetrical two-way supply with greater airflow to the left or right (75-25%). Selected configurations can be changed later by plugging nozzles.

Comfort modules are also equipped with nozzles, but with a simplified operation. Through a simple adjustment, you can change the units' nozzle setting for more or less air volume individually on all four sides. This means that you have the potential to adapt the air volume and air direction to the prevailing requirements throughout the product's entire life.

Air deflectors

The different ceiling units' distribution patterns are normally specified in the supplier's calculation program. The intention is to provide a basis to determine the distance to the wall or other obstacle, and the distance between the counter blowing units. Among the serious manufacturers, the given distances between the units and to walls are reference parameters based on both laboratory measurements and experience.

Using air deflectors the distribution pattern as well as the penetration depth can be individually modified on each ceiling unit, which helps to make them very flexible. The number of setting combinations is very large and the most common distribution patterns can be found in the documentation.



Examples of different air deflector settings

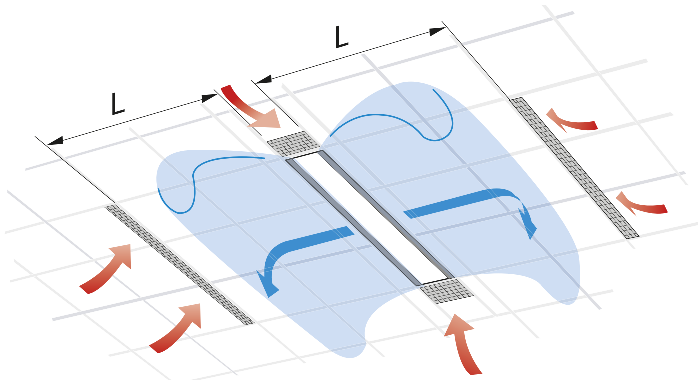
Distance between climate beams

Detailed recommendations about correct distances between the units and to walls are obtained with the help of the supplier's calculation program.

Chilled beams recessed in suspended ceilings

In order for the chilled beam to receive the right circulating air flow, it is important that circulation air openings are arranged in the suspended ceiling. Circulation air openings with a net area of at least 0.1 m^2 in the suspended ceiling are required for every metre of climate beam.

If possible, the circulation air openings should be located by the unit's short sides. Where this is not possible, locate the circulation air openings at a distance (L) of at least half the recommended distance between the two units, see figure 2.



Recommended placement of circulation air openings in the suspended ceiling to the chilled beam

Project planning climate beams

Individual offices

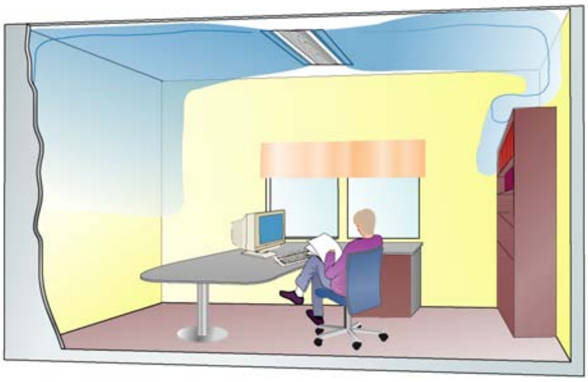
In an individual office, you always have a long mixing zone before the low-temperature air that leaves the ceiling unit reaches the people in the occupied zone.

From the position of the climate beam, the air can follow the ceiling, wall and part of the floor and throughout this distance mix with room air and raise the temperature of the circulating air, so that air conditioning of the premises is performed without the risk of draughts. Thanks to the long mixing zone, the air volume and impulse can also vary over a wide area without the risk of draughts in the occupied zone becoming too high, so that the nozzle configuration does not necessarily need to be altered with a change in airflow.

Corridor – facade

This is an installation option for modular offices. The product can be freely placed either centrally in the middle of the room or along one of the partition walls. Central placement is the conventional option to suit any beams, both with and without supply air and/or heating. Placed along the partition wall can be done with products with and without supply air. In the supply air instance it may be appropriate with an asymmetric air supply.

Advantages: Easy connection of water and air to the corridor.

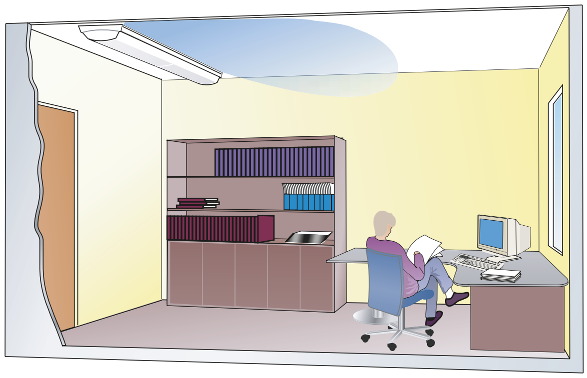


Corridor/perimeter wall oriented climate beams in individual offices

Rear edge

The rear edge solution suits modular offices and large offices. Depending on the volumes of supply air either symmetric or asymmetric air supply is chosen. Given that most heat loads are normally found next to the perimeter wall, a downward air supply interacts with the heat load resulting in too high air velocity along the floor. Therefore at least 50% of the supply air should be directed to the perimeter wall and controlled using air deflectors. Products with radiant heat should generally be avoided at the rear edge. In order to obtain small radiation asymmetry, it is recommended instead to install radiators or radiant heat on the outer wall.

Advantages: Easy connection of water and air to the corridor.



Rear edge placed climate beams in individual offices

Front edge

The front edge solution suits modular offices and especially large offices. The placement is well suited for products with cooling, supply air and heating functions. For placement near the facade with a high nozzle pressure and airflows an asymmetrical air supply is appropriate. With placement closer to the centre of the room uniform air supply works well.

Advantages: All the climate functions in one unit makes design and installation simple. Usually gives the opportunity of longer distances between the products in an large office than when mounted in the direction corridor – facade, which may increase the possibility of higher cooling capacities. Greater flexibility is created by leaving a space between the short sides of the units to construct or move partition walls without disturbing the comfort of the room.

Front edge placed climate beams in individual offices

Large rooms

Larger rooms can include shops or open-plan offices. In a large room the conditions for climate beams are different compared to individual offices. Usually there are no wall surfaces that the low-temperature air can flow along, other than the facade, which is why air deflectors are always recommended and especially then for corridor/perimeter wall oriented installations with counter blowing units. Front edge placement according to the previous page can also be recommended.

Many climate beams and comfort modules are able to produce different air distributions such as symmetrical, asymmetrical and displaced, which permits adaptation to the shape of the room (proximity to walls, other installations, etc.).

With mixing systems an exchange occurs in the vertical direction between the room air and the cooled supply air and through the needs of flexibility regarding free furnishing, partition wall placement, etc. which generally exist for modern offices, the system must be planned from the outset in a way so the risk of draughts is eliminated. The calculation program for climate beams includes all the necessary data needed to carry out design planning.

Advantages: Products with supply air give good air supply and mixing without the noise and draughts.

To consider: Passive chilled beams should be in the direction of the convection flow, otherwise there is a risk of reduced capacity. Similarly, air supply diffusers should be positioned so that they blow along the chilled beams. Air deflectors should always be fitted on the active units.

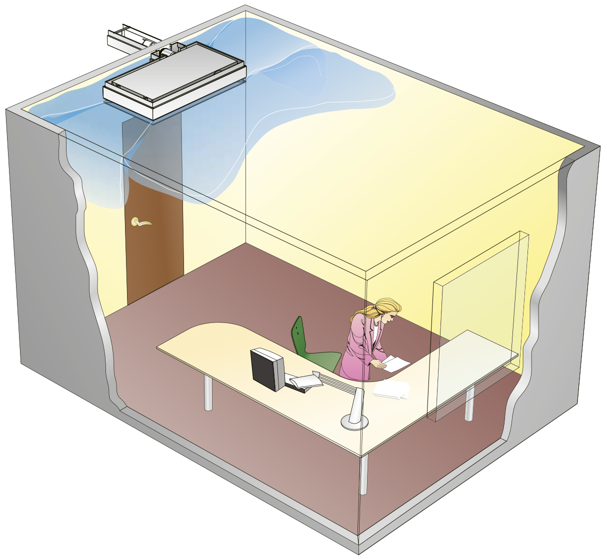
The performance of climate beams in large rooms.

Project planning comfort modules

Individual offices

A comfort module can be placed anywhere in an office room without causing a draught. By setting the airflow direction depending on the location of the unit you can optimise the comfort in the room from case to case. A good example of this is a rear edge placed comfort modules in a suspended design as shown to the right. With 4-way air supply the corridor wall, ceiling and partition walls are utilised to mix the air with the room air. Unlike a traditional two-way supply this gives lower air velocities, particularly at floor level.

Advantages: The same product provides optimum comfort irrespective of placement in the room.



Rear placed comfort modules in individual offices.

Large rooms

In large rooms, the comfort module's 4-way air supply comes into its own. Unlike climate beams' tendency to give slightly higher air velocities locally in the large rooms, comfort modules give even distribution throughout the whole room. This means that the risk of draughts is less and comfort better. As the life of the product is normally longer than the intervals between rebuilding or renovation of the premises, high demands are made on flexible systems. With the help of the built-in nozzle control and air deflectors there are vast possibilities to retain the existing installation, even when conditions change.

Advantages: Very even distribution of air velocity in the room. Great flexibility with respect to air flow changes and any changing conditions.

Heating function

Today in a new build, facades are constructed with very low U-values, while the quality of the windows has been significantly improved. That means a higher surface temperature on the inside of windows with less cold radiation as a result and a minimum of cold draughts. A better facade also reduces the heating load in winter, which means heating rarely needs to be supplied to the room when the room is in use. Internal heat loads are usually sufficient to heat the room during the day. With less cold radiation, the need of traditional radiators to increase the operating temperature is reduced.

As the comfort modules mix the supply air faster than a chilled beam the temperature difference decrease faster. This, combined with the improved construction technology makes it possible to heat rooms with comfort modules in a better way than with traditional climate beams. It is actually possible in many renovation projects to heat rooms with comfort modules. Using calculation software you can, among others, calculate the operative temperature in very efficiently. The illustration to the right shows temperatures from full-scale tests in a simulated winter conditions.

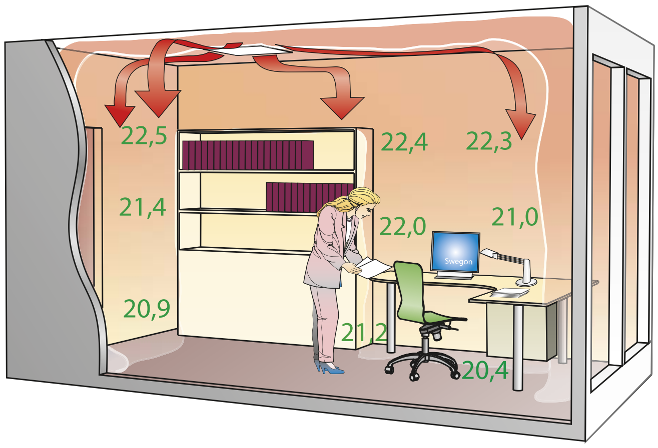


Illustration from the full scale tests, heating with comfort module.

Project design for induction units

Facade placement

High quality systems always includes the functions cooling, heating and ventilation. The system provides excellent comfort for both cooling and heating.

Advantages: Complete climate system including control equipment. Suitable for low ceilings. Low maintenance and low service costs. Gives flexible room solutions where partition walls can be moved without disturbing the comfort of the room.

Ceiling placement

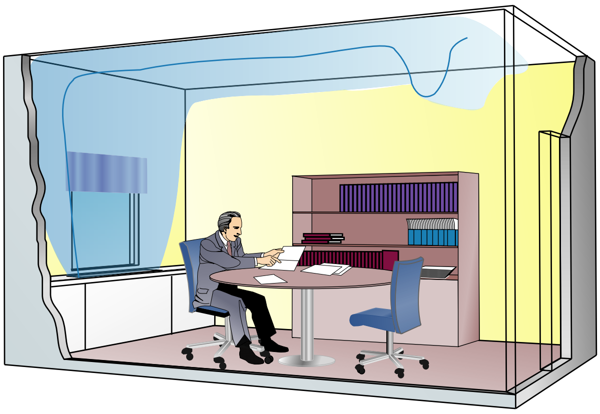
Another option is placement in the ceiling’s front and rear edges. A typical installation example is in a hotel room. Here the hall section is normally equipped with a suspended ceiling which transforms into a room without a suspended ceiling. In the space above the hall's suspended ceiling, a horizontal induction unit with the same basic design as a perimeter unit is installed. This provides the room with both ventilation cooling and heating.

Advantages: Complete climate system including control equipment.

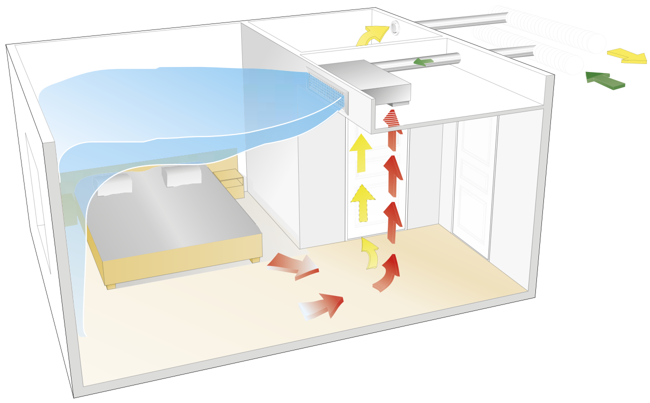
Recessed in the floor

Induction units can also be placed horizontally and recessed in raised floors when fitted.

Advantages: Complete climate system including control equipment. Enables solutions with windows that go down to the floor (so-called glazed facades) without the installation affecting the overall impression.



Principle for wall-mounted perimeter units for cooling function



Principle for ceiling-mounted perimeter units for cooling function

Full scale tests in the laboratory environment

In well equipped ventilation engineering laboratories full scale tests can be performed where the indoor climate is simulated. The purpose of these tests is to find out the practical limits for the use of the products and what recommendations are needed for advice and guidance through the design phase.

To be able to carry out a simulation in the laboratory requires a layout and furnishings that are so similar to the premises in question as possible.

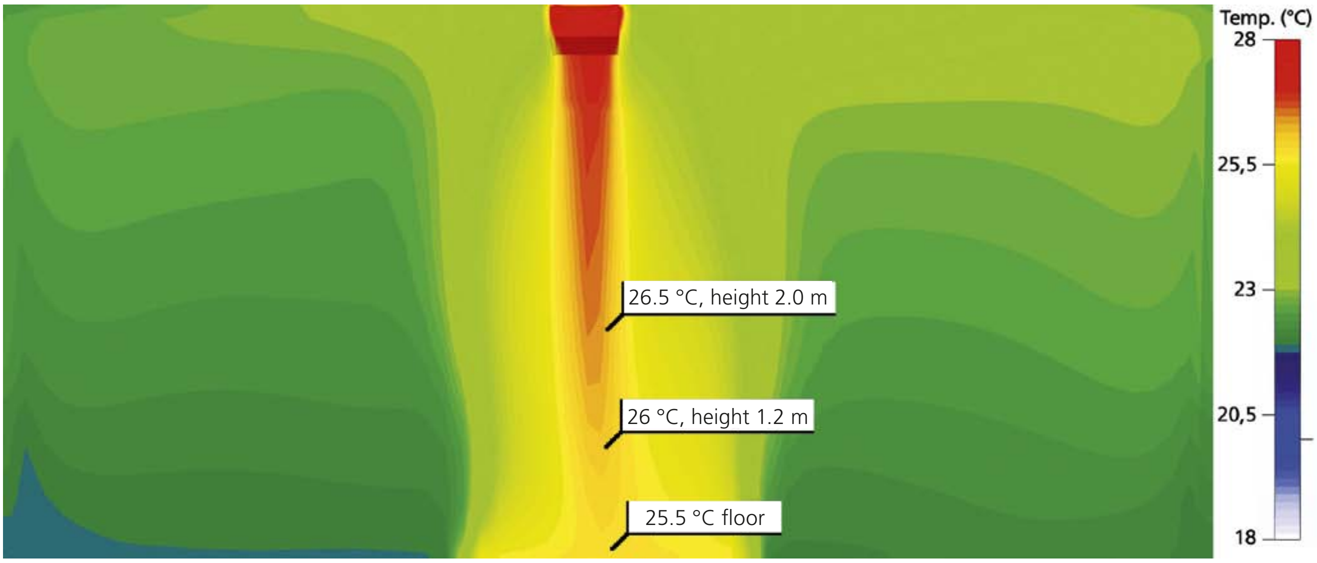
A climate calculation should always form the basis of the climate system design to achieve the right level of heating and cooling.

CFD

An alternative to full-scale tests can be advanced computer simulations, CFD (Computational Fluid Dynamics). This may be of particularly interest in cases where it is not practical to physically replicate the premises in question. Examples of this include sports arenas, concert halls and other large spaces. Simulation with CFD provides a good indication of the course of events in a room and can easily be visualized with different colour scales.

The illustration below shows a simulation of vertical distribution of hot air, with cold walls and ceiling.

Full scale tests in the laboratory



Simulation of temperature with CFD

Guidelines waterborne climate systems

Test methods

Test standards are needed to be able to compare products from different suppliers. It is only then that product data will be of equal value, comparable and unambiguous.

The standards that have been applicable in Europe have been the Nordtest method NT VVS 078 "Ceiling Cooling Systems" and the so-called V Method in Swedish Association of Air Handling Industries V Publication 1996:1 (mainly in Sweden). At the same time in Germany, a DIN Standard has been drawn up for flat chilled ceilings, which is also used for passive chilled beams (chilled beams without supply air).

Many climate beams and comfort modules are now Eurovent Certified, which guarantees correctly specified data. This will facilitate design and quality assurance.

Nordtest and V Publication

The V Method is a slightly revised version of the Nordtest method and was produced by the Swedish Testing and Research Institute and the then Swedish Association of Air Handling Industries (now: Svensk Ventilation, also called SwedVent) together with some suppliers from Sweden and Finland. The main difference between the two methods is at which water flow the products should be tested. In the Nordtest method, the water flow is fixed and related to the pipe diameter in the heat exchanger, i.e. all products are tested with the same water flow irrespective of the cooling capacity.

The purpose of the V Method was to allow products to be tested using an operating scenario that reflected true conditions and it was then chosen to steer towards the water flow that provides a temperature difference of two degrees through the heat exchanger. This means that if two products have a different cooling capacity, then they should be tested at different water flows.

The test room consists of two rooms, where heat is indirectly supplied to the inner test chamber via the floor and walls. The heating balances the supplied cooling capacity during the measurement sequence on the test object so that the test chamber's temperature is constant.

DIN 4715

In the German DIN 4715 the test chamber consists of an insulated room. Inside the test chamber, the test object's cooling capacity is balanced by means of "dummies" (occupant simulators) that emit heat directly in the room.

EN Standards

A work was started in 1996 to produce common European Test Standards. Since the start, several suppliers have been an active party in this joint European effort, which has resulted in three new standards:

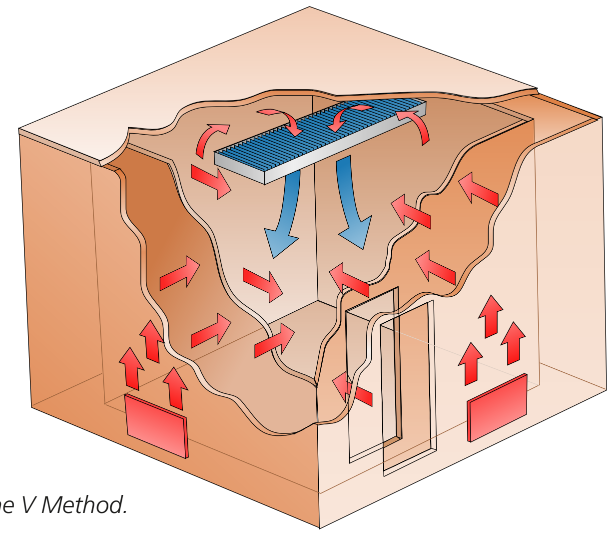
- EN 14240 for flat chilled ceilings
- EN 14518 for passive chilled beams
- EN 15116 for active chilled beams

Today most reputable suppliers have tested and certified climate beams and comfort modules through the Eurovent Certification Company. Through this third-party certification, clients and consultants are guaranteed that the technical data presented corresponds with reality.

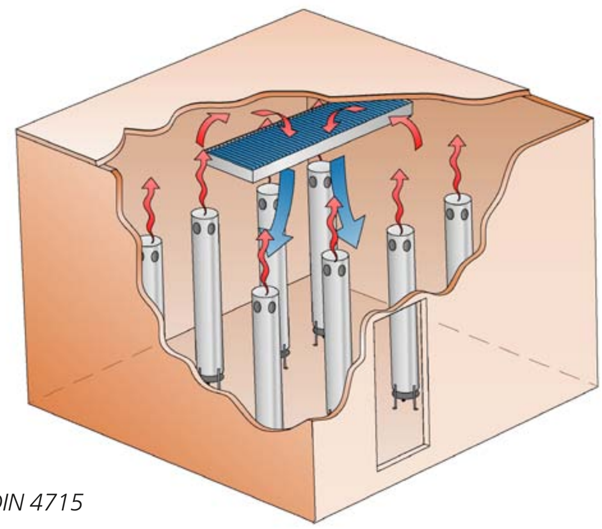
To consider

Regardless of which standard is referred to, there is an underlying test method for producing product performance data. The new EN standards are very similar to the old and the difference is first and foremost how the reference temperature in the test room is measured. Previously, it was calculated at a level 1.1 metres above the floor whereas the new standards for chilled beams always refer to the "on coil" temperature, i.e. a temperature near the chilled beam. Previously it has been discussed whether chilled beams give a fully mixing system or whether it is warmer up by the ceiling. When we use "on coil" temperature as a reference, this discussion becomes uninteresting.

Note that the current standards' test methods do not necessarily correspond with the operating instances that may come into question when designing an indoor climate system. In the standards, the cooling water system is designed for a 2 degree temperature difference on the supply and return, while most systems are designed for a 3-4 degree difference. The standardised test methods intended to simplify the comparison between different beams. To determine the correct cooling performance, the cooling capacity must therefore be recalculated for the applicable cooling water flow.



The V Method.



DIN 4715

Project design for residential ventilation

General
Basic principles of indoor climate in premises also applies to residential buildings. This section therefore addresses only the specific requirements and considerations relating to residential buildings.

Airflows
Previously in Sweden there were general requirements for different rooms in the dwelling. Now overall requirements are stated instead to avoid standardized sizing and special consideration is given to sensory pollution loads from persons, moisture and the occurrence of food odours.

According to the rules established by the Swedish Board of Housing, Building and Planning, Boverket the supply of outdoor air to homes must continuously be at least 0.35 l/s per m^2 floor area when occupants are in the home. If the ventilation system is demand-controlled via a timer or air-quality sensors, for instance, the flow of outdoor air can be decreased to at least 0.10 l/s per m^2 floor area when no occupant is present in the home.

In addition, it is recommended that the outdoor air flow should not be less than 4 l/s per person. It should be noted that significantly greater airflow is required at Swedish workplaces, the recommendation for housing is therefore low.

Boosting
The system must include provision for forced ventilation or a means of airing in rooms intended for activities of daily living, the preparation of food, sleep and relaxation as well as rooms for personal hygiene. A 30% increase of the normal airflow is usually estimated for boosting.

Circulation air
The use of air recirculation shall only be permitted within one and the same home and the extract air from the kitchen, hygienic rooms and the like must not be recirculated. If air is recirculated in the home, the supply of outdoor air with an airflow of at least 0.35 l/s per m^2 floor area.

Moisture load
Moisture load requires special attention as our habits over time have changed. For example, many people take daily showers and washing and drying is often done in the dwelling. Tumbler dryers that extract air or drying cabinets should not be connected to the ventilation system. Instead they should be equipped with their own local air extraction duct. Condenser type tumbler dryers without duct connection are recommended.

Lower airflow
A lower airflow when no one is spending time in the home means operational economy. This saves on fan energy and the home heating system does not have to heat as much outdoor air during the cold season. However, the lower airflow must be designed to ventilating out emissions from furnishings and building materials and above all moisture.

Negative pressure
The extract airflow should be approx. 5-10% more than the outdoor airflow. In this way there will be less negative pressure in the home, which reduces the risk that moisture from inside the home will be pressed out into the walls and ceiling.

Cooker hood
Cooker hoods are frequently equipped with a built-in fan or can be connected to a ceiling fan. Some ventilation units for residential properties are able to connect the cooker hood to the ventilation unit's extract fan. However, there may be local and national restrictions in place that prevent this.

Installation and investment costs will be lower if the ventilation unit's extract air fan is also used for the cooker hood. However, it does not affect heat recovery as the extract air from the cooker hood is rarely led through the heat exchanger.

Often negative pressure in the dwelling is accepted during the limited time the cooker hood is in use. Nevertheless, there is potential for negative pressure compensation, if the ventilation unit is equipped to speed up the supply air fan when the cooker hood is in use. This can be done via a differential pressure sensor in the cooker hood's extract air duct or via a microswitch on the cooker hood damper.

Example of design airflows
Example of the minimum airflows for a home with 120 m^2 and without consideration given to the cooker hood:
– Normal outdoor airflow: 0.35 l/s x 120 = **42 l/s**
– Outdoor airflow, no occupants in the home: 0.10 l/s x 120 = **12 l/s**
– Extract air flow: 42 l/s + 10% = **46 l/s**
– Boost: 46 l/s + 30% = **60 l/s**

Accordingly the ventilation unit must be able to manage an airflow of 60 l/s with the applicable pressure drop for the duct system and air diffusers. In addition, the energy and sound requirements must be met (see next section).

Energy use

Homes in Sweden must be designed in such a way that the specific use of energy in the building (all energy over and above the electricity used for operating household appliances) shall amount to the following values:

	kWh/(m^2 x year)
Climate zone I (northern Sweden)	130
Climate zone II (central Sweden)	110
Climate zone III (southern Sweden)	90

Homes with electrical heating have stricter requirements, but for homes with other heating energy requirements are not particularly difficult to satisfy. A likely development is that energy requirements will be strengthened in the future.

The specific fan power of the ventilation system (SPF_v) should not exceed SFP_v 2.0 kW/(m^3/s). Higher values are, however, acceptable for ventilation systems with variable airflows, such as reduction of the airflow when the home is unoccupied.

Ventilation systems for homes shall at least be equipped with control equipment that adjusts the power consumption according to the outdoor temperature, usually via outdoor temperature sensors. If even operating times and airflows can be demand-controlled, energy use is reduced even further.

Acoustics

The highest permissible sound pressure level for installations in homes that continuously generate sound is 30 dB in rooms for sleep, relaxation and activities of daily living and 35 dB in other rooms.

35 dB and 40 dB are permissible from installations that briefly generate sound. Installations that can be directly controlled by the user, e.g. forcing of the airflow in cooker hoods, are not included in the scope of the requirements.

The flow-generated sound from the ducting as well as the sound generated by the air handling unit itself and emitted to the surroundings should be taken into consideration in the sizing of ventilation systems. The risk of counteract crosstalk via the duct system in multi-family buildings should also be considered.

Sound emitted to duct systems

The duct system should be arranged with as few bends and branches as possible since these components generate sound. In most cases sound attenuators need to be installed in the supply air and extract air duct as well as in the air recirculation duct, whenever applicable. Install the sound attenuators as close to the air handling unit as possible.

Sound emitted to the surroundings

If the sound from the unit to the environment is too high, sound attenuation measures can be implemented using e.g. acoustic panels or the unit can be enclosed.

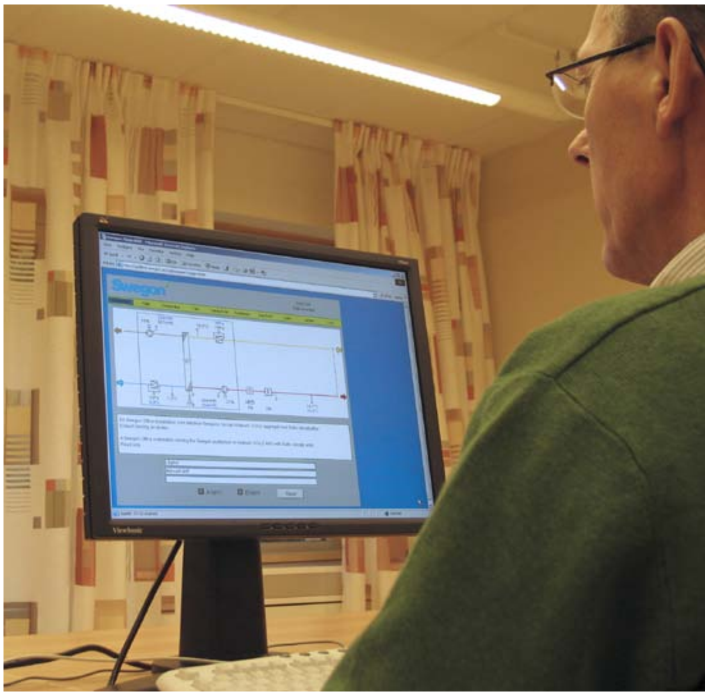
In order to avoid the transmission of sound in the building structure the air handling unit should be installed against a wall and should be furnished with intermediate insulation board, sound attenuating pads or the like. The wall itself should be insulated. However the unit should not be installed against a wall that borders a room intended for daily use, sleep or rest.

Software utilities for project design

The use of software designed for the purpose can facilitate different types of calculations during project planning. Some software works locally on your own computer and over the Internet. Some are also linked to other software commonly used in the design calculations, for example, CAD software.

Development has accelerated in this field and therefore only some general examples are mentioned:

- Calculation software for sizing air handling units.
- Calculation software for sizing cooling units and heat pumps.
- Calculation software for sizing residential ventilation.
- Calculation software for sound generation in duct systems as well as for sizing and selection of sound attenuators.
- Calculation software for sizing air diffusers.
- Calculation program for heat balance in the room
- Sizing tools for the different types of climate system



Measuring and commissioning

Measuring and commissioning

General

Poor adjustment is a common cause why many ventilation systems do not work as intended. As a rule, this is because the technical prerequisites for commissioning the installation have not been taken care of and there is a lack of knowledge in commissioning and metrology.

The commissioning must be carefully prepared from as early as the design stage. You should also take into account any changes in the installation during the construction period. Commissioning work is therefore time consuming if not prepared properly.

It is important that sufficient resources are allocated for this work. This investment can be an insurance to ensure the installation works as designed.

The recommendations on how airflow measurements are to be performed are documented in the report: "Methods for measuring airflow in ventilation systems", which can be obtained from the Swedish Building Centre.

Measurement

In a ventilation system, the deviation from the prescribed value for an airflow must not be greater than 15%, including measurement error.

The instruments must be calibrated using a method that gives a known low error rate. Calibration curves where the correction is given as a function of the reading should be used.

Accurate measurements can never be made. When measuring e.g. an airflow you must always take the following three errors into account.

- Instrument error – designated m_1 and can for instance be due to friction within the instrument or be residual from systematic errors in a calibration.
- Method error – designated m_2 and may be due to incorrect measurement points. Method error m_2 is obtained for the recommended measurement methods from the above report, "Methods for measuring airflow in ventilation systems".
- Reading error – designated m_3 and may be due to poor scale resolution.

A probable measurement error can be calculated using the following formula:

$$m = \sqrt{m_1^2 + m_2^2 + m_3^2}\ \%$$

Measuring in ducts

There are three fundamentally different methods for measurement in ventilation ducts:

A1 – measuring with Prandtl tubes

A2 – measuring with fixed measurement tappings

A3 – measuring with trace gas

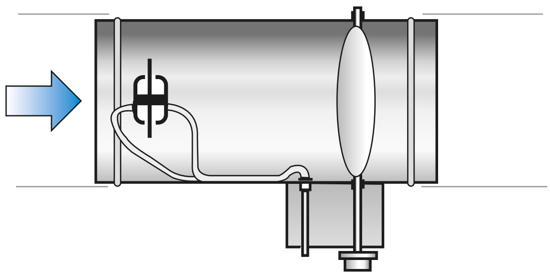
Method	Designation	Method error m_2
Traversing with a Prandtl tube in the duct:	A1	
Circular cross section	A1	4-5% recommended measurement plane, 7% alternative measurement plane
Rectangular cross section	A1	4% recommended measurement plane, 7% alternative measurement plane
Fixed measurement tapping	A2	See installation dimensions for error limits 5 and 10%
Determine the airflow using trace gas	A3	5 and 10%

Methods for measuring airflows in duct systems.

In order to reduce the time consumed and increase the accuracy of commissioning it is recommended that fixed measuring devices (method A2) are used as far as possible. There are different types of measuring units to choose from:

Most of the large manufacturers' products have fixed measuring tappings. In addition, there are adjustable measuring units, where the measurement function has been supplemented with a damper for easy adjustment of the airflow.

The airflow is determined by measuring the resulting pressure difference across the centre body. The flow is then obtained by using special measuring instructions.



Adjustable measuring unit

Measuring on supply air diffusers

There are three recommended methods for measuring airflows on supply air diffusers.

Method	Designation	Method error
Pressure drop measurement with fixed measurement tappings	C2	5%
Bag method	C5	3%
Measurements with conventional sleeve equipped anemometers, supplemented with extension connection fitting	C3	5%

Methods for measuring airflows on supply air diffusers.

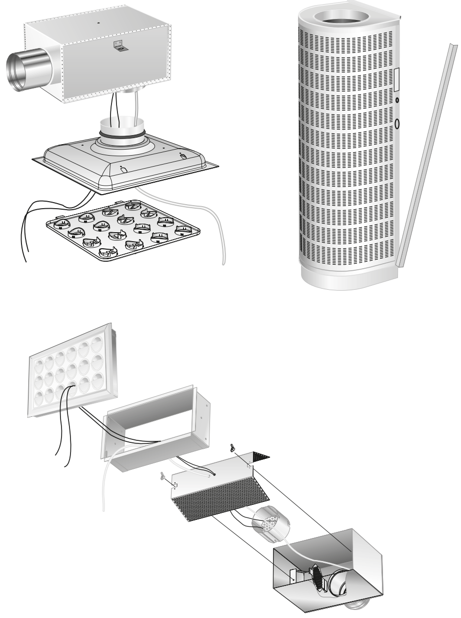
The diffusers from major manufacturers are often equipped with fixed measurement tappings according to method C2. The measurement tappings are either directly in the diffuser or in the pressure reduction box, which is available as an accessory for most diffusers. A pressure gauge is connected to this and a characteristic pressure measurement is taken. The flow is obtained as a function of the characteristic pressure difference.

In the pressure reduction box is an adjustment damper, which should be accessible through the diffuser. The design of the box shall ensure good sound attenuation and a smooth inflow of air to the supply air diffuser, which will guarantee an even distribution pattern.

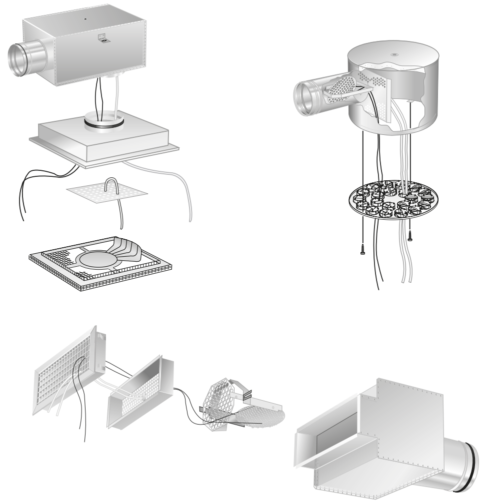
The damper in the adjustment box should be easy to remove to facilitate cleaning.

Depending on the type of pressure reduction box and the air diffusers, the number of measurement tappings varies between 1 or 2 tappings.

Example of air diffusers with one or two measurement tappings



One measurement tapping



Two measurement tappings

Commissioning factor air diffuser

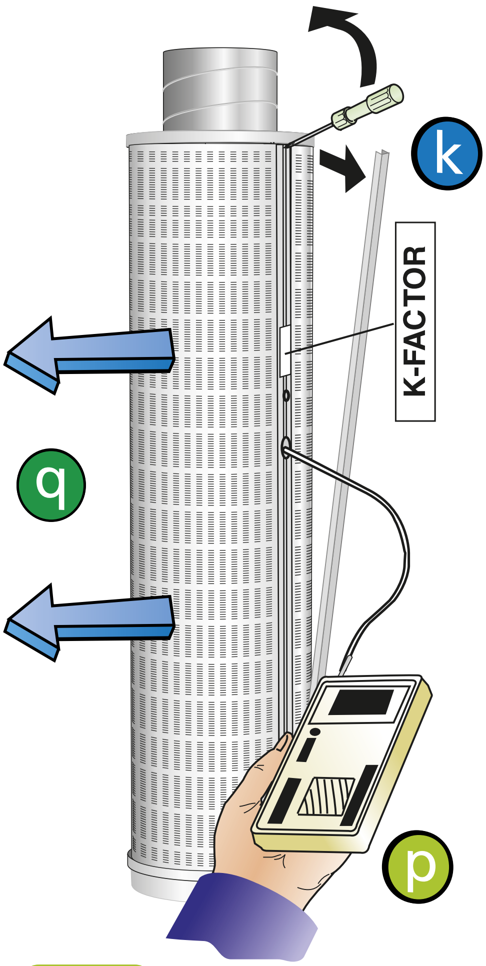
For each air diffuser or plenum box there are commissioning factors specifically drawn up for each product where the flow can be calculated.

This way of determining the airflow applies to both A2 methods and C2 methods.

Continuous control of the commissioning data is carried out at the manufacturer's laboratory. There is often separate documentation with instructions for diffusers installation, commissioning and maintenance.

In order to simplify commissioning, the measurement tappings should be equipped with plastic tubing and the adjustment dampers should be equipped with cord control (different coloured cords for opening, closing the damper), so that commissioning can be carried out without the need of dismantling suspended ceilings or diffusers.

When sizing supply air diffusers with demands of very low sound levels or short throw lengths, a low total pressure drop over the diffuser is also obtained. This means the commissioning pressure Δp_i becomes low, and difficulties with the pressure measurement arise.

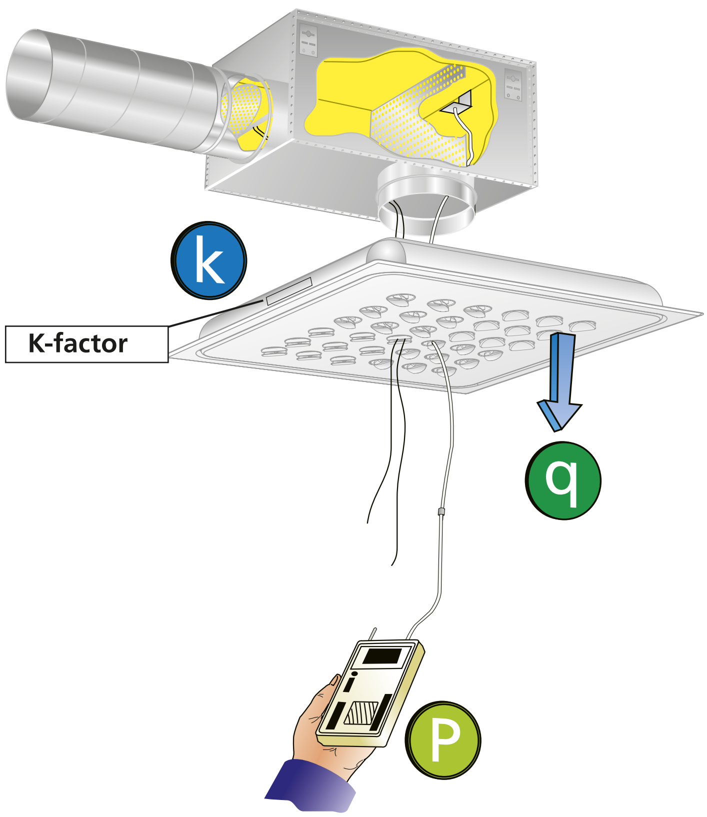


$$q = k \cdot \sqrt{p}$$

$$p = \left(\frac{q}{k}\right)^2$$

q = measured flow l/s
p = read measurement pressure (Pa)
k = commissioning factor

Example on how to adjust low velocity diffusers



$$q = k \cdot \sqrt{p}$$

$$p = \left(\frac{q}{k}\right)^2$$

q = measured flow l/s
p = read measurement pressure (Pa)
k = commissioning factor

Example of air diffuser commissioning with control cord and measurement tube

Commissioning chilled/climate beams and comfort modules

In comparison with a standard air diffuser, commissioning climate beams and comfort modules is more extensive as it also requires adjustment of the water flow for cooling and possibly heating.

The description below only shows how adjustment of the beam/comfort module can be performed. When it concerns adjustment of the entire distribution system, refer to the consultant who designed the system.

Commissioning the supply airflow

The extent of commissioning varies depending on the type of system in question. The simplest is for a ring feed system, where the pressure in the feed duct is low and constant. In these cases, the correct airflow can be set directly on the nozzles in the beam or comfort module.

In more common types of system with main ducts, branch ducts and adjustment dampers, the following workflow can be applied:

1. Check that the product to be adjusted is the right one for the particular application (the right type, size, nozzle configuration and setup of air deflectors).

2. Look up the required commissioning pressure for the specific product in the supplier's sizing software.

3. Connect a pressure gauge to the requisite pressure tapping and measure the static commissioning pressure.

4. Adjust the adjustment damper until the planned commissioning pressure is set. The airflow will then also be correct.

Commissioning the water flow

1. Check that the product to be adjusted is the right one for the particular application (the right type, size, nozzle configuration and setup of air deflectors).

2. Look up the design water flow and pressure drop across the coil from the supplier's sizing software.

3. The pressure drop across the valve must be at least as high as the pressure drop across the coil in order for the valve to have sufficient authority.

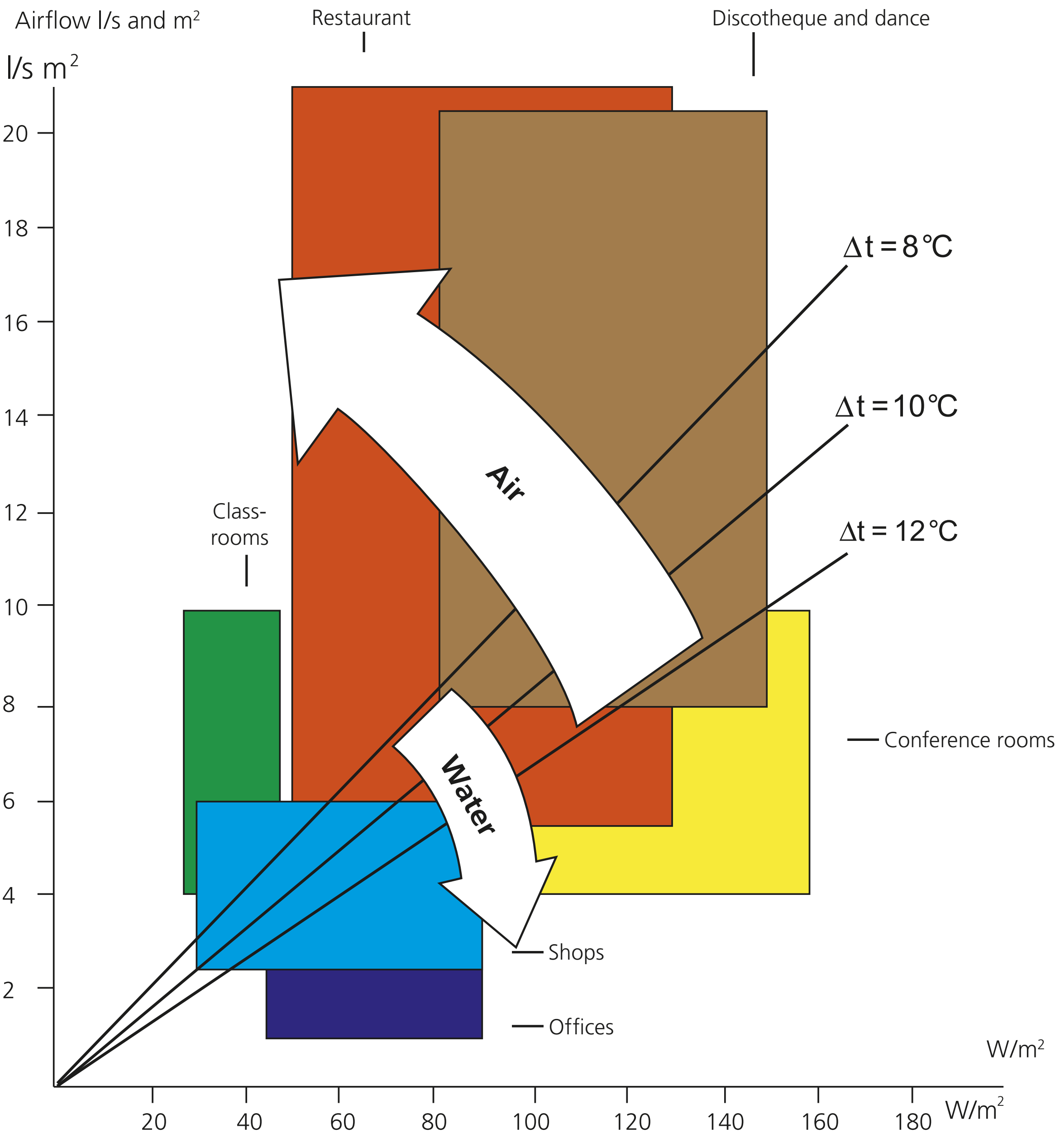
4. Depending on the pressure drop required across the valve and the required water flow, the valve's k_v value can be calculated. Example: 0.05 l/s at 5 kPa...

k_v=V (m^3/h)/ $\sqrt{\Delta p}$ (Bar) = 0.05x3.6/ $\sqrt{0.05}$ = 0.80

The k_v value must therefore be set to 0.80 or less for the valve to have authority.

Airflows and cooling capacity levels for different activities

The figure illustrates schematically the airflow and cooling effects required by different activities.



The airflow ranges in the diagram are based on carbon dioxide levels of 600 ppm (max flow) and 1000 ppm (minimum flow). The background level is assumed to be 350 ppm. Carbon dioxide production is 18 l/h and person.

Example
Conference room with a heat surplus of 60 W/m² has an airflow need of at least 4 l/s and m².
a) If the airflow 4 l/s and m² are chosen a solution with water as the energy carrier is required. A solution of convection beams can then ideally be selected.
b) If instead the airflow is increased to 6 l/s and m², the cooling load can be managed with an airborne option. In this case improved air quality is also obtained.

Example of the airflow requirement for different premises

Type of premises	Pers/m²	Airflow in l/s, m² for CO_2 content 600 ppm	800 ppm	1000 ppm
Office 1)				
- single person	0.1	2.0	1.1	0.8
- large office	0.12	2.4	1.3	1.0
- conference room	0.5	10.0	5.6	3.8
Schools				
- classroom	0.5	10.0	5.6	3.8
- laboratory	0.3	6.0	3.3	2.3
- assembly hall	1.5	30.0	16.7	11.5
- gymnasium	0.3	6.0	3.3	2.3
Library	0.2	4.0	2.2	1.5
Shop	0.2-0.3	6.0	3.3	2.3
Restaurant				
- dining room	0.7	14.0	7.8	5.4
- cafeteria	1.0	20.0	11.1	7.7
- bar	1.0	20.0	11.1	7.7
Nursing home 2)	0.4	3.8	2.2	1.5
Discotheque 3)	1.0	27.2	15.1	10.5
Waiting Hall/Lobby	1.5	30.0	16.7	11.5

Airflow requirement:
1) Average weight 70 kg, sitting
2) Average weight 25 kg, walking
3) Average weight 70 kg walking

Background level CO_2 = 350 ppm

Factual information

Terms and definitions

A-weighted sound level
Sound pressure level is determined with a sound level meter that has the A-filter engaged. Written dB(A).

Decibel
Decibel (dB), one tenth of a bel (B), is a logarithmic measure commonly used to define and measure the signal and noise level. A normal conversation is usually around 60-70 dB.

Density
Density (specific weight) expresses mass per unit volume. For gases the unit 1 kg/m^3 is used.

Equivalent sound absorption area
A spatial equivalent absorption area is a measure of the limitation surfaces' area multiplied by their average absorbency capacity.

Power
International unit 1 watt (1 W). The unit is used for all forms of power, for example, electric power, mechanical power and heating power.

Energy
The international unit is 1 Joule (1 J). This unit is henceforth used for, among others, thermal energy. For electrical energy the unit 1 kWh is normally used.

Flow
Flow is expressed per unit of time 1 second (1 s). Volume per unit of time is given in m^3/h , m^3/s or l/s. In some contexts, there is need to use the concept of mass flow, which is usually indicated in g/s or kg/s

Frequency
In an acoustic context, frequency is the number of variations in pressure per second. Frequency has the unit Hertz (Hz).

Sound absorption
The reduction of sound energy (conversion to thermal energy in absorbent material).

Sound power, sound power level
The sound power, measured in W, is the power supplied to the air and causes variations in pressure (sound). The logarithmic function is called sound power level and usually has the unit dB. Unit B also occur occasionally (1 B = 10 dB).

Sound pressure, sound pressure level
Sound pressure, measured in Pa, is a measure of the magnitude of pressure fluctuations in the air. The logarithmic function is called sound pressure level and the unit dB.

Mass - weight - force
The international mass unit is 1 kilogram (1 kg). The unit kg should only be used to specify the material content of a body, i.e. mass. This will remain unchanged no matter how the body moves on the earth or in space. The word weight should be avoided as a synonym for mass where there is the potential for misunderstanding.

Weight refers to the force of gravity on the mass and is consequently not a synonym for mass. The weight of a body varies if it is moved between different locations on earth. In a satellite where the G-force = 0, the body is weightless (not less weight), but still retains its mass.

The international power unit is 1 Newton (1 N). 1 N is the force needed to cause a 1 kg mass to accelerate 1 m/s^2.

Octave band
A standardized division of the frequency ranges. Octave bands are named according to their mid-frequencies.

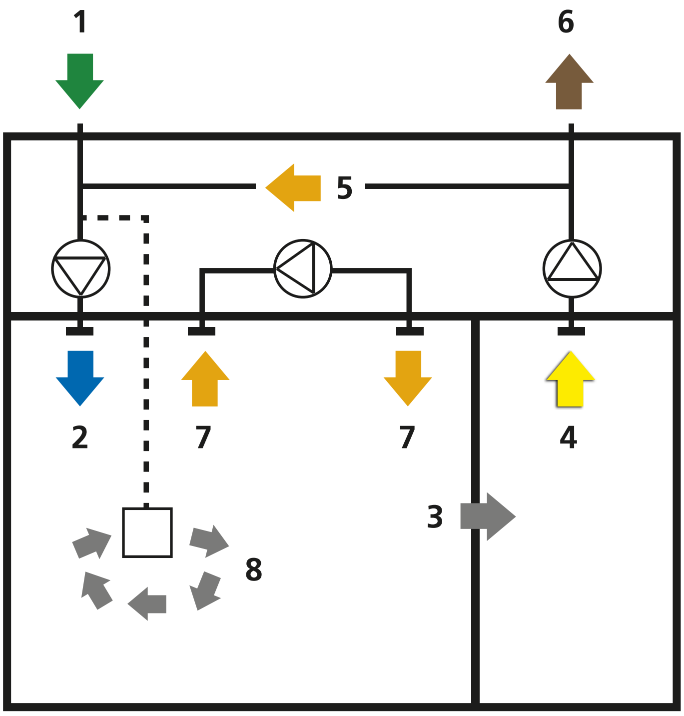
Temperature
For absolute temperature the unit 1 Kelvin (1 K) is used. Usually in HVAC engineering the unit 1 degree Celsius (1 °C) is used. Temperature difference is indicated in unit 1 degree. The unit 1 degree indicates the difference in temperature 1 °C or 1 K.

Total sound power level, $L_{w,tot}$
The logarithmic sum of sound power levels in octave bands 125-8000 Hz. Used as the baseline value for calculating the sound power in an octave band when specifying sound generation.

Pressure
Pressure is force per surface area. The unit of pressure is Pascal, Pa. In some cases, this unit gives impractically large numbers. The unit 1 bar = 100 kPa can be used. For barometric height the unit 1 millibar (1 mbar) is used. That unit is used in meteorology.

Velocity
The unit of rotational speed in SI system is 1 radian per second (1 rad/s). This unit gives a fully divergent concept towards the unit 1 r/min. 1 r/m = 2π/60 rad/s. A move to the unit 1 rad/s will be made when motor manufacturers introduce this unit.

Ventilation terminology



Terminology and colours for different types of air according to EN 13779:2007:

1	Outdoor air	Green
2	Supply air	Blue
3	Transfer Air	Grey
4	Extract air	Yellow
5	Recirculated air	Orange
6	Exhaust air	Brown
7	Circulation air	Orange
8	Induction air	Grey

Many ventilation product manufacturers do in fact use other colours to describe the air paths. In the Nordic countries, for example, it is common to use red for the supply air, as it is imagined it is usually heated.

Efficiency concepts for air

- Ventilation efficiency is a measure of how efficiently a pollutant is removed.
- Air exchange efficiency is a measure of how efficiently the air in a room is exchanged.

A main goal for designers is therefore to size and position the supply and extract air diffusers so that both the air exchange and ventilation efficiency are as high as possible.

The ventilation efficiency depends on a number of different parameters:

- Placement of supply air and extract air diffusers.
- Type of diffuser.
- Supply air velocity.
- Temperature difference between supply and extract air.
- Occurrence of disturbances, for example, heat sources, activity, etc.

According to the proposal from the Nordic Ventilation Group the term "specific airflow" (n) is introduced instead of the term "rate of air change".

Specific air flow denotes the relationship between the supply airflow's share of outside air and the ventilated room volume. Specific flow was previously called "rate of air change" with the unit chge/h. However, this has often led to the perception that the air in the room is changed so many times per hour that the number indicates. How quickly the air in the room is changed is determined not only by the size of the supply airflow and the room volume, but also largely by how the air flows in the room.

Excess heat can be considered as an impurity. It is therefore appropriate to introduce the concept "temperature efficiency".

As excess heat can be considered an impurity, we can therefore exchange concentration with temperature to give temperature efficiency.

We distinguish between "average temperature efficiency" which applies as an average value for the whole room and the "local temperature index" that apply to a particular point in the room.

Ventilation efficiency, ε_{rc}

Defined at a given contamination discharge as the ratio between the concentration in the extract air, and average concentration in the room, i.e.

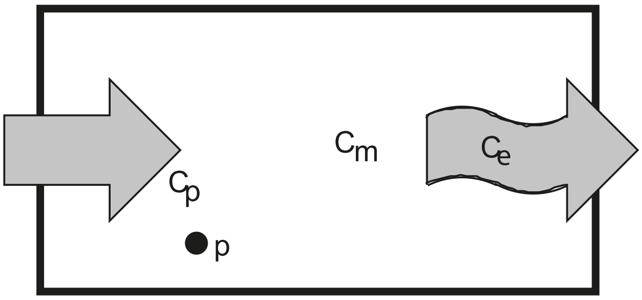
$$\varepsilon_{rc} = C_e/C_m \; 100\%$$

where C_e = equilibrium concentration in the extract air
where C_m = mean concentration in the room at equilibrium

Local air quality index, ε_{pc}

where C_p = equilibrium concentration at point p

$$\varepsilon_{pc} = C_e/C_p \; 100\%$$



Air exchange efficiency, ε_{ra}

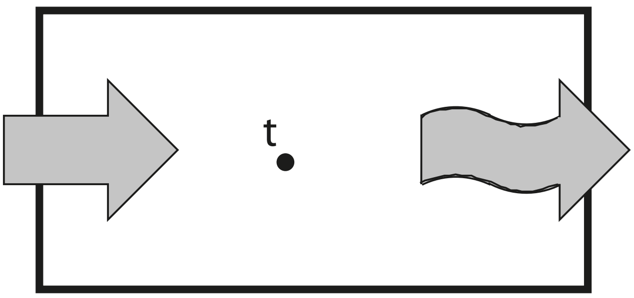
Defined as the ratio between the nominal time constant and the exchange time for the air in the room.

$$\varepsilon_{ra} = \tau_n / (2 \cdot \tau_m) \; 100\%$$

where τ_n = nominal time constant
τ_m = air's average age in the room
$2 \cdot \tau_m$ = exchange time for air in the room

Note:
The average age of the air is directly related to the time it takes to replace the air in the room.
In order to replace all the air in the room, it takes on average a period equal to twice the average age of the air in the room.

The average age of the air can be determined by measuring in the extract air duct.



Specific airflow, n

$$n = \frac{q}{V} \quad \frac{m^3/h}{m^3}$$

where q = outdoor air flow (m^3/h)
V = room volume (m^3)

Nominal time constant, t_n

Nominal time constant (t_n) is the time that the supplied ventilation flow's share of outside air q, on average stays in the room.

$$t_n = \frac{V}{q}$$

where q = outdoor air flow (m^3/h)
V = room volume (m^3)

Temperature efficiency, ε_{rt} (mean value)

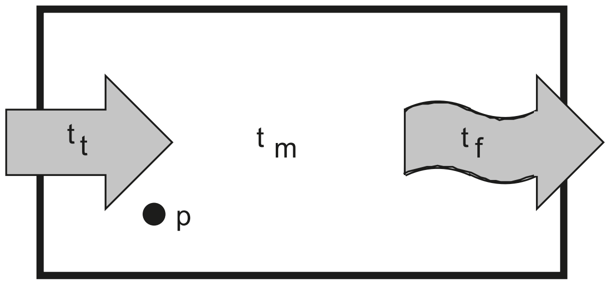
$$\varepsilon_{rt} = (t_f - t_t) / (t_m - t_t) \; 100\%$$

where t_f = extract air temperature
t_m = the rooms average temperature (at equilibrium)
t_t = supply air temperature

Local temperature index, ε_{pt}

$$\varepsilon_{pt} = (t_f - t_t) / (t_p - t_t) \; 100\%$$

where t_p = temperature at point p at equilibrium



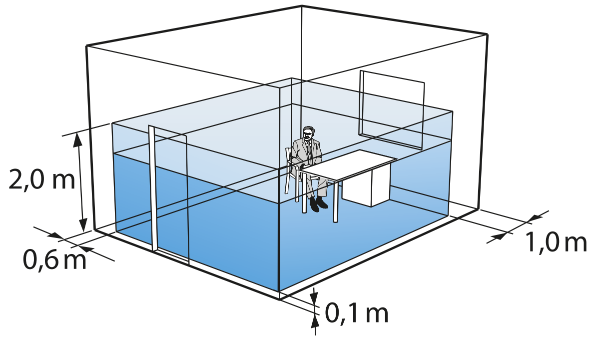
Comfort zones

Occupied zone

The occupied zone is the part of the room, where people are normally present, and is defined in consultation with the developer and architect. Its volume is limited by the planes, which are parallel to walls, ceiling and floor. The distance between the occupied zone plane and room surfaces varies according to room usage.

The table provides a summary of the normal distances.

Room surface	Normally the range of variation of the distance between the surface and the occupied zone (Swedish Boverket's recommendations in parentheses)
Outer wall	0.2 - 1.0 m (1.0 m)
Inner wall	0 - 0.6 m (0.6 m)
Floor, lower limit	0 - 0.1 m (0.1 m)
Floor, upper limit standing person	1.8 - 2.0 m (2.0 m)



Coloured surface marks a defined occupied zone

Near zone

The near zone is a concept used in conjunction with low velocity diffusers and is therefore of primary interest in displacement ventilation.

According to the testing regulations SS EN 122 39 the near zone is defined by the dimensions a_v and b_v as shown to the right.

The measurement a_v represents the greatest horizontal distance from the wall (or the middle of the diffuser for a cylindrical diffuser) to the isovel[1)] for v m/s.

The measurement b_v represents the greatest horizontal distance perpendicular to av between the isovel's end points.

It should be emphasized in the test method that the isovel must be measured where the velocity is greatest, i.e. not at a specific distance from the floor. Velocity v m/s for the isovel has in the test method been set to:

- 0.2 m/s for low velocity diffusers designed for comfort ventilation.
- 0.3 m/s for low velocity diffusers designed for industrial ventilation.

For ceiling mounted low velocity diffusers, the near zone is defined as illustrated in the bottom right.

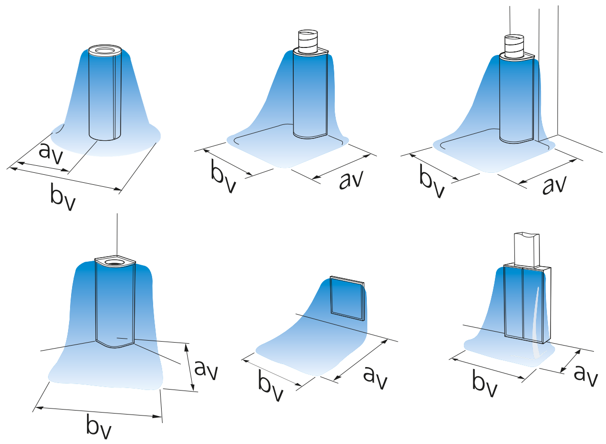
It is important that the ventilation designer is observant of how different manufacturers specify the near zones. Different methods are used. For example:

- Comfort zone, which has its own special definition.
- The isovel at the level 0.05 m above the floor.
- The isovel at the level 0.10 m above the floor.

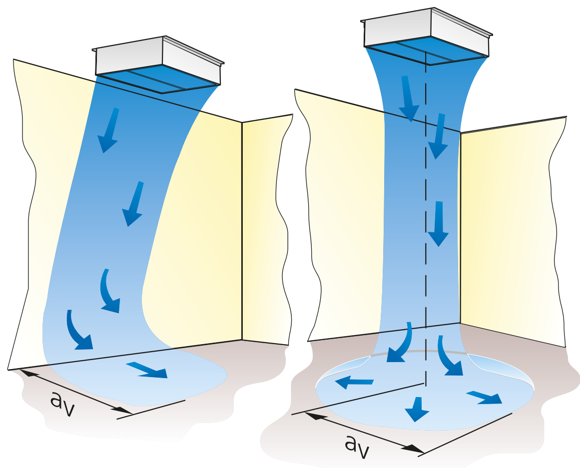
Relatively large differences are obtained in the a_v and b_v values when we abandon the agreed measurement and reporting methods in the test method.

A deviation from the method always gives shorter values for the near zone!

1) Isovel - The line that connects the same points with the same air velocity.



Floor and wall-mounted low velocity diffusers



Ceiling mounted low velocity diffusers

Remote zone

The remote zone is a concept used in conjunction with displacement ventilation. Remote zone is defined as the zone outside the near zone where the density flow prevails. Characteristics for density flow are:

- it is driven by the density difference between the supply air and the room air
- it gives small co-ejection of the ambient air
- it is very thin, usually about 10 cm
- it has slightly lower velocity fluctuations (turbulence) than a jet

Air velocities in the remote zone are determined by:

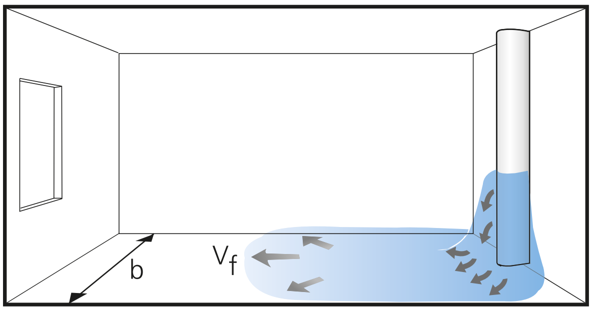
- thermal load in the room
- room geometry (width)

The air speed in the remote zone is calculated when we have an equalisation of the airflow over the width of the room.

The air velocity in the remote zone calculated using the following equation:

$$v_f = \left(\frac{9{,}81 \cdot q \cdot \Delta t}{b \cdot T}\right)^{1/3} \quad \text{(m/s)}$$

Where q = supply airflow (m^3/s)
Δt = difference between room temperature and supply air temperature (K)
b = room width (m)
T = room's absolute temperature (K)
v_f = air velocity in the remote zone (m/s)



Remote zone in rooms with displacement air diffusers.

Example:
Offices of the width 3.6 m
Semi-circular low velocity diffuser placed at the rear edge
Supply air temperature 18 °C
Room temperature 24 °C
Air flow 30 l/s

$$v_f = \left(\frac{9{,}81 \cdot 0{,}030 \cdot 6}{3{,}6 \cdot 297}\right)^{1/3}$$

$$v_f = 0{,}12 \text{ m/s}$$

This air velocity should be considered as the lowest value that can be obtained where a density flow prevails.

Conversion factors, symbols and units

Conversion factors

The table comprises a selection of the most common units in fan and air conditioning technology. Conversion factors are truncated where appropriate to three decimal places.

For practical purposes, useful approximations with an error not exceeding 2% are given in parentheses.

Magnitude	Designa-tion	SI unit	Previous unit	Conversion factor	
Power	**F**	N	kp	1 N = 0.102 kp (1 N 0.1 kp)	1 kp = 9.807 N (1 kp 10 N)
Pressure	**p**	Pa	mm wg	1 Pa = 0.102 mm wg (1 Pa 0.1 mm wg)	1 mm wg = 9.807 Pa (1 mm wg 10 Pa)
		bar	kp/cm^2	1 bar = 1.020 kp/cm^2 1 bar 1 kp/cm^2	1 kp/cm^2 = 0.981 bar (1 kp/cm^2 ≈1 Bar)
		mbar	dry[1)]	1 mbar ≈ 0.750 dry (1 000 mbar ≈ 760 mm Hg	1 dry ≈1.333 mbar
Flow	**q**	m^3/s	m^3/h	1 m^3/s = 3 600 m^3/h	1 m^3/h = 0.278 x $10^{-3} m^3/s$ (1 000 m^3/h ≈ 0.28 m^3/s)
Power	**P**	kW kW	hp kcal/h	1 kW = 1.360 hp 1 kW = 860 kcal/h	1 hp = 0.736 kW al/h = 1.163 x 10^{-3}kW
Energy	**W**	kJ	kcal	1 kJ = 0.239 kcal	1 kcal = 4.187 kJ
Enthalpy	**in**	kJ/kg	kcal/kg	1 kJ/kg = 0.239kcal/kg	1 kcal/kg = 4.187 kJ/kg
Specific heat capacity	c_p	kJ/kg °C	kcal/kg °C	1 kJ/kg °C = 0.239 kcal/kg °C	1 kcal/kg °C = 4.187 kJ/kg °C
Thermal conductivity	λ	W/m °C	kcal/m °C	1 W/m °C = 0.860 kcal/m °C h	1 kcal/m °C h = 1.163 W/m °C
Thermal transmittance	**U**	W/m^2 °C	$kcal/m^2$ °C h	1 W/m^2 °C = 0.860 1 $kcal/m^2$ °C h	1 kcal/m2 °C h = 1.163 W/m^2 °C

1) 1 dry = 1 mm Hg at 0°C and g = 9.80665m/s^2.

Conversion factors

PRESSURE				
Pa (= 10^{-2} mbar)	kp/cm^2	mmvp	lb/in^2 (psi, pound per square inch)	in of water (inch of water)
1	$10.20 \cdot 10^{-6}$	0.1020	$0.1450 \cdot 10^{-3}$	$4.015 \cdot 10^{-3}$
98.07 · 103	1	104	14.22	393.7
9.807	10^{-4}	1	$1.422 \cdot 10^{-3}$	$39.37 \cdot 10^{-3}$
$6.895 \cdot 10^{3}$	$70.31 \cdot 10^{-3}$	703.1	1	27.68
249.1	$2.540 \cdot 10^{-3}$	25.40	$36.13 \cdot 10^{-3}$	1

ENERGY				
J (= Ws)	kpm	kcal	kWh	Btu (British Thermal Unit)
1	0.1020	$0.2388 \cdot 10^{-3}$	$0.2778 \cdot 10^{-3}$	$0.9478 \cdot 10^{-3}$
98.07	1	$2.342 \cdot 10^{-3}$	$2.724 \cdot 10^{-6}$	$9.295 \cdot 10^{-3}$
4.187 · 10/	426.9	1	$1.163 \cdot 10^{-3}$	3.968
$3.6 \cdot 10^{6}$	$0.3671 \cdot 10^{6}$	859.8	1	$3.412 \cdot 10^{3}$
1.055 · 10/	107.6	0.2520	$0.2931 \cdot 10^{-3}$	1

POWER					
W (= J/s)	kW (= kJ/s)	kcal/h	hp	Btu/h	TR (ton of refrigeration)
1	10^{-3}	0.8598	$1.36 \cdot 10^{-3}$	3.412	$0.2843 \cdot 10^{-3}$
10/	1	0.8598 · 10/	1.360	3.412 · 10/	0.2843
1.163	$1.163 \cdot 10^{-3}$	1	$1.581 \cdot 10^{-3}$	3.968	$0.3307 \cdot 10^{-3}$
735.5	0.7355	632.4	1	2.510 · 10/	0.2091
0.2931	$0.2931 \cdot 10^{-3}$	0.2520	$0.3985 \cdot 10^{-3}$	1	$83.33 \cdot 10^{-6}$
3.517 · 10/	3.517	3.024 · 10/	4.783	12 · 10/	1

Conversion factors

LENGTH		
m	in (inch)	ft (feet)
1	39.370	3.281
$25.4 \cdot 10^{-3}$	1	$83.33 \cdot 10^{-3}$
0.3048	12	1

AREA	
m²	sq ft (square feet)
1	10.76
0.09290	1

VOLUME		
m³	ft³	US gallon
1	35.32	264.2
$28.32 \cdot 10^{-3}$	1	7.481
$3.785 \cdot 10^{-3}$	0.1337	1

MASS	
kg	lb (pound)
1	2.205
0.4536	1

VOLUME FLOW			
m³/s	l/s	m³/h	cfm (cubic feet per minute)
1	10/	3600	2119
10^{-3}	1	3.6	2.119
$0.2778 \cdot 10^{-3}$	0.2778	1	0.5886
$0.4720 \cdot 10^{-3}$	0.472	1.699	1

VELOCITY	
m/s	fpm (feet per minute)
1	196.9
$5{,}080 \cdot 10^{-3}$	1

Symbols and units

ROOM		
Symbol		Unit
l	Length	m
b	Width	m
h	Height	m
δ	Thickness	m
r	Radius	m
d	Diameter	m
A	Area	m^2
V	Volume	m^3

VENTILATION AND HEATING		
Symbol		Unit
C_P	Specific heat capacity	kJ/kg · K
C	Radiation coefficient	$W/m^2 \cdot K^4$
d_h	Hydraulic diameter	m
d_e	Equivalent diameter	m
E	Energy	J
F	Power	N
g	Acceleration of gravity	m/s^2
h	Enthalpy	J/kg
m	Mass	kg
P	Power	W = J/s
Pr	Prandtl number	
p	Pressure	$Pa = N/m^2$
p_d	Dynamic pressure	Pa
p_s	Static pressure	Pa
P_{atm}	Atmospheric pressure	Pa, mbar
p_t	Total pressure	Pa
Δp	Pressure difference	Pa
q	Volume flow	m^3/s
r	Heat of vaporisation	kJ/kg
R	Thermal resistance	$m^2 \cdot K/W$
T	Temperature, Kelvin	K
t	Temperature, Celsius	°C
t	Time	s
ΔT	Temperature difference	K
Δt	Temperature difference	°C
U	Coefficient of thermal transmit-tance	$W/m^2 \cdot K$
v	Velocity	m/s
α	Coefficient of thermal transmit-tance	$W/m^2 \cdot K$
ε	Emission factor, efficiency	
ν	Kinematic viscosity	m^2/s
ρ	Density	kg/m^3
φ	Relative humidity	%
η	Efficiency	
λ	Thermal conductivity	W/m · °C

ACOUSTICS		
Symbol		Unit
A	Absorption area	m^2 Sabine
c	Speed of sound	m/s
D	Level difference	dB
f	Frequency	Hz
in	Intensity	W/m^2
L	Sound level	dB ref $2 \cdot 10^{-5}$ Pa
L_A	A-weighted sound pressure level	dB ref $2 \cdot 10^{-5}$ Pa
L_p	Sound pressure level	dB ref $2 \cdot 10^{-5}$ Pa
L_W	Sound power level	dB ref 10^{-12} W
L_{WA}	A-weighted sound power level	dB ref 10^{-12} W
L_i	Sound intensity level	dB ref 10^{-12} W/m^2
L_{eq}	Equivalent sound level	dB ref $2 \cdot 10^{-5}$ Pa
Q	Directivity factor	
R	Room constant	
R	Reduction index	dB
T	Reverberation time	s
α	Absorption factor	
λ	Wavelength	

AIR EXCHANGE		
Symbol		Unit
ε_{rc}	Ventilation efficiency	%
ε_{pc}	Local ventilation index	-
ε_{ra}	Air exchange efficiency	%
ε_{rt}	Temperature efficiency	%
ε_{pt}	Local temperature index	-
τ_n	Nominal time constant	h
τ_m	Room air's mean age	h
τ_r	Exchange time of air in the room	h
n	Specific airflow	Room volume/h

Heating data

Air, p = 1 bar

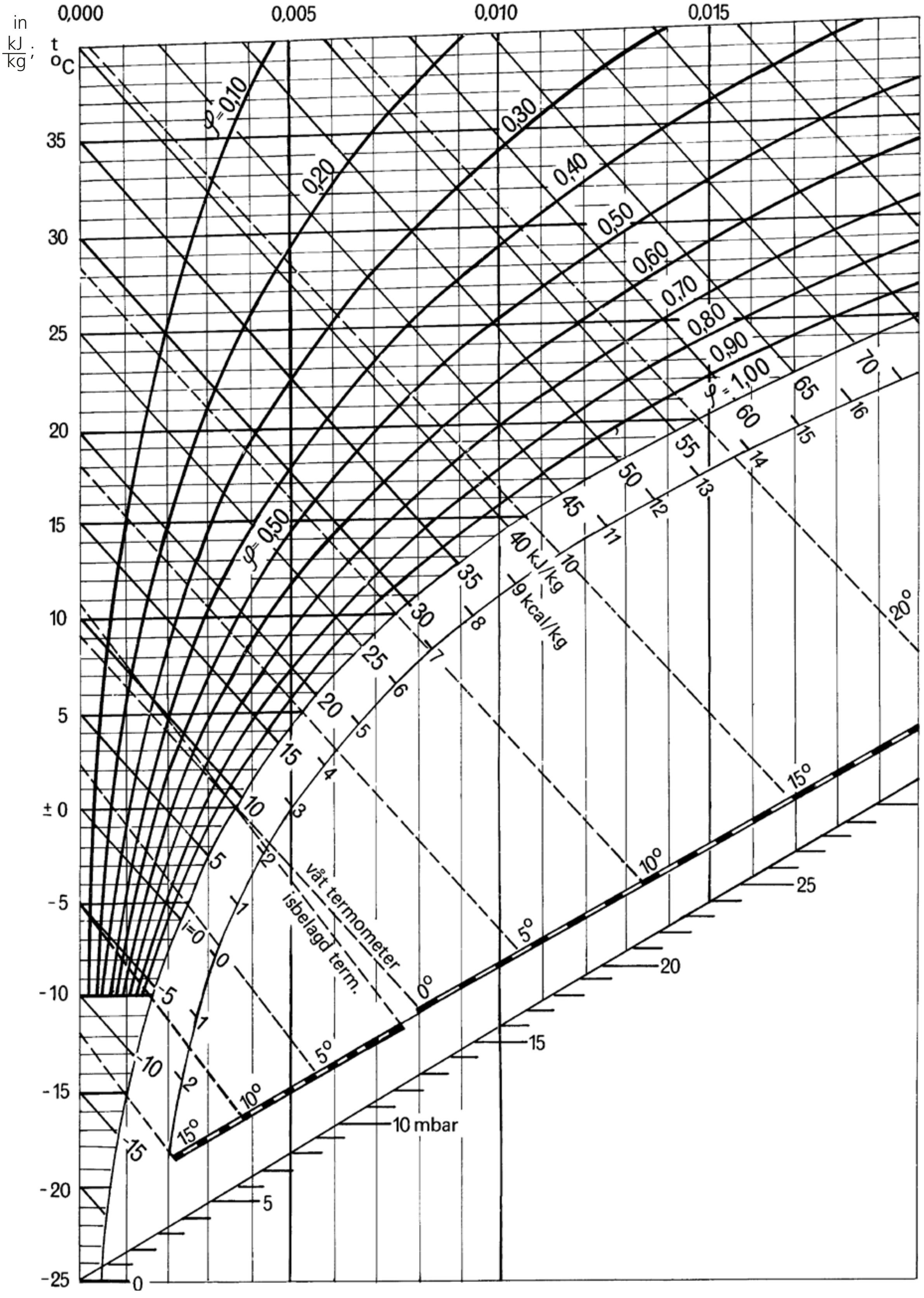
Temperature °C	Density kg/m^3	Spec.heat kJ/kg degree	Thermal conductivity W/m degree
0	1.275	1.006	0.0242
20	1.188	1.007	0.0254
40	1.112	1.008	0.0267
60	1.045	1.009	0.0279
80	0.986	1.010	0.0295
100	0.933	1.012	0.0318

Water, p = 1 bar

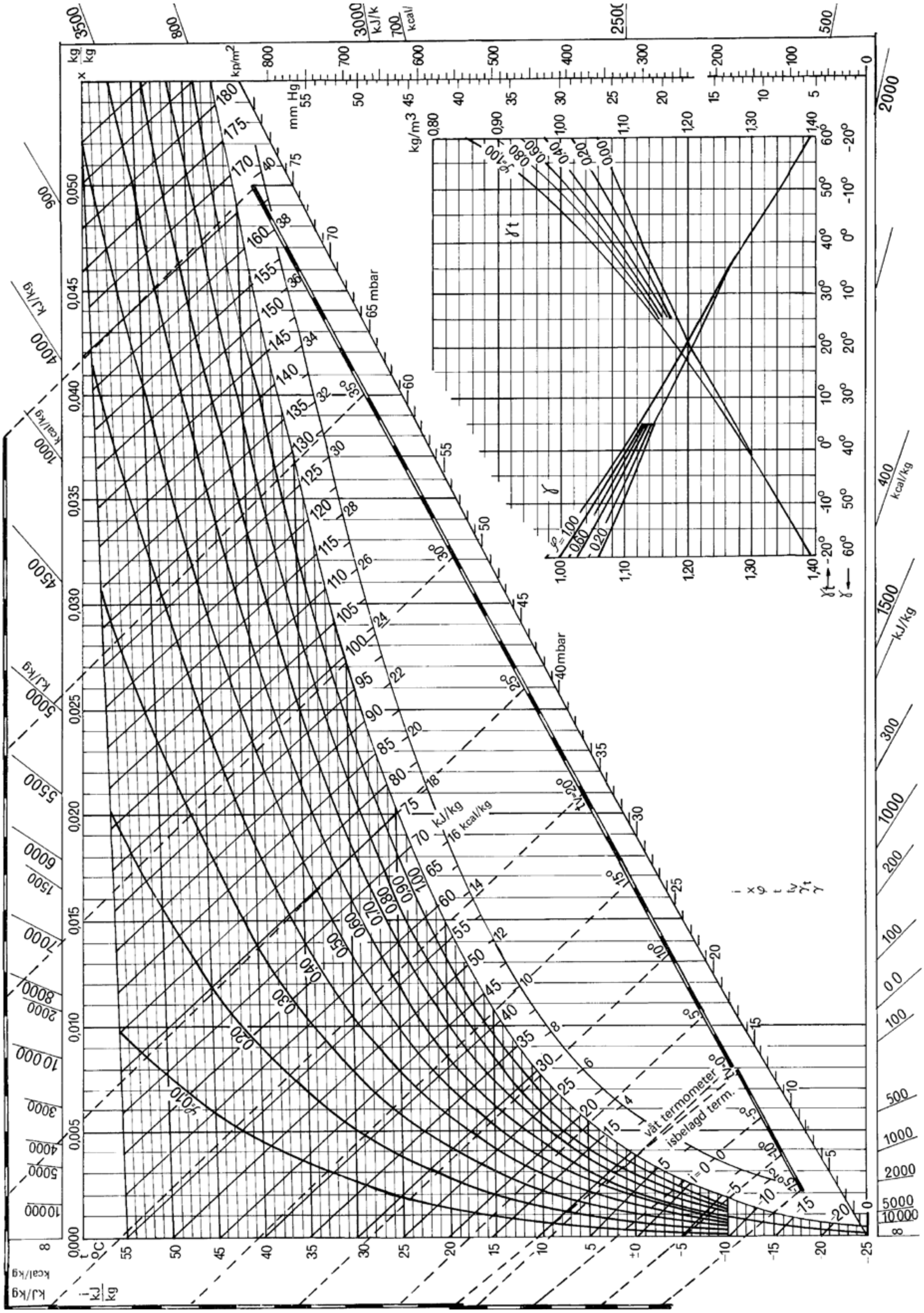
Temperature °C	Density kg/m^3	Spec.heat kJ/kg degree	Thermal conductivity W/m degree
0	999.8	4.212	0.550
20	998.0	4.187	0.599
40	992.2	4.178	0.634
60	983.3	4.180	0.659
80	971.9	4.193	0.675
100	958.4	4.216	0.684

Mollier diagrams

Mollier diagram for humid air -25 to +40 °C



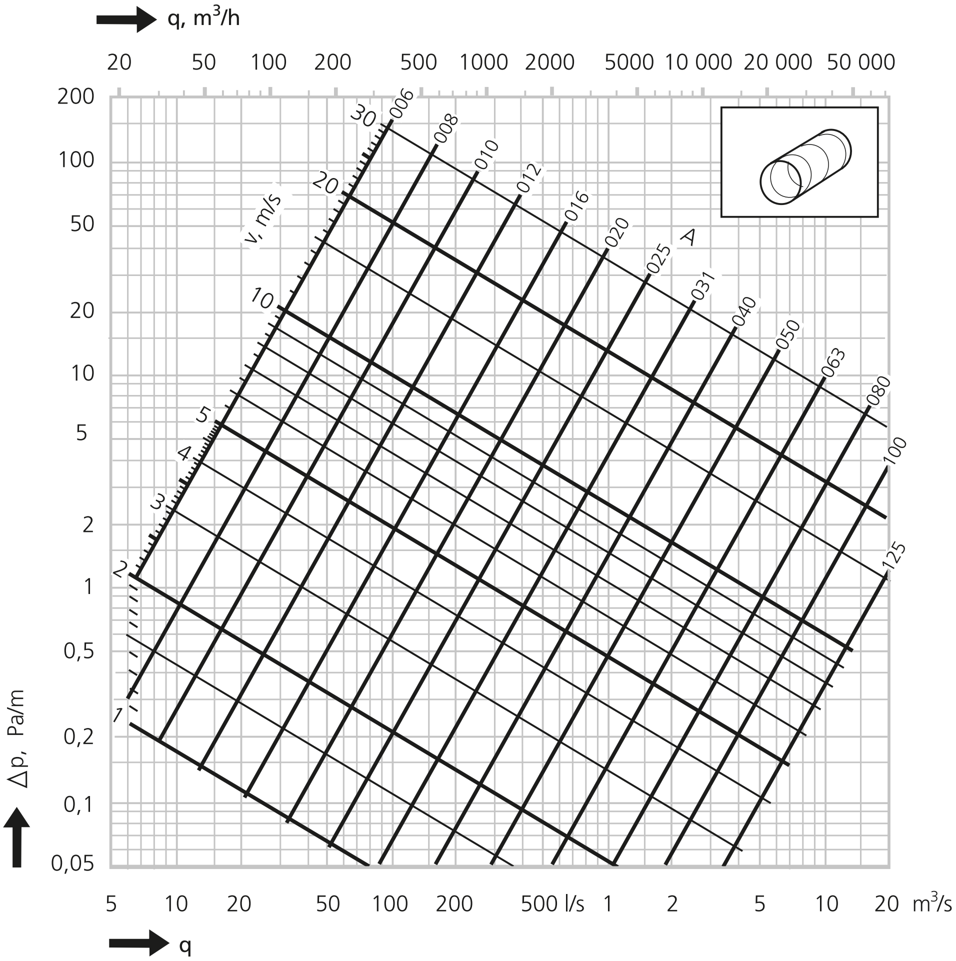
Mollier diagram for humid air –25 till +55 °C



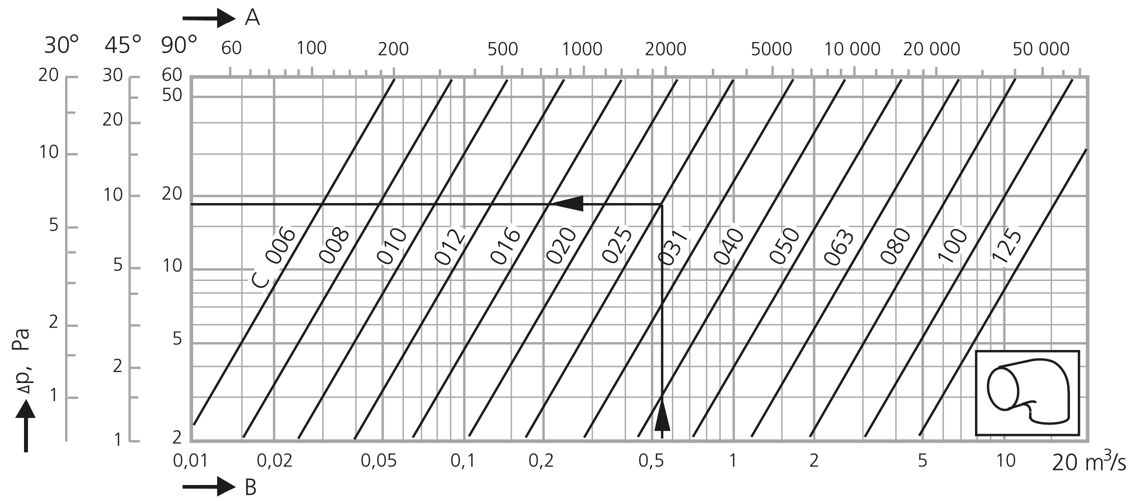
Pressure drop diagrams ducts

In this section pressure drop diagrams for ducts and elbows are presented, both circular and rectangular. The diagrams are used when selecting the duct system.

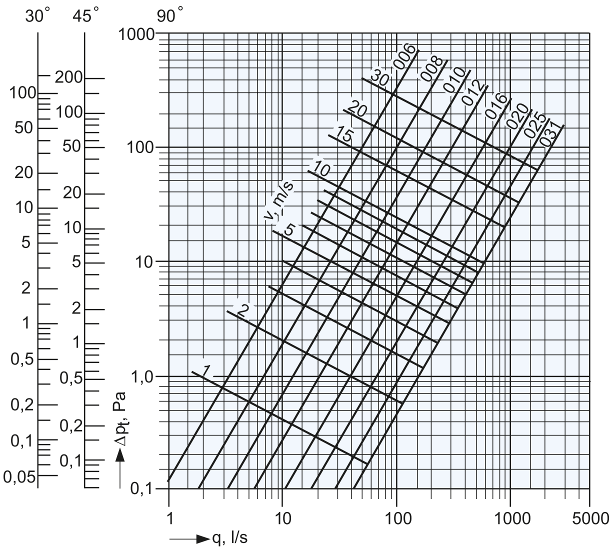
Circular ducts



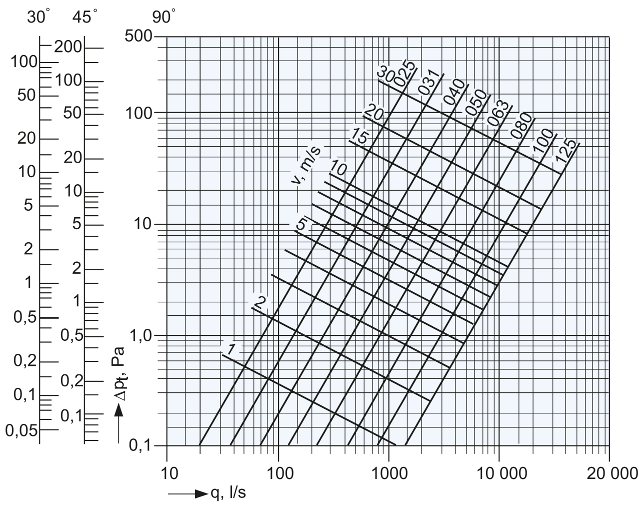
Circular elbow



Pressure drop diagram pressed circular elbow

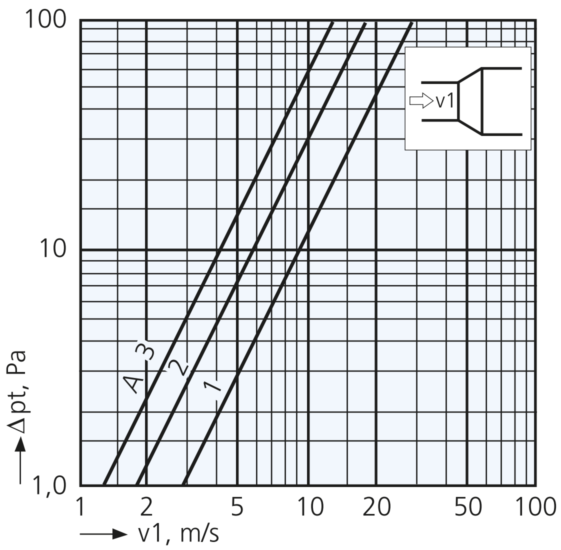


Pressure drop diagram segment built circular elbows



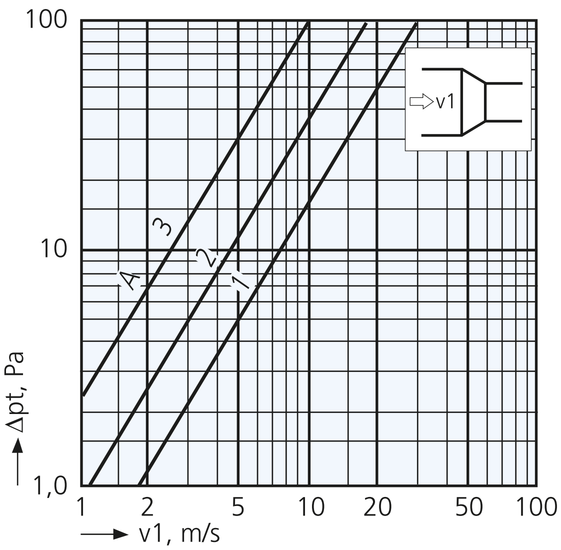
Pressure drop diagram circular dimension changes

Area increase



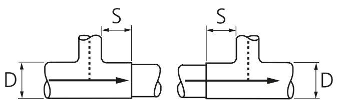
A = Number of dimension increments
(Example from 016 - 020 = 1 dim. increment)

Area reduction

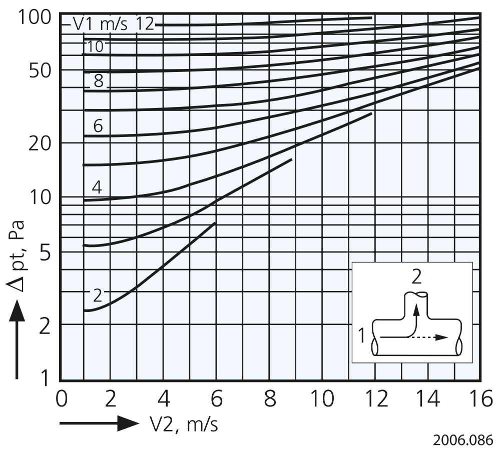


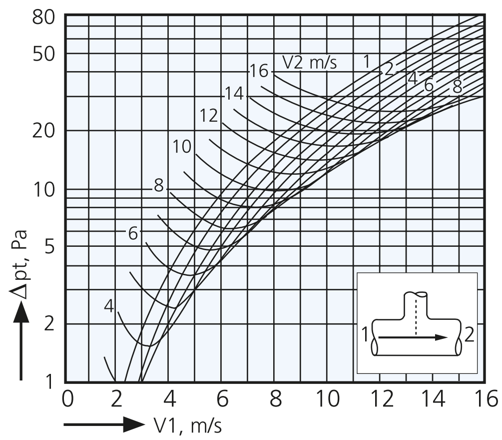
A = Number of dimension increments
(Example from 020 - 016 = 1 dim. increment)

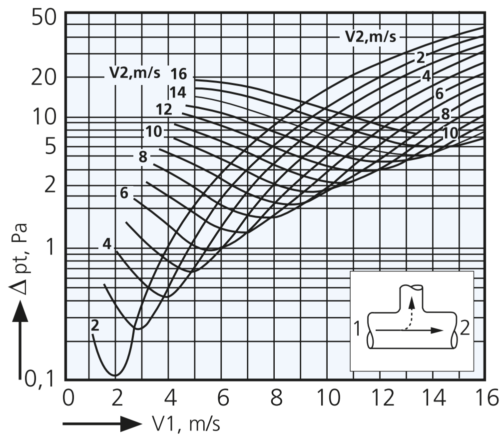
Pressure drop diagram for circular branches, T-pipes and manifolds

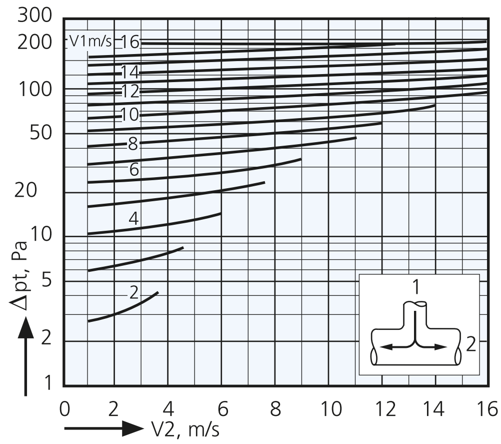


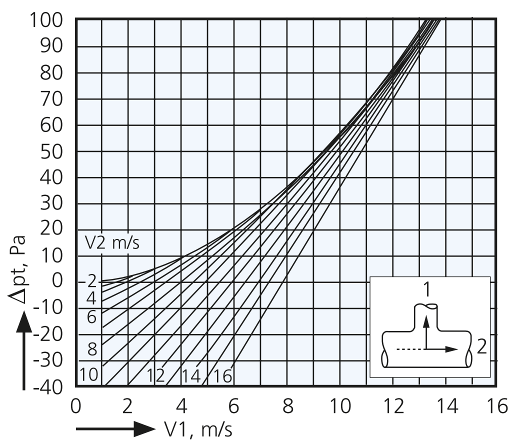
The pressure drop includes any reduction as stated above, if S < 3 x D.

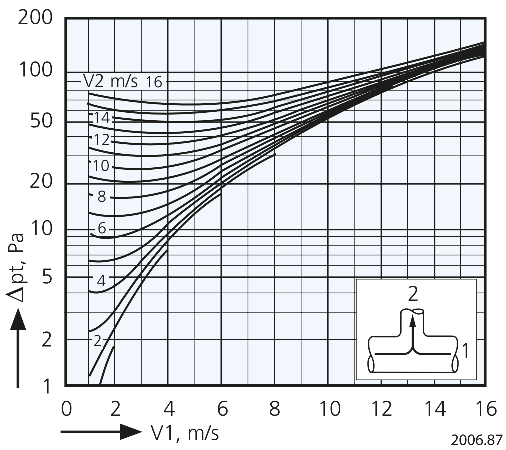




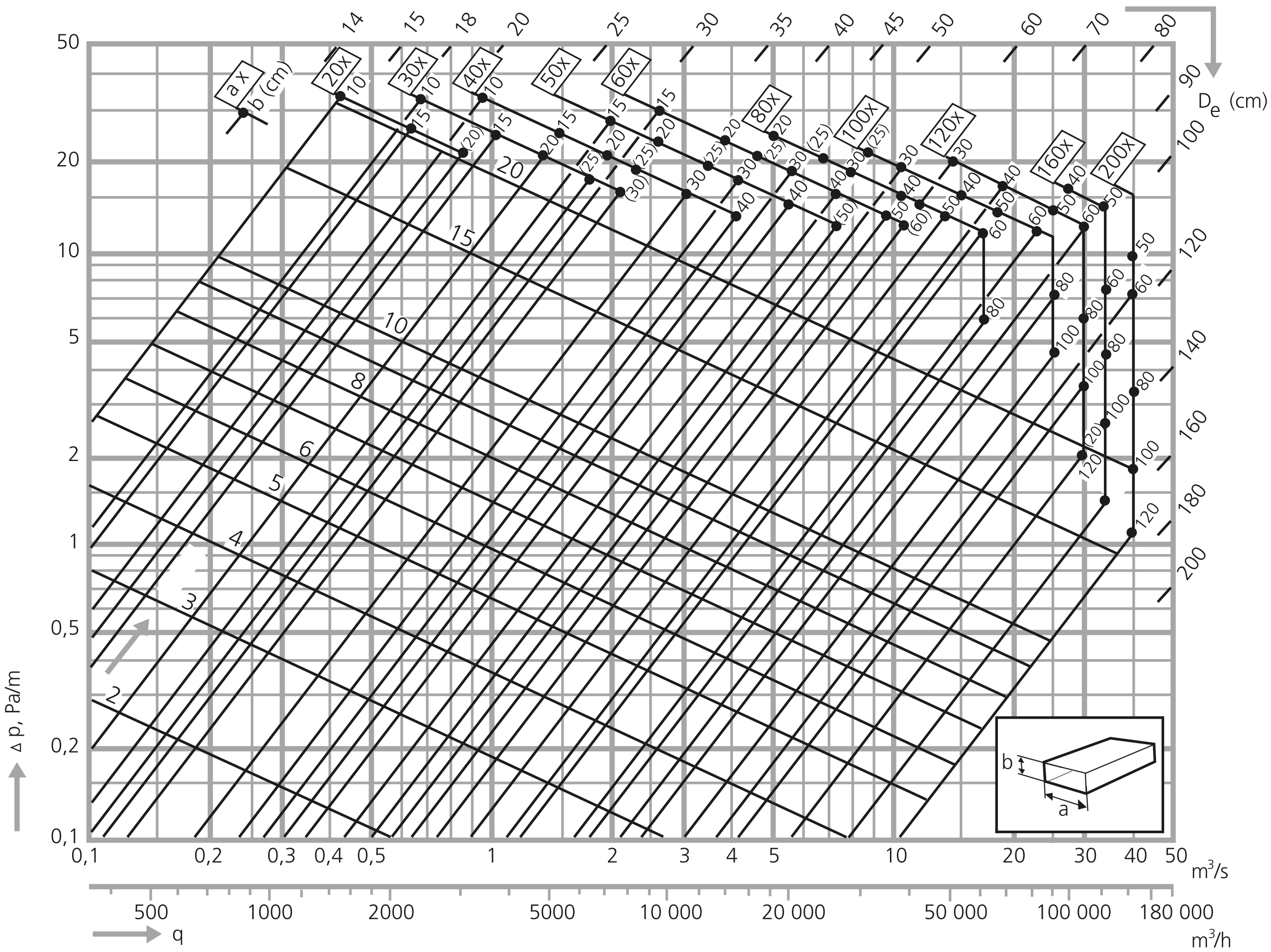






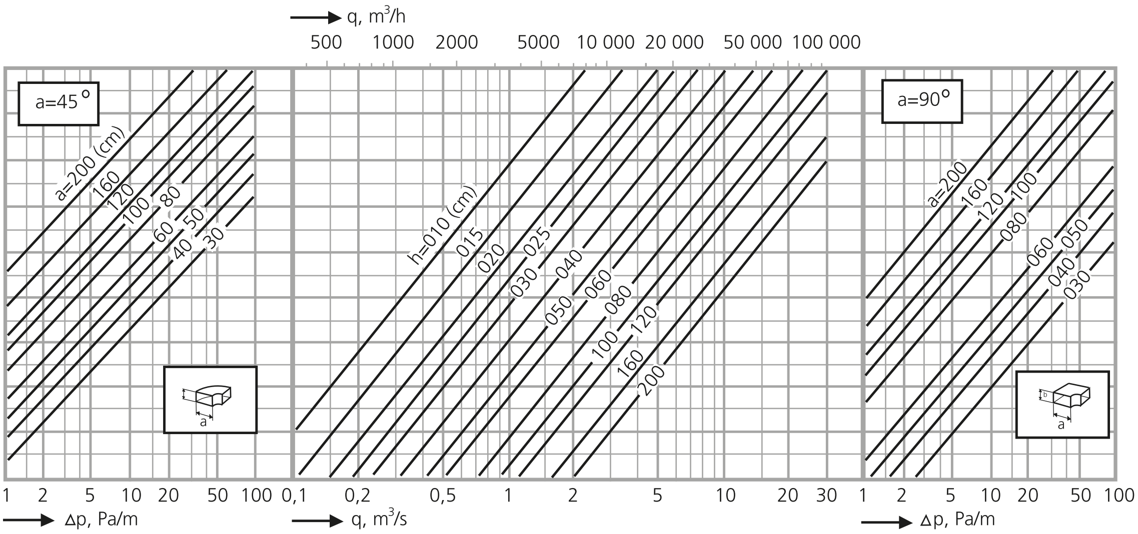


Rectangular ducts



Ducts with the same equivalent diameter De, can have different cross sectional areas. The plotted velocity curves are therefore approximate. Maximum error <5%.

Rectangular elbows



Formulas

Airflow, q m³/s
$q = A \cdot v$ A = cross sectional area, m² v = air velocity, m/s

Dynamic pressure, p_d Pa
$P_d = \rho \cdot v^2/2$
ρ = air density, kg/m³
v = air velocity, m/s

Hydraulic diameter, d_h m
$d_h = 4 \cdot A/O$ A = cross sectional area, m² O = duct circumference, m
d_h = for rectangular ducts $d_h = 2 \cdot a \cdot b / (a + b)$ a and b are the duct's sides
d_h for circular ducts d_h = d = duct diameter

Total pressure – supply air, p_t Pa
$p_t = p_s + p_d$ p_s = static pressure, Pa p_d = dynamic pressure, Pa

Total pressure – extract air, p_t Pa
$p_t = (-p_s) + p_d$ p_s = negative static pressure, Pa p_d = dynamic pressure, Pa

Cross-section circular duct, A m²
$A = \pi \cdot d^2 / 4$ d = diameter of the duct, m

Circumference of a circular duct, O m
$O = \pi \cdot d$
d = diameter of the duct, m

Air density, ρ kg/m³
$\rho_t = 1.293 \cdot B / 1013 \cdot 273 / (273 + t)$
B = barometric height, mbar
t = air temperature, °C

Cooling/heating capacity for air, P kW
$P = q \cdot \rho \cdot Cp \cdot \Delta t$ q = airflow, m³/s ρ = air density, kg/m³ Cp = air's specific heating capacity, kJ/kg, K (≈1.0 at 20 °C) Δt = required temperature increase/reduction, °C

Throw length for alternative end velocity, L xm
$L_x = l_{0.2} \cdot 0.2/V_x$ $l_{0.2}$ = throw length for end velocity 0.2 m/s according to catalogue data, m V_x = selected alternative end velocity, m/s

Worth reading - bibliography

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