Rooms		
Symbols		Units
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		
Symbols		Units
C_p	Specific heat capacity	kJ/kg · K
C	Coefficient of radiation	$W/m^2 \cdot K^4$
d_h	Hydraulic diameter	m
d_e	Equivalent diameter	m
E	Energy	J
F	Force	N
g	Acceleration due to gravity	m/s^2
h	Enthalpy	J/kg
m	Mass	kg
P	Power	W = J/s
Pr	Prandtl's 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 differential	Pa
q	Volume flow	m^3/s
r	Heat of vaporisation	kJ/kg
R	Heat 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	Heat transfer coefficient	$W/m^2 \cdot K$
v	Velocity	m/s
α	Heat transition coefficient	W/m· K
ε	Emission factor, efficiency	
ν	Kinetic viscosity	m^2/s
ρ	Density	kg/m^3
φ	Relative humidity	%
η	Efficiency	
λ	Heat conduction factor	W/m · °C

Acoustics		
Symbols		Units
A	Absorption area	m^2 Sabine
c	Speed of sound	m/s
D	Level difference	dB
f	Frequency	Hz
i	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	Pressure level	dB ref $2 \cdot 10^{-5}$ Pa
L_W	Sound pressure 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	Directionality factor	
R	Room constant	
R	Reduction factor	dB
T	Resonance time	s
α	Absorption factor	
λ	Wavelength	m

Air exchange		
Symbols		Units
ε_{rc}	Ventilation efficiency	%
ε_{pc}	Local ventilation efficiency	-
ε_{ra}	Air exchange efficiency	%
ε_{rt}	Temperature efficiency	%
ε_{pt}	Local temperature index	-
τ_n	Nominal time constant	h
τ_m	Average age of room air	h
τ_r	Exchange time for room air	h
n	Specific air flow	Room volumes/h

Conversion factors

Length		
m	in (inch)	ft (feet)
l	39.370	3.281
$25.4 \cdot 10^{-3}$	1	$83.33 \cdot 10^{-3}$
0.3048	12	1

Area	
m^2	sq ft (square feet)
l	10.76
0.09290	1

Volume		
m^3	ft^3	US gallon
l	35.32	264.2
$28.32 \cdot 10^{-3}$	1	7.481
$3.785 \cdot 10^{-3}$	0.1337	1

Mass	
Kg	lb (pound)
l	2.205
0.4536	1

Quantity (Air flow rate)			
m^3/s	l/s	m^3/h	cfm (cubic feet per minute)
l	10^3	3600	2119
10^{-3}	1	3,6	2.119
$0.2778 \cdot 10^{-3}$	0.2778	1	0.5886
$0.4720 \cdot 1\ 0^{-3}$	0.472	1.699	1

Velocity	
m/s	fpm (feet per minute)
l	196.9
$5 \cdot 08010^{-3}$	1

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	$0.1450 \cdot 10^{-3}$	$4.015 \cdot 10^{-3}$
$98.07 \cdot 10^3$	1	10^4	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	$0.2388 \cdot 10^{-3}$	$0.2778 \cdot 10^{-6}$	$0.9478 \cdot 10^{-3}$
98.07	1	$2.342 \cdot 10^{-3}$	$2.724 \cdot 10^{-6}$	$9.295 \cdot 10^{-3}$
$4.187 \cdot 10^3$	426.9	1	$1.163 \cdot 10^{-3}$	3.968
$3.6 \cdot 10^6$	$0.3671 \cdot 10^6$	859.8	1	$3.412 \cdot 10^3$
$1.055 \cdot 10^3$	107.6	0.2520	$0.2931 \cdot 10^{-3}$	1

Effect					
W (= J/s)	kW (= kJ/s)	kcal/h	hk	Btu/h	TR (ton of refrigation)
1	10^{-3}	0.8598	$1.36 \cdot 10^{-3}$	3.412	$0.2843 \cdot 10^{-3}$
10^3	1	$0.8598 \cdot 10^3$	1.360	$3.412 \cdot 10^3$	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	$2.510 \cdot 10^3$	0.2091
0.2931	$0.2931 \cdot 10^{-3}$	0.2520	$0.3985 \cdot 10^{-3}$	1	$83.33 \cdot 10^{-6}$
$3.517 \cdot 10^3$	3.517	$3.024 \cdot 10^3$	4.783	$12 \cdot 10^3$	1

Different concepts of efficiency

During the last few years we have begun to study the efficiency of ventilation systems and their role in removing air contaminants. In this connection, we talk about two different concepts:

- VENTILATION EFFICIENCY, which is a measure of how efficiently contaminants are removed.
- AIR EXCHANGE EFFICIENCY, which is a measure of how efficiently the air in a room is exchanged.

One of the main objectives for a designer therefore is to design and position supply and exhaust air terminals so that air exchange and ventilation efficiency are as high as possible.

The ventilation efficiency depends on several different parameters:

- positioning of supply and exhaust air terminals
- type of terminal
- supply air velocity
- temperature difference between supply and exhaust air
- the occurrence of disturbances, heat sources, activity etc.

According to a proposal by the Nordic Ventilation Group, the expression "specific air flow rate", (n), should be used instead of the expression "air change rate".

The specific air flow rate was called ”air change rate” with the unit airchanges/h. This however often lead to the misunderstanding that the air in a room is exchanged by the number of times per hour that the number indicates. The problem here is that the speed at which the air in a room is exchanged depends not only on the supply air flow and the volume of the room, but also to a large extent on the nature of the air currents in the room.

The surplus heat can be seen as a contaminant. The introduction of the concept of ”temperature efficiency” is therefore relevant.

Since surplus heat is considered to be a contaminant, we can replace ”concentration” with ”temperature” to obtain the temperature efficiency.

We differentiate between ”average temperature efficiency”, which applies to the whole room as an average value, and ”local temperature index” which applies to specific points in the room.

Basic concepts

Ventilation efficiency, ε_{rc}

Defined at a certain level of contamination, as the ratio between the concentration in the extract air and the average concentration in the room, i.e.

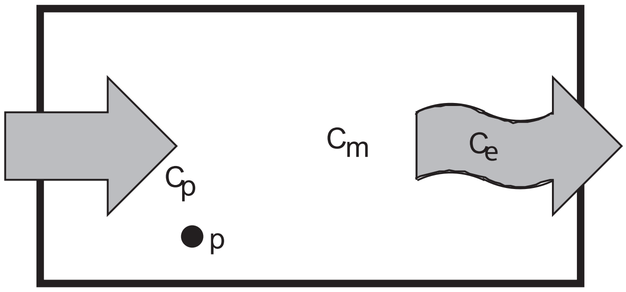
$$\varepsilon_{rc} = \frac{C_e}{C_m}\ 100\%$$

where C_e = equilibrium concentration in the exhaust air
where C_m = average concentration in the room at equilibrium

Local ventilation index, ε_{pc}

where C_p = equilibrium concentration at point p

$$\varepsilon_{pc} = \frac{C_e}{C_p}\ 100\%$$



Air exchange efficiency, ε_{ra}

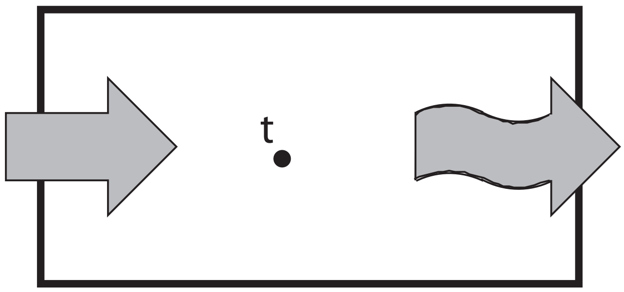
Defined as the ratio between the nominal time constant and the exchange time for the air in the room.

$$\varepsilon_{ra} = \frac{\tau_n}{2 \cdot \tau_m}\ 100\%$$

where τ_n = nominal time constant
τ_m = average age of the air in the room
2· = the exchange time for the air in the room

Note:
The average age of the air is directly related to the time it takes to exchange the air in the room.
To exchange all the air in the room takes an average time which is equal to twice the average age of the air in the room.

The average age of the air can be determined by measurements made at the exhaust air duct.



Specific air flow rate, n

$$n = \frac{q}{V} \quad \frac{m^3/h}{m^3} \quad \text{or} \quad \frac{\text{number of room volumes}}{h}$$

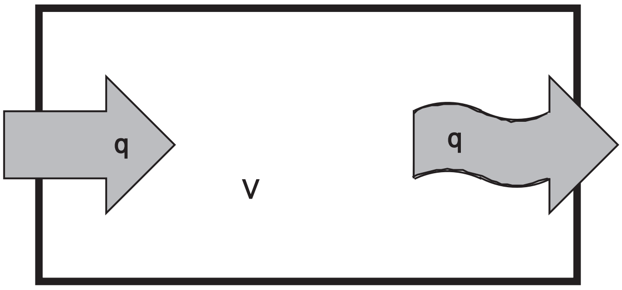
where q = the supply air flow rate (m/h)
V = room volume (m)

Nominal time constant, τ_n

The nominal time constant (t_n) is the time during which the supply, q, on average remains in the room.

$$\tau_n = \frac{V}{q}$$

where q = the supply air flow (m/h)
V = room volume (m)



Temperature efficiency, ε_{rt}

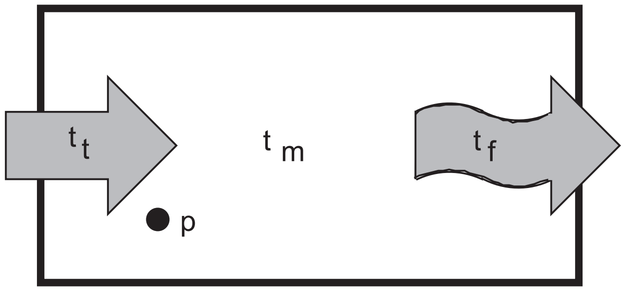
$$\varepsilon_{rt} = \frac{t_f - t_t}{t_m - t_t} \; 100\%$$

where τ_f = exhaust air temperature
τ_m = average room temperature (at equilibrium)
τ_t = supply air temperature

Local temperature index, ε_{pt}

$$\varepsilon_{pt} = \frac{t_f - t_t}{t_p - t_t} \; 100\%$$

where t_p = the temperature at point p at equilibrium.



Occupied zone

The occupied zone is that part of the room, which is normally occupied by people, and should be defined together with both the builder and the architect. Its volume is determined by planes which are parallel with the walls, the ceiling and the floor of the room. The distance between the planes of the occupied zone and the room planes vary depending upon the purpose of the room.

The floor is the lowest horizontal plane of the occupied zone.

The following table provides an overview of the normal distances between the room plane and the occupied zone plane, as well as the usual distance ranges for each set of planes.

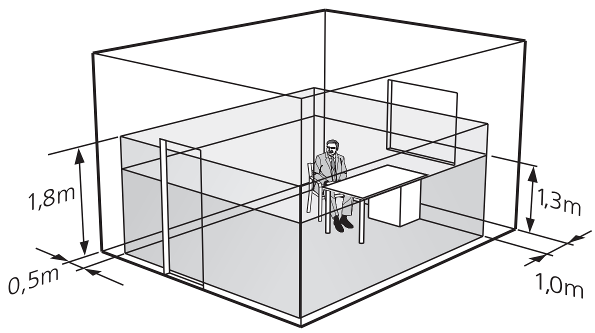


Figure 1. The marked surfaces define the default occupied zone.

Room plane	Typical distance range from the room plane to the occupied zone	Default value for distance between the room plane and the occupied zone according to European standard, EN 13779
External walls	0.15 - 0.75 m	0.5 m
External windows and doors	0.5 - 1.5	1.0
Inner wall	0.15 - 0.75 m	0.5 m
Floor, lower limit	0 - 0.2 m	0.05 m
Floor, upper limit - standing person	1.3 - 2.0 m	1.8 m

Table 1. Limits of the occupied zone

Affected zone

Affected zone is a concept used for low-velocity devices and is therefore of primary interest for displacement ventilation.

According to testing regulations currently in force (EN 122 39) Nordtest standard for Heating, Ventilation and Sanitation), the affected zone is defined by the dimensions a_v and b_v, according to Figure 2.

In the figure, dimension a_v represents the greatest horizontal distance from the wall (or the centre of the device in the case of a cylindrical device) to the isovel for v m/s.

Dimension b_v is the greatest horizontal distance at a right angle to a_v between the end points of the isovel.

In the testing method it is pointed out that the isovel should be measured where the velocity is highest, i.e., not at a specific distance from the floor. In the testing method currently being proposed as an European norm (EN 12239 Aerodynamic testing and rating for displacement flow applications), the velocity v m/s for the isovel has been defined as:

- 0.2 m/s for low-velocity devices intended for comfort ventilation
- 0.3 m/s for low-velocity devices intended for industrial ventilation

For ceiling-mounted, low-velocity devices, the affected zone is defined according to figure 3.

It is essential that the ventilation designer notes how different manufacturers define affected zones. Various methods are used, such as:

- comfort zone, with its own special definition
- isovel on a level of 0.05 m above the floor
- isovel on a level of 0.10 m above the floor

Relatively large differences in the a_v and b_v values can result if measurement and reporting methods deviate from those specified in the testing standard.
DEVIATION FROM THE ACCEPTED METHOD ALWAYS RESULTS IN SHORTER AFFECTED ZONES!

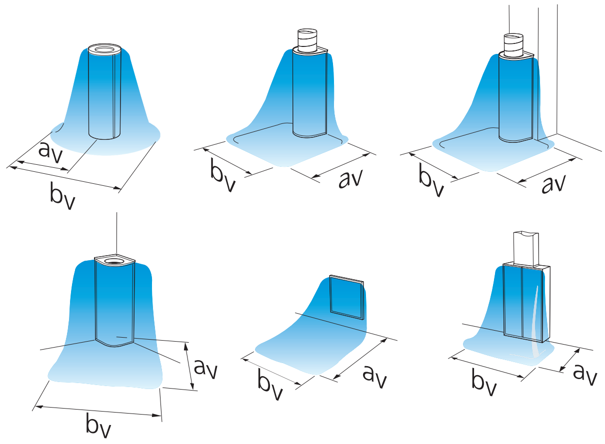


Figure 2. Floor-mounted and wall-mounted, low-velocity devices.

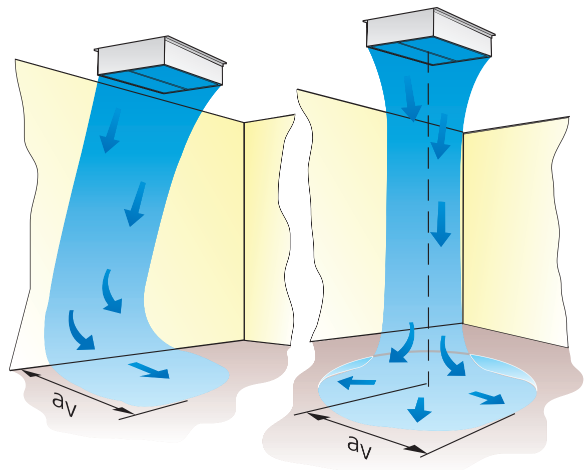


Figure 3. Ceiling-mounted, low-velocity devices.

Far zone

The far zone is a concept which is used in connection with displacement ventilation. The far zone is defined as the zone outside the affected zone where density current takes place. The characteristics of density current are:

- it is powered by the difference in density between the supplied air and the room air.
- it causes a small induction of surrounding air
- it is very thin, normally around 10 cm
- it has somewhat lower velocity fluctuations (turbulence) than a jetstream.

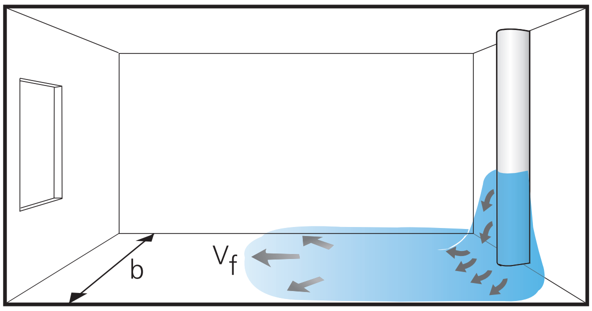


Figure 4. Far zone in the room with displacement unit.

The air velocities within a far zone are determined by:

- the heat load in the room
- the geometry of the room (width)

When an equalisation of the airflow over the width of the room has been achieved, the air velocity in the far zone can be calculated according to 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 air flow (m/s)
	Δt =	difference between supply air temperature and room air temperature (K)
	b =	width of the room (m)
	T =	absolute temperature of the room (K)
	v_f =	The air velocity within a far zone (m/s)

Example:

Office of width 3,6 m

Semicircular section diffuser placed in rear edge

Supply air temperaturer	18°C
Room temperature	24°C
Air flow	30 l/s

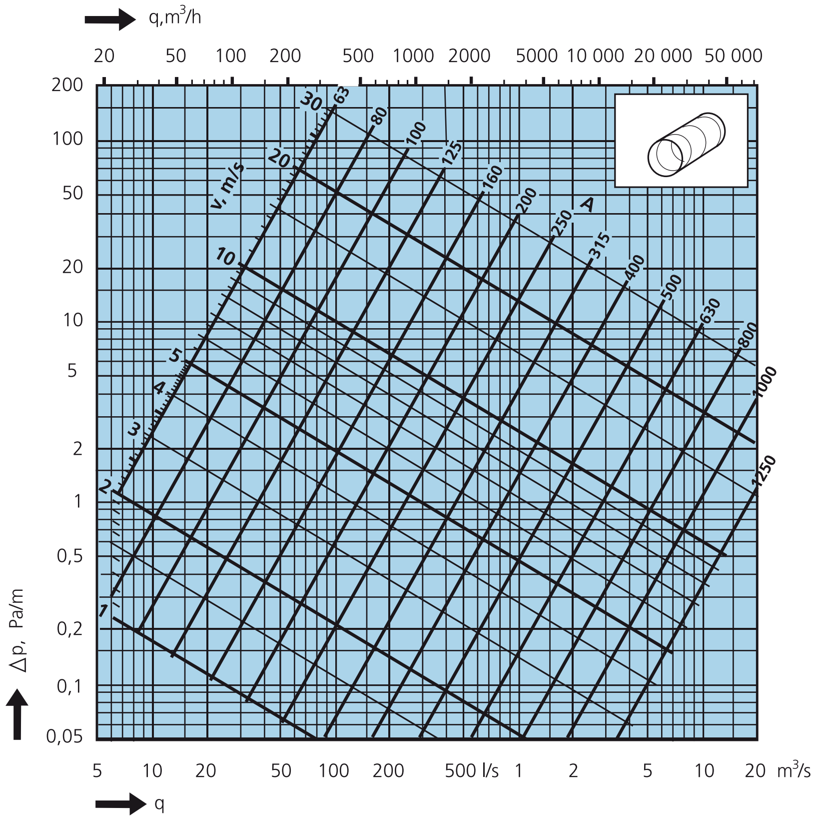
$$v_f = \left(\frac{9{,}81 \cdot 0{,}030 \cdot 6}{3{,}6 \cdot 297}\right)^{1/3}$$

$$v_f = 0{,}12 \text{ m/s}$$

This air velocity should be considered as the lowest value which can be obtained under density current conditions.

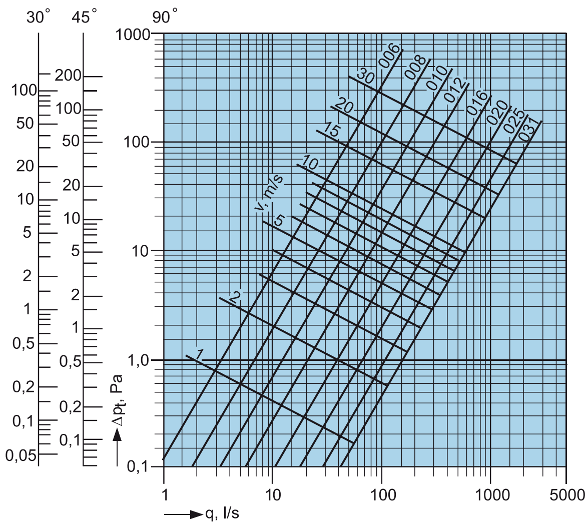
Graphs and formulae

Pressure drop graph for circular section ducts

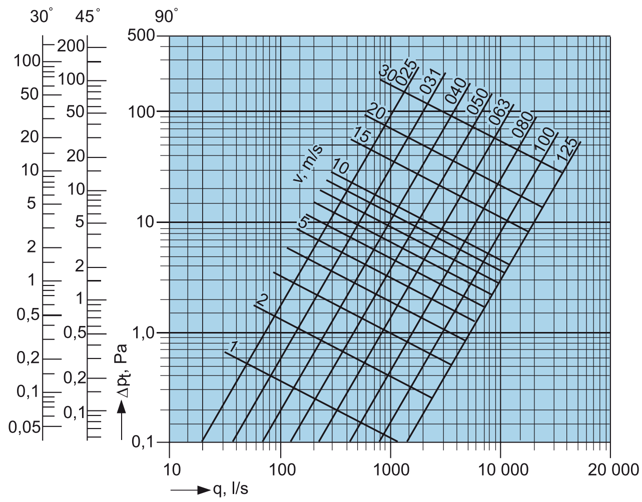


A = Size, mm

Pressure drop graph for circular section bends

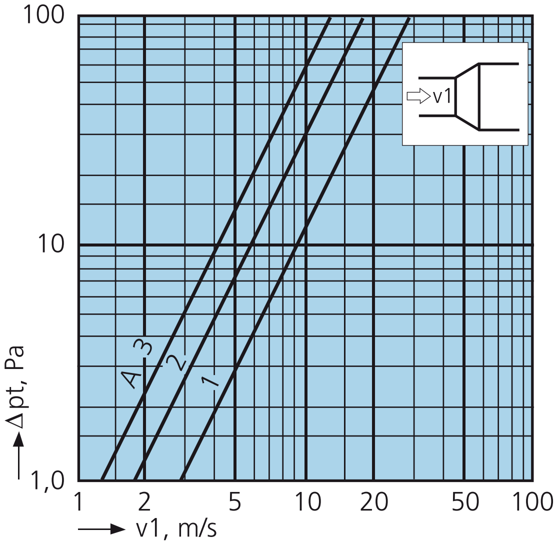


Pressure drop graph for circular section bends



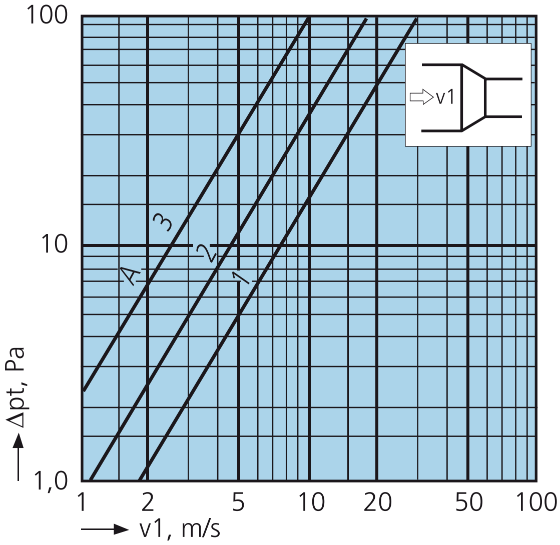
Pressure drop graph, dimensions changes in circular ducts

Area increase



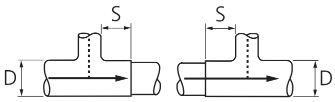
A = Number of dimension changes.
(Example from 016 - 020 = 1 dim.changes.)

Area reduction

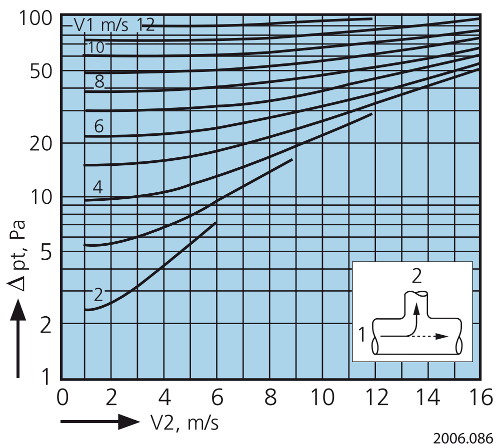


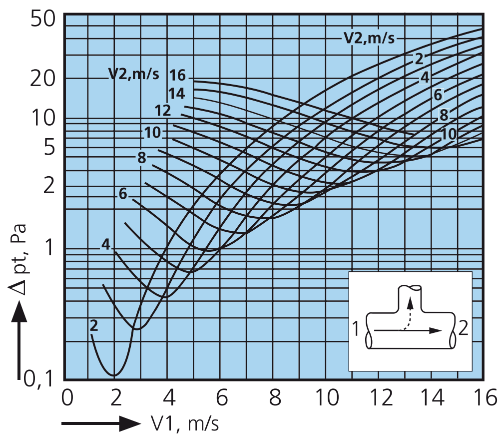
A = Number of dimension changes.
(Example from 020 - 016 = 1 dim.changes.)

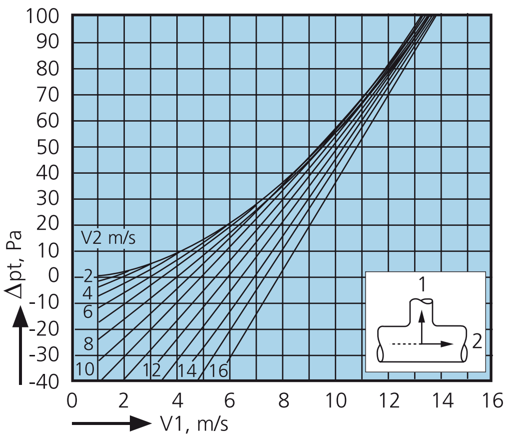
Pressure drop graphs for circular T and Y joints

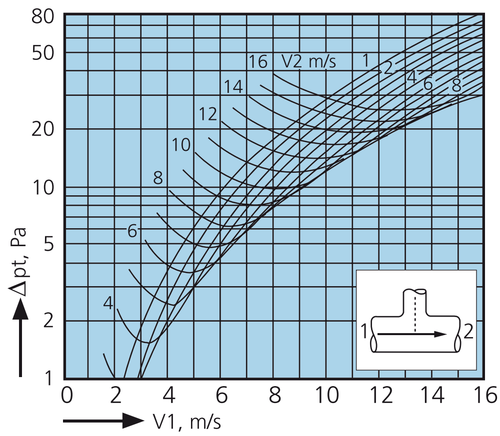


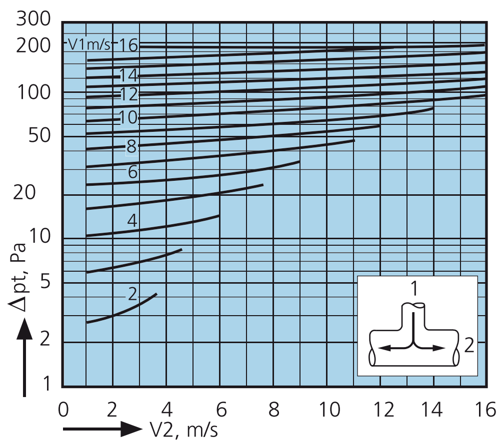
Pressure drop includes dimensionen reduction where S < 3 x D, as shown in the diagram.

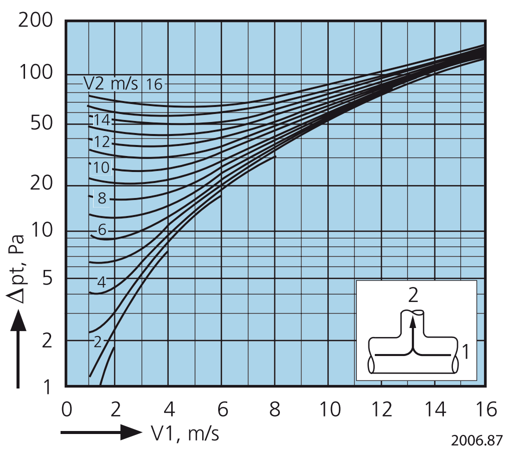




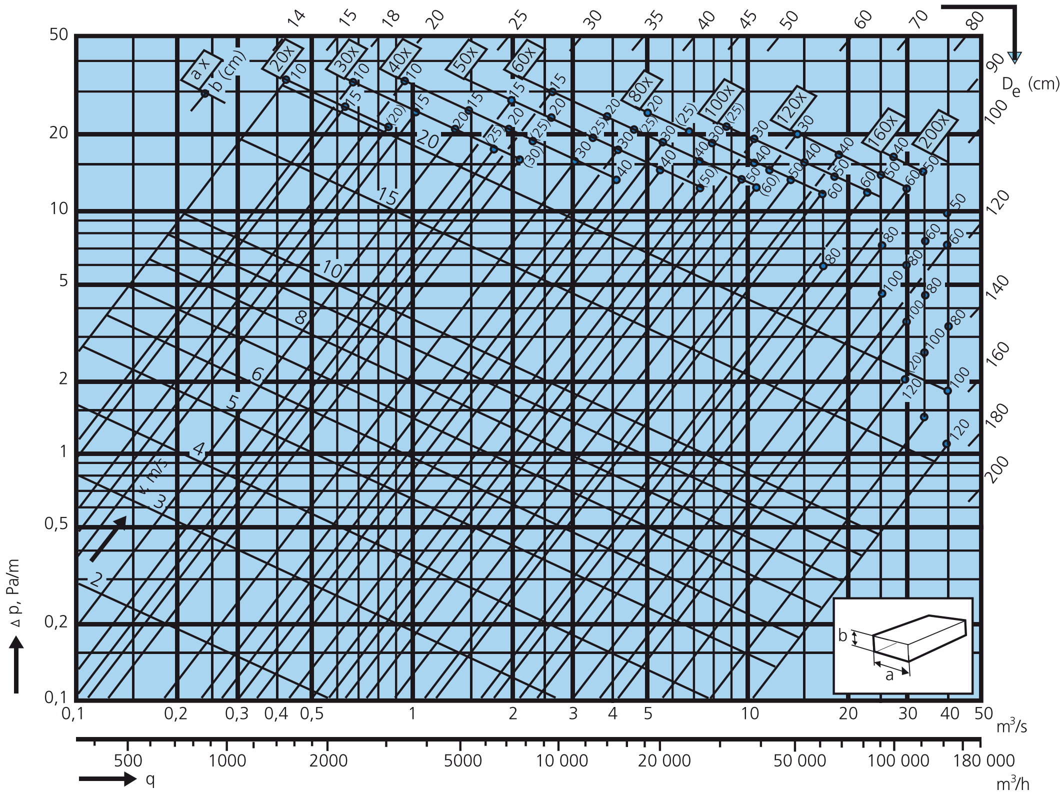






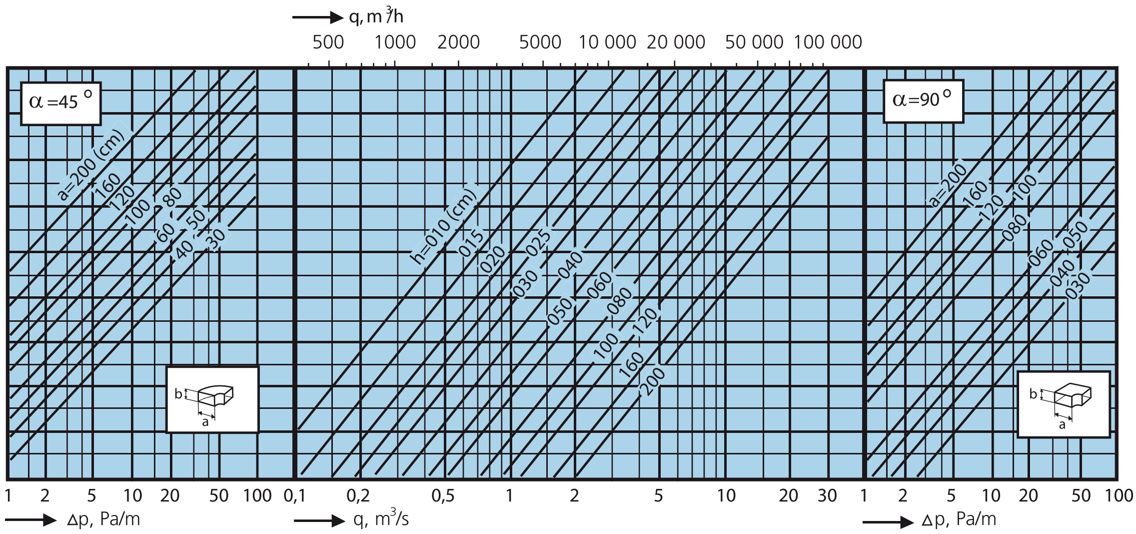


Pressure drop graph for rectangular section ducts

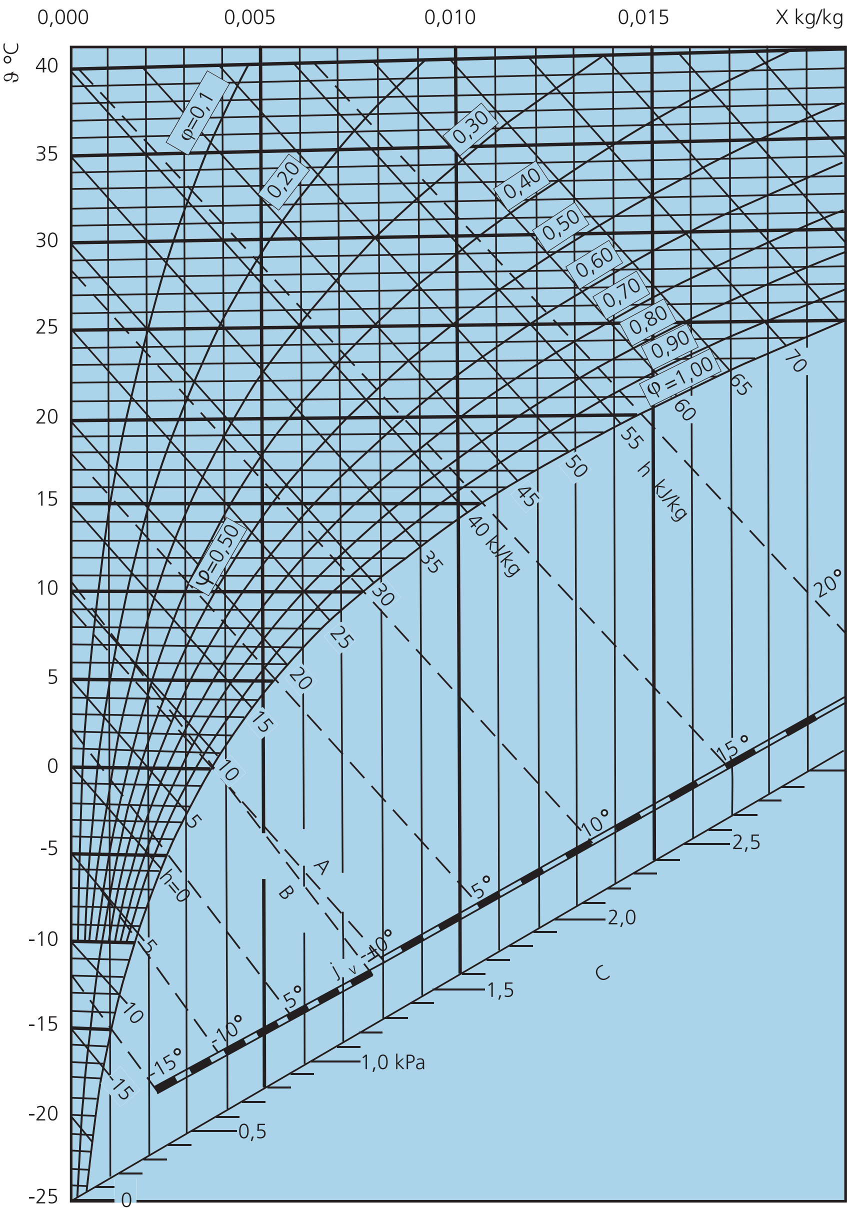


Ducts with the same equivalent diameter D_e may have different cross section areas. The velocity curves plotted above are therefore approximate with a maximum error of <5%.

Pressure drop graph for rectangular section bends



Mollier graph



A = Wet thermometer
B = Iced-over thermometer
C = Water vapour saturation pressure, kPa

Symbols:

h	= enthalpy for 1 kg dry air, kJ/kg
x	= water content of 1 kg dry air, kg/kg
φ	= relative humidity
ϑ	= dry thermometer temperature, °C
v	= wet thermometer temperature, °C

The graph is correct at atmospheric pressure = 101.3 kPa = 1013 mbar.

Formulae overview

Air flow, q m³/s
$q = A \cdot v$
A = cross section area, m^2
v = air velocity, m/s

Dynamic pressure, p_d Pa
$P = \rho\, v^2/2$
ρ = air density kg/m^3
v = air velocity, m/s

Hydraulic diameter, d_h m
$d = 4 \cdot A/O$
A = cross section area, m^2
O = circumference of duct, m
d_h for rectangular ducts
$d_h = \frac{2\,ab}{a + b}$
a and b are the duct sides
d_h for circular ducts
d_h = d = duct diameter

Total pressure drop - supply air, p_t Pa
$p_t = p_s + p_d$
p_s = static pressure drop, Pa
p_d = dynamic pressure drop, Pa

Total pressure drop - exhaust air p_t Pa
$p_t = (-p_s) + p_d$
p_s = negative static pressure drop, Pa
p_d = dynamic pressure drop, Pa

Cross section area, circular duct, A m²
$A = \pi \cdot d^2 / 4$
d = duct diameter, m

Circumference, circular duct, O m
$O = \pi \cdot d$
d = duct diameter, m

Air density, ρ kg/m³
$\rho = 1.293 \cdot (\frac{B}{1013}) \cdot (\frac{273}{273 + t})$
B = barometer reading, mbar
t = air temperature, °C

Cooling/heating effect, P kW
$P = q \cdot \rho \cdot Cp \cdot \Delta T$
q = air flow, m^3/s
ρ = air density, kg/m
Cp = specific heat capacity of air, kJ/kg,K (≈ 1.0 at 20 °C)
Δt = temperature difference, °C, between supply and exhaust air

Throw at different terminal velocities, L_x m
$L_x = l_{0.2} \cdot 0.2/V_x$
$l_{0.2}$ = throw to terminal velocity 0.2 m/s in accordance with catalogue data, m
V_x = selected alternative, m/s

General

Ventilation is used for different purposes. Its principal purpose is to exchange contaminated air with clean and tempered air. It can be said that contaminated air is air which is too warm and which contains gases and dirt particles.

It is also important to create a room climate without draught problems and only slight temperature changes in the occupied zone. Room climate depends on several factors, including air velocity, air temperature and radiant temperature. When people occupy rooms, significant changes in any of these factors can cause draught problems.

In order to maintain these basic functions, the air conditioning system should be:

- sufficiently stable to resist disturbance. External disturbances could be wind and temperature; internal disturbances could be convection currents from heat sources. To provide stability against external disturbances the system must be designed with a certain pressure drop.
- easy to check and measure. The system must be designed for either fixed measurement outlet diffusers or fixed measurement units in main and manifold ducts. Measurement diffusers of these types considerably decrease the time taken for commissioning and thereby reduce the total cost of installation.

Indoor climate

It is important to remember that the main purpose of the ventilation system is to remove pollution as quickly and efficiently as possible from the ventilated room.

A very important associated condition is that the requirements for comfort must be met. It is therefore necessary to have relevant demands on the indoor climate.

The indoor climate consists of the four technical climates:

- **air hygiene**
- **thermal climate**
- **acoustic climate**
- **visual climate**

Creating a system which functions in every respect requires the co-operation of installation personnel, construction personnel and architects. For example, early in the building process the heating, water and sanitation consultant should state the requirements for spaces, the consequences of the design of the building and so forth, so that the demands on the indoor climate can be met at the lowest possible cost.

Air hygiene

The word ventilation comes from the Latin word ventilare, which means to blow. Today, we give the term ventilation a more concrete definition, air exchange. That is, we replace used and polluted air with clean air.

It might therefore be interesting to take a look at how estimations were made in the past to achieve sufficiently large air flows.

- Pettenkofer (1818-1901).
  CO_2 -content can be used as an indicator of air quality.
  - Min. hygiene requirement: ≤1000 ppm CO_2
  - Requirement for good air quality ≤700 ppm CO_2

- Elias Heyman (1829-1889) Karolinska Institute, Sweden:
  Measurements in schools
  - without ventilation: = 5000 ppm CO_2
  - with some ventilation: 1500-3000 ppm CO_2

Summary: Not one classroom had sufficient ventilation.

Measurement in homes:
Summary: We cannot depend on natural ventilation if we want clean air indoors.

- Yaglou et al (1936)
  Studied the relationship between human odours and ventilation flows.

Results:
8 l/s, p or 15 l/s, p, when half the air flow was recirculated. The disadvantage of recirculation is a large accumulation of pollution in the ducts. This requires both a large air flow and regular cleaning of the ventilation system.

Yaglou's studies:
- were the basis of guidelines for flow sizes over long periods
- typical minimum value for an office was: 7.5 l/s,p (ASHRAE)

The oil crisis of the 1970's led to reduced air flows in ventilation systems.
- ASHRAE, in an area where smoking is not allowed ≥2.5 l/s,p
- NKB ≥4 l/s,p

Today's generally increased attention to better air hygiene has prompted the increase in air flows:

- ASHRAE (1989) ≥10 l/s,p in an office
- NKB (1991) ≥11 l/s,p
- Fanger (1988) ≈50 l/s, p in buildings with high pollution loads and »14 l/s, p in buildings with low pollution loads.

Since the major problems related to indoor climate began in earnest as a result of the energy savings which resulted from the oil crisis in the 1970's, a simple overview indicates the risks associated with insufficient air flows.

When a new Swedish building standard was introduced in the mid-1970's (SBN 75), air flows were reduced in parallel with requirements for increased "tightness" of buildings.

During the last two decades, there have been many newspaper articles about "sick buildings" which smell, where mould abounds and which give people headaches because of the bad indoor climate. The problem seem to be found in all kinds of buildings. Behind the headlines, the same kind of symptoms repeatedly occur in the people who are affected. The symptoms are found mainly in the mucous membrane:
- irritated eyes, nose and throat
- constant sniffles and stuffy nose
- dryness in the mucous membrane
- repeated throat infections
- cough
- hoarseness

Skin symptoms such as:
- dry and irritated skin
- skin reaction on the face and body

Even general symptoms such as:
- headaches
- tiredness
- heavy-headed feeling
- difficulty in concentrating
- indisposition
- dizziness

The causes of these "sick buildings" are many and complex. The following are some of the major reasons:
- efforts to minimise production costs
- insufficient documentation
- purchasing procedures
- built-in humidity/moisture in the structure
- insufficient operation and maintenance procedures
- energy-saving
- new building materials
- poor cleaning

Lesson

We can learn from the past that the requirements for energy economizing must never be so stringent that proper ventilation is put at risk.

In the R1 guidelines series, the Swedish Indoor Climate Institute supplies the following values for the acceptable impurity content of indoor air in different air quality classes:

Pos	Substance	Highest content in mg/m^3 class		Note
		AQ1	AQ2	
1	Carbon monoxide, total			
	MV 0,5 h	60	60	See note 1a
	MV 8 h	6	6	See note 1b
	- from tobacco smoke MV 1 h	2	5	See note 3
2	Carbon dioxide MV 1 h	1000	1800	See note 2
	(i ppm*)	600	1000	
3	Ozone MV 1 h	0.05	0.07	See note 3
4	Oxides of nitrogen MV 1 h	0.11	0.11	See note 1b
	MV 24 h	0.08	0.08	
5	Volatile organic carbons (VOC)			
	- total MV 0.5 h	0.2	0.5	See note 4
	- formaldehyde MV 0.5 h	0.05	0.1	See note 5
6	Particles from tobacco smoke,			
	inhalation only MV 1 h	0.1	0.15	See note 3
7	Dust**	0.06	0.15	See note 6
8	Mould*** cfu/m^3	50	150	See note 7
9	Bacteria cfu/m^3	4500	4500	

Table 2. Indoor air quality.

The classification of the air quality classes AQ1 and AQ2 is based on statistical studies of the proportion of people who are expected to experience measurable inconvenience or be dissatisfied with the indoor climate.

MV: mean value over the time period as shown:

* ppm is converted from microgram/m(g/m) by the following formula: ppm = 24,1 x mg/m/ molweight

Molweight: Carbon dioxide 44, carbon monoxide 28, sulphur dioxide 64, ozone 36, nitrous dioxide 44, nitric oxide 30, formaldehyde 30.

** Dust in mg/mcan be converted into an approximate number of particles using the formula = number mg x 5000. (Applies to a particle size of about 10 µm, i.e relatively large particles.)

*** 1cfu = 1 colony forming unit. Pathogenic mould must be 0.

Note 1a: Values according to WHO-AQG.

Note 1b: Values according to the Swedish Environmental Protection Agency proposal 8 Augst 1989.

Note 2: Values for AQ1 according to ASHRAE 62-1989 and for AQ2 according to Morey et al (IAQ 1986).

Note 3: Values according to WHO-Euro 103, 1986.

Note 4: Values, partly according to Mölhave, partly from a summary Healthy Buildings 1988 (HB-88).

Note 5: Values according to WHO-IAQ and Berglund et al 1985.

Note 6: Values according to O Seppanen 1989.

Note 7: Values according to Holmberg (Sunda huset 1987) and Canadian Ministry of Health 1987.

Figure 5. The quality of air is affected by many factors, such as emissions from building materials, human secretion etc. Smoking is an example of a contaminant that negatively affects the quality of the air.

Thermal indoor-climate

An acceptable room temperature is one which is not less than 18°C or more than 28°C.

The comfortable temperature range is considerably narrower and normally falls between 20°C and 24°C (see figure 6). The figure illustrates that it is difficult to meet all requirements for a suitable temperature. Therefore, we must always consider using our clothes as a comfort regulators.

The effect of the room temperature on performance capabilities is shown in figure 8. The figure shows schematic and very simplified results from tests to determine mental and physical performance. It is quite clear how quickly mental performance and rate of work diminish as the room temperature rises. It is therefore easy to show that it is profitable to invest in suitable air-conditioning equipment.

Example:
Summer clothing, office work at a desk.
Room temperature 25°C.
IIn comparison with working at the comfort temperature, the work rate has dropped to 70 % and mental performance to 90%. In other words, the employer will only get 70% output from his employees at this higher temperature.

Assume an hourly cost of 20 Euro.
Assume that the room temperature exceeds 25°C for approx. 100 working hours per year.

The output loss per year will be
0.3 x 20 x 100 = 600 Euro/employee.
The cost of a suitable air-conditioning installation is, at the most, approx. 30 Euro/m^2.

At 20 m^2/person the increase in costs is 600 Euro/person, i.e. the investment pays for itself after the first summer season.

Figure 6. How we perceive the ambient climate is greatly influenced by how we are dressed.

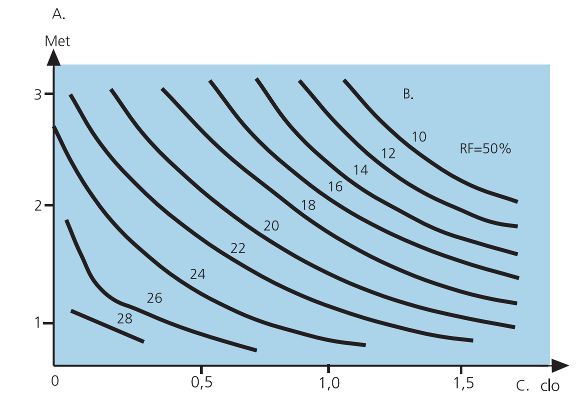


Figure 7. Relationship between working temperatures, met and clo values.
A = Activity level
B = Working temperature (optimal) °C
C = Clothes clo

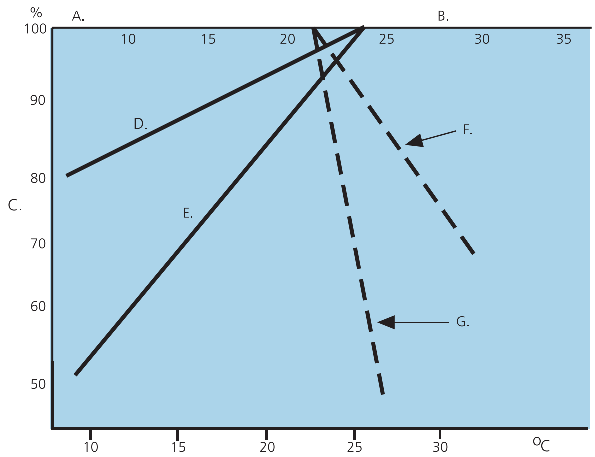


Figure 8. Work performance as a function of indoortemperature (according to Wyon).
A = Work actively
B = Working sitting still (1,0 clo)
C = Working performance
D = Manual work
E = Finger speed, dexterity
F = Mental performance (during experiment)
G = Work rate (during work in progress)
°C = when sitting still and working (60 W/m^2) and wearing summer clothing (0,6 clo)

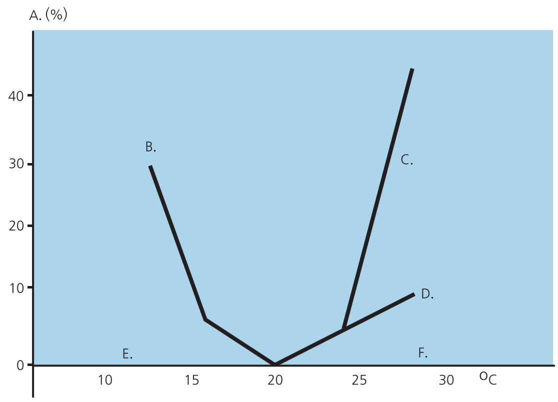
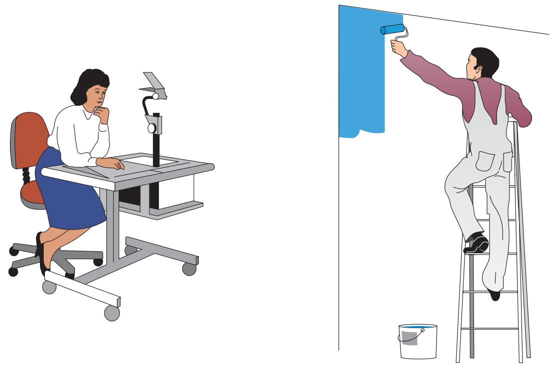


Figure 9. This graph shows how the frequency of accidents during factory work increases with change in indoor temperature (according to Wyon).
A = Increase in number of accidents, %
B = Accidents
C = Men
D = Woman
E = Working actively (0,6 clo)
F = Working sitting still (1,0 clo)

Additional reasons for having a good indoor climate are shown in figure 9, below. This is a schematic and highly simplified illustration of the relationship between occupational accidents and deviations from the comfortable temperature range.



Figur 10. How indoor climate is perceived is greatly influenced by the type of work being carried out. The more active you are the lower the temperature that is desirable.

Requirements for the indoor climate

Examples of demands that should be made on the indoor climate can be found in the Indoor Climate Institute's report R1, Indoor Climate Systems in Various Categories.

One problem to bear in mind is that each person places an individual set of demands on the working environment. This means that it is essential to prioritise system solutions that are as flexible as possible, i.e. systems that make it possible to meet individual needs.

Table 3 illustrates the demands on the thermal climate which are most commonly used (Indoor Climate Institute Guideline series R1). The classification of the indoor climate is based on human perception of various conditions. As a measurement of these perceptions of thermal climate, the so called PPD-index is used, which indicates the expected percentage of dissatisfied persons in a larger group after the group has been exposed to a certain number of persons in a larger group who exhibit an effect from the factor in question.

PPD-values which were used as a basis for the specification of the thermal quality classes were:

TQ1 < 10%
TQ2 10%
TQ3 20%

TQ2 corresponds to the requirements described in ISO 7730.
TQ3 corresponds to ASHRAE 61-1989.
TQ1 values are considered to be attainable only through individual adjustment of temperature and air flow.

The classes TQ1 and TQ2 are applicable at normal office work.

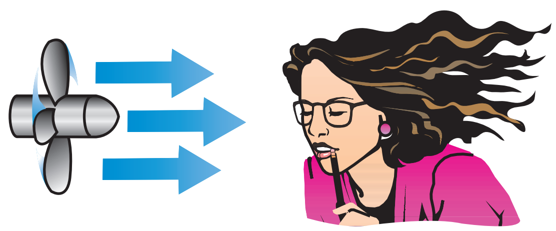


Figure 11. Air speeds affect how indoor climates are perceived.

	Factor value in quality class		
Indoor climate	TQ1	TQ2	TQ3
Operating temperature (t)			
Winter 1)			
- highest value °C	23	24	26
- optimal value °C	22	22	22
- lowest value °C	21	20	18
Summer 2)			
- highest value °C	25.5	26	27
- optimal value °C	24.5	24.5	24.5
- lowest value °C	23.5	23	22
Air velocity in the occupied zone 3)			
- winter m/s	0.15	0.15	0.15
- summer m/s	0.20	0.25	0.40
Vertical temperature diff. 4)			
- summer-winter °C	2.5	3.0	3.0
Radiant temp. asymmetry			
- towards warm ceiling K	4	5	7
- towards cold wall (window) K	8	10	12
Temperature change rate °C/h	-	-	-
Air humidity	-	-	-
Floor temperature			
- highest value °C	26	26	(32)
- lowest value °C	22	19	16
according BFS 1988:18			
- highest value °C	27	27	27
- optimal value °C	24	24	24
- lowest value °C	16	16	16
Temperature adjustment °C	± 2	(± 1)	-

Table 3. Thermal quality. Acceptable values for various factors in different quality classes.

1) Applies to a clothing factor of 1,0 clo.

2) Applies to a clothing factor of 0,5 clo.

3) Air velocity is given as a mean value over a 3 min. period.

4) Between levels 1.1 m and 0.1 m above the floor.

General

Many of us have at some time complained about the ventilation at work, in lecture halls or some other large room. It is either too warm or too cold, it is too draughty, the air does not feel fresh and so on. It is difficult to install the right ventilation system for some rooms from the start since there are no rules for all the possible variables involved in the installation.

A full-scale trial in the laboratory, where an office for example is built with the intended ventilation system, is the safest method of investigating how a planned system will function in practice. Swegon's laboratory for full-scale tests provides planners and construction engineers the opportunity of testing a designed ventilation system in the early stages of the building process.

There are two main principles when ventilating a room:

- **Mixing ventilation**
- **Thermally controlled ventilation**

In addition to these, two more types can be mentioned:

- **Piston flow**
- **Short circuiting flow**

Basic facts

Exchange time and exchange air efficiency		
Exchange time and air exchange efficiency for room air with different air flow conditions.		
Air flow pattern	Exchange time for room air	Air exchange efficency
	$2\ \tau_n$	ε_{ra}
Displacement and equalizing	$> \tau_n$ $< 2\ \tau_n$	< 100% > 50%
Mixing	$2\ \tau_n$	50%
Piston flow	τ_n	100%
Short circuiting flow	$> 2\ \tau_n$	<50%

Mixing ventilation

The mixing principle is characterised by a relatively high velocity of the supply air, which causes a high rate of induction of the room air. This creates movements of air which cause mixing to take place and thereby a uniform distribution of those contaminants which are produced in or enter the room. The temperature distribution in rooms ventilated by the mixing principle is also relatively even.

Thermally controlled ventilation

This ventilation type is characterised by low velocity supply air, which is at under-temperature. The thermal forces therefore control the flow to a larger extent than the dynamic forces. The movement of air in the room is thus determined largely by the density differential between the supply air and the room air, as well as by the positioning of the supply and exhaust units. Thermally controlled ventilation can be divided into two categories:

a) displacement ventilation
b) equalising ventilation

Displacement ventilation

This type of ventilation has the following characteristics:

1. the supply air is at floor level
2. no significant mixing of the room air with the supply air is aimed at
3. the supply air is at low velocity
4. the supply air is under tempered
5. exhaust air is removed at ceiling level

Equalising ventilation:

This type of ventilation has been named due to the efforts made to equalise the temperature distribution, primarily in the area of occupation. Its characteristics are as follows:

1. the aim is to mix room air and supply air
2. supply air is at low velocity
3. the supply air is at under-temperature
4. exhaust air is removed at ceiling height

The mixing of room air in the supply air is achieved among other things by:

1. placing the supply air high up in the room
2. creating induction of room air either in or near to the supply diffuser

Piston flow

This type of ventilation is only used in rooms where extremely high demands are placed on the quality of the air. The principle involved means that supply air is distributed evenly over the whole area of, for example, the ceiling in such a way that the direction of movement of the air is uniform and in one direction only - in this case, downwards.

The air can be said to move through the room like a piston. The exhaust air is removed in this case at floor level. In order for this system to function well, a relatively high air velocity is required. To achieve a stable piston flow in a room, the air velocity required is between ≥0.35 and 0.40 m/s.

Piston flow ventilation is never an alternative in the context of comfort ventilation due to the high air velocity involved.

Short-circuiting flow

This is a phenomenon which occurs in those cases when all or part of the supply air is removed with the exhaust air without passing through the occupied area. This type of flow should naturally be avoided as much as possible.

The risk of short-circuit flow is present whenever, for example both the supply units and the exhaust units are located at ceiling level, the supply air velocity is low and if the supply air temperature is higher than the room temperature.

Basic facts

Principle for displacement air flow
Displacement air flow involves providing low velocity air at floor level. The supply air more or less spreads out over the floor surface and is affected by any heat sources in the room. Supply air is drawn up to the ceiling, where it is extracted.

One of the conditions for this principle to function without compromising the comfort requirements, is that there is an even air distribution over the supply air terminal. By adjusting the supply air flow and the number of supply air terminals to the heat emitting sources (machines, people, etc.) a highly effective and comfort-inducing ventilation system can be achieved.

Principle for complete mixing
The supply air is supplied at such a velocity that the contaminants concentration is the same throughout the room. It follows that the temperature difference is negligible throughout the room and is therefore an advantage in terms of comfort. Due to the comfort advantages, this is the most common air movement in use.

Supply air is normally supplied at ceiling level or, alternatively, under a window at a relatively high velocity. To achieve draught-free conditions, a careful choice of supply air terminal is required. The throw, jet spread etc. must be properly balanced in relation to the size of the room.

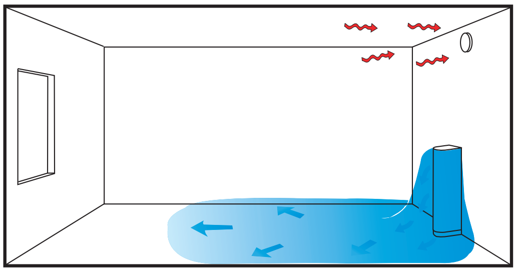


Figure 12. A = Supply air, floor - Exhaust air, ceiling.

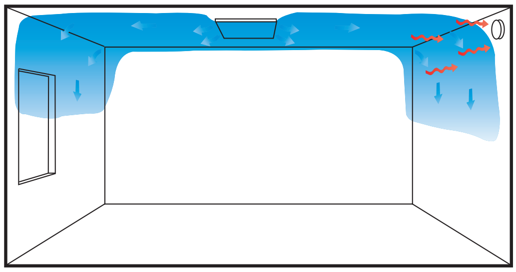


Figure 13. B = Supply air, ceiling - Exhaust air, ceiling.

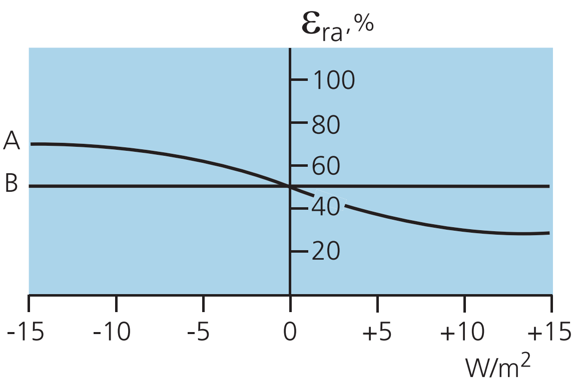


Figure 14. The graph illustrates which air exchange efficiencies ε_{ra} can be obtained as a function of the energy supplied with the air.
A = Supply air at floor level and exhaust air in ceiling level
B = Supply air in ceiling and exhaust air at ceiling level

Piston Flow

One-directional (piston) flow

One-directional flow means that ventilation air is distributed in such a manner that the direction of the air is defined and has one direction only. The air can be said to move like a piston through the room.

If one-directional flow is to function, relatively high air velocities are necessary. Velocities of ≥ 0.35 to 0.40 m/s are necessary to achieve a stable piston flow in the room. Because of the high air velocity, piston flow is never used in comfort ventilation. It is only used in clean rooms where the air quality requirements are extremely high.

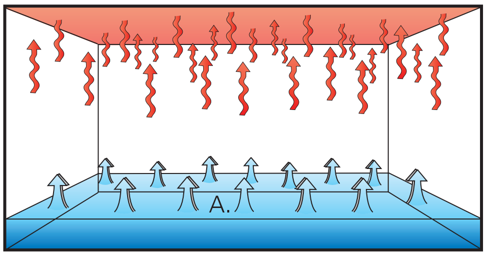


Figure 15. A = Supply air, floor - Exhaust air. ceiling. With pure piston flow, the air exchange efficiency is 100%.

Short-circuiting flow

This type of flow should always be avoided.

Short-circuit flow means that part of the supply air goes directly out with the exhaust air, without entering the occupied zone and without contributing to the room climate, i.e. the air has been short-circuited.

There are several conditions which prevail in this situation: both the supply air and exhaust air devices are placed at ceiling level, the supply air velocity is too low and the supply air temperature is higher than the room temperature.

Short-circuit flow can also occur in situations where the supply air terminals are placed at floor level, (low velocity terminals) and the supply air (which is chilled) is drawn out through open doors and low level exhaust air terminals.

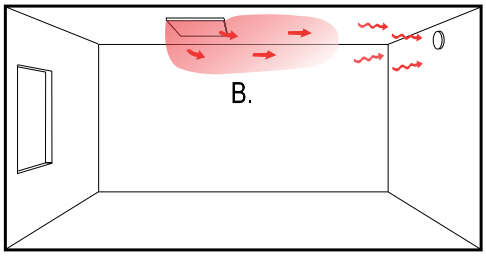


Figure 16. B = Supply air, ceiling - Exhaust air, ceiling. (Heated air

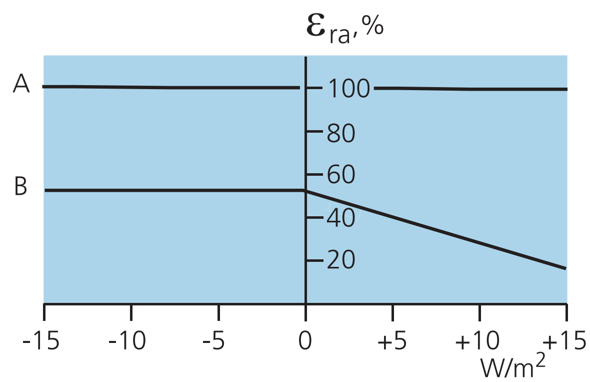


Figure 17. The graph illustrates the air exchange efficiencies era.
A = Supply air at floor level and exhaust air in ceiling level
B = Supply air in ceiling and exhaust air at ceiling level

Practical guidelines

Below are illustrations of the different air systems normally used for comfort ventilation in rooms. The maximum cooling loads given refer to air-transferred cooling power, where consideration has been given to accumulation in the building frame.

The maximum cooling loads for the various alternatives are shown in Table 5.

Define the occupied area carefully. In those cases where downward air flows are acceptable near the walls, this principle is advantageous. Note carefully the convection streams from the window.

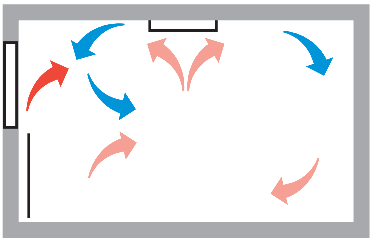


Figure 18. Ceiling.

The importance of throw must be taken into account when selecting supply air terminals. For example in office space where work is carried out near windows the throw $l_{0.2}$ should be equal to the depth of the room multiplied by 0.7, when cooled air is supplied. A longer throw can be of benefit if the occupied zone is not closer than 0.75 m to the window.

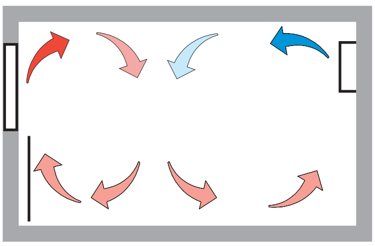


Figure 19. Side wall.

Supplying under-temperature air usually gives a satisfactory result, providing the diffuser has a throw which is somewhat longer than the depth, (1-2 m), of the room.

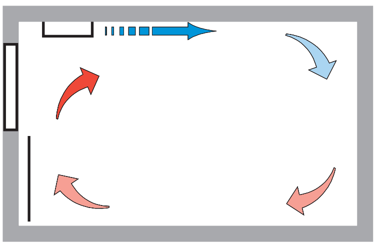


Figure 20. Front, ceiling.

Pay attention to the supply air and window temperature. In cases where there is a risk that both these temperatures are lower than the room air temperature, it is possible that a downward current of air can occur in the area nearest the window (1-2 m).

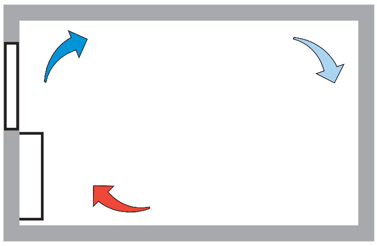


Figure 21. Front window sill.

The positioning requires a high outlet velocity from the supply air unit in order to prevent down-streams in the occupied zone. It is essential that the window sill is designed to prevent air being directed into the room.

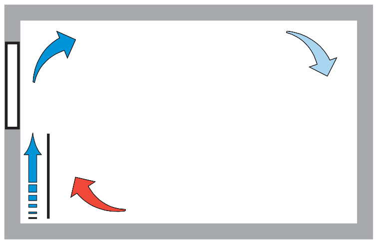


Figure 22. Front, behind radiator.

If a diffuser with an adjustable spread pattern is used, a temperature difference between the room air and supply air of maximum 6°C may be used. The maximum cooling load is approx 35 W/m² at ceiling heights around 2.8 m.

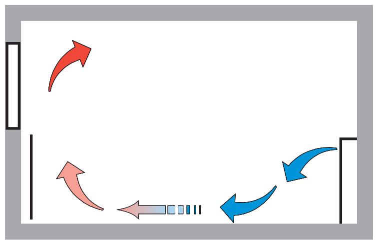


Figure 23. Floor-mounted, low velocity (displacement).

With a special ejector on the low-velocity unit, the temperature distribution is balanced over the room's occupied zone. Temperature differences of between 6°C and 9°C between the room air and the supply air can therefore be used.

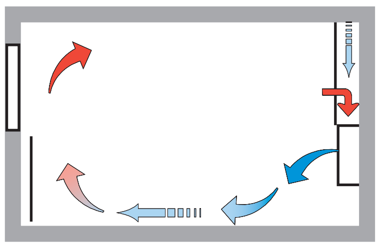


Figure 24. Floor-mounted, low velocity terminal with induction.

Cooling capacity

The approximate limits to the ability of the various ventilation principles to cool a room were given in earlier chapters.

The limits described are variable and highly dependent on conditions such as the size of the occupied zone. Therefore, when determining comfort requirements, it is necessary to relate these requirements to where in the room they are to be meet. Since the requirements can change with time depending on the activities performed in the room, it is essential that the air distribution in the room can be easily changed. This can be done by selecting supply air terminals with a flexible, adjustable spread pattern.

The cooling capacities given for the various ventilation principles apply only in those cases where air carries the energy. If water is used to carry the energy, i.e., use of a cooling ceiling of the radiation type, or if so-called chilled beams for convective cooling are used, considerable heat loads can be transported from the room.

The graph below gives an indication of when it can be relevant to choose either an air-cooling or a water-cooling system.

By beginning with the required air flow (l/s m²) for various types of rooms (see table illustrating air flow requirements), it is easy to determine the most suitable alternative if you know the required cooling effect.

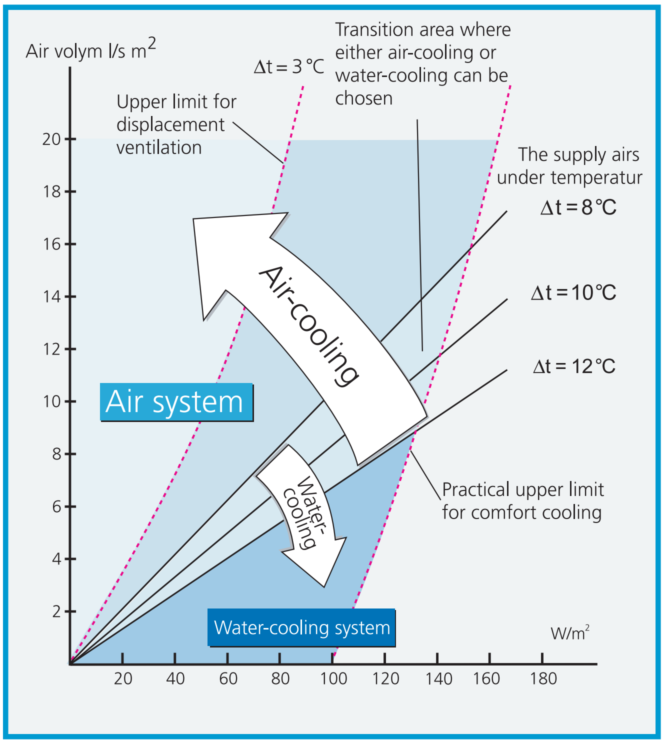


Figure 25. The graph gives an indication of whether cooling can be achieved using an air-cooled system or if this should be supplemented with a water-cooled system, given the previous knowledge of specific cooling requirements and specific air flow requirements.

Type of location	Pers/m	Air flow in l/s, m² at CO_2-content 600 ppm	800 ppm	1000 ppm
Office				
- private	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
- auditorium	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
Day-care	0.4	3.8	2.2	1.5
Discotheque	1.0	27.2	15.1	10.5
Waiting room/Lobby	1.5	30.0	16.7	11.5

Table 4. Air flow requirements.
1) Average weight 70 kg, sitting
2) Average weight 25 kg, walking
3) Average weight 70 kg, walking

Example:

A conference room with a heat load of 60 W/m² has an air flow requirement of at least 4 l/s, m².

a) If the air flow of 4 l/s, m² is selected, a system with water as the energy carrier is required. A solution with convection chilldes beam can then be selected.

b) If the air flow is instead increased to 6 l/s, m², the cooling requirement can be met using an air carrier alternative. This would also produce better air quality.

The correct values for the ability of the various ventilation principles to remove heat loads are given in the table below.

Table 5 applies to cases where there is an occupied zone that extends from the floor to 1.8 m above the floor, 0.2 m from inner walls and 0.5 m from walls with windows. The values apply to a room height of approx. 2.8 m.

Ventilation principle	Max. cooling capacity W/m, floor level.	
	Comfort requirements according to guideline TQ1	Comfort requirements according to guideline TQ2
Airborne systems mixing ventilation:		
- ceiling-, disc. diff.	100[1)]	120
- ceiling-, perf. diff.	60	80
- side wall	40	60
- front ceiling	50	70
- window sill	50	70
Balancing ventilation:		
- ceiling diffuser	40	45
- ejector1.2 m above the floor	35	40
Displacement vent.:		
- floor diffuser	30	35
- wall unit 0.6 m above the floor	35	40
Waterborne system:		
- facade units induction type	50	70
- fan convector on facade	50	70
- cooled ceilings, radiation		
mixing. vent.	100	120
displ. vent.	70[2)]	85
- cooled ceilings, convection		
mixing. vent.	70	85
displ. vent.	50[2)]	60

Table 5. Recommended maximum cooling capacities for different systems and ventilation principles.

1) Using special versions of diffusers, heat loads of up to 150-200 W/mhave been removed.

2) The maximum recommended cooling capacity using ventilation air is approx. 20 W/m.

Air flow

Excess heat

The air flows required to remove excess heat are determined for both mixing and thermally controlled systems by using the temperature difference between the exhaust and the supply air. The following equation can be applied:

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

where
q = air volume in l/s
P = cooling requirement in W
t_f = exhaust air temperature °C
t_t = supply air temperature °C
ρ = air density (1.2 at 20 °C)
C_p = specific heat capacity of air(1.0 at 20 °C)

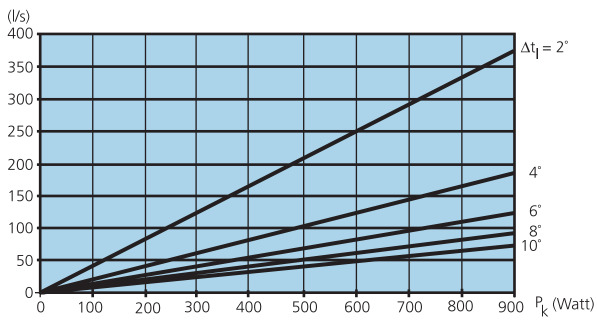
For a mixing system, t_f does not usually deviate more than a few degrees from the room temperature 1.1 m over the floor (normal reference point for the room's temperature). For a thermally controlled system, however, and with normal ceiling heights, assume t_f to be 3-5 °C above the room temperature.

The supply air temperature is usually restricted to:

15 °C for mixing ventilation and
18 °C for displacement ventilation.

Therefore, at maximum cooling load, the temperature difference (t_f- t_t) will be generally the same for the different ventilation principles as well as the required air flows. One condition for utilising large temperature differences between the exhaust air and the supply air is:

- For mixing ventilation: The supply air terminal unit must be correctly dimensioned according to the instructions in the chapter "Mixing ventilation".
- For displacement ventilation: The supply air terminal unit provides a wide distribution of air over the entire floor and the air distribution can be controlled according to the design of the occupied zone.



$$q = \frac{P_k}{1.023 \cdot 1.2 \cdot \Delta t_l} \quad l/s$$

Figur 26. Maximum air flow as a function of the cooling demand.

Air quality

A correctly planned and dimensioned ventilation system provides a higher air exchange efficiency if the system is based on displacement instead of mixing. In practice, however, the difference is not large. Therefore the same air volume can be used for both alternatives.

In general the following formula can be used to determine the required air flow in terms of air quality

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

where
- q = ventilation air volume (l/s)
- m = generation of pollution (l/s)
- C = recommended maximum pollution content (ppm)
- C_{in} = initial pollution concentration (background level) (ppm)

Example:
An office with people as the dominant source of pollution. A person sitting weight 70 kg, generates approximately 18 litre CO_2 per hour, (0.26 x 70), i.e., 0.005 l/s. The background level of CO_2 is a minimum of 350 ppm. In urban city areas this value can be considerably higher.

At a background level of 400 ppm CO_2 the required air flow per person is:

$$qv = \frac{0{,}005 \cdot 10^6}{C - 400} \text{ l/s, person}$$

In quality class AQ1, the requirement for CO_2 content is £600 ppm and in AQ2 £1000 ppm.

The required air volumes then becomes:

AQ1 = 25 l/s, person
AQ2 = 8.3 l/s, person

Activity	CO_2, l/h, kg
At rest, lying	0.17
Sitting	0.26
Standing	0.30
Walking	0.35

Table 6. CO_2 exhaled from a person per kg body weight.

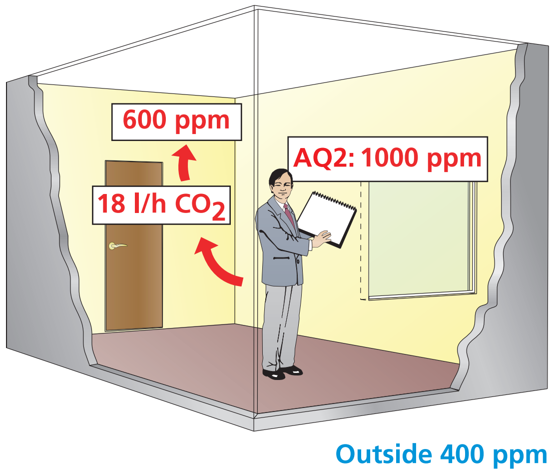
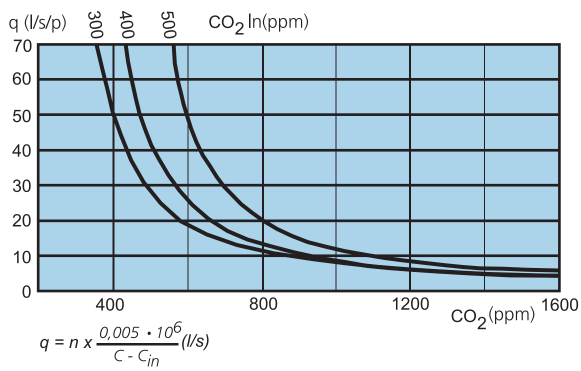


Figure 27.



Figur 28. Maxium air flow as a function of the air quality. (A person sitting, weight 70 kg.)

Traditional types of ventilation systems

The choice of a suitable technical solution is an important planning factor. The choice of system should be made after considering the following main factors:

Practicability. The ability of the technical solution to meet the required quality demands.

Reliability. The ability of the technical solution to function satisfactorily over time.

Efficiency. The efficiency of the technical solution regarding power, cost-effectiveness etc. When choosing a technical solution one should always strive to attain simplicity, intelligibility and ability to cope with fluctuations of operating factors. Avoid technical solutions that do not allow the layout of the premises to be changed, windows to be opened or which are in any other way sensitive to external disturbances.

Basic principles and characteristic properties:
There are different ventilation solutions that can be implemented to fulfil the demands for correct air flows to all parts of a system. The main categories are:

- CAV systems (Constant Air Volume), systems with constant air flows. The simplest and generally the "cheapest" alternatives.
- VAV systems (Variable Air Volume), systems with variable air flows, as a rule regulated by thermostats in each room. The fan is fitted with some form of pressure regulation device.
- DCV systems (Demand Controlled Ventilation), systems controlled by demand, as a rule regulated via an air quality or presence sensor.
- All system solutions can, of course, be designed for either mixing or displacement regulated ventilation.

Both CAV and DCV systems can be combined with different heating and cooling units for regulating the indoor temperature.

CAV system

CAV systems are used where both heat and pollutant production is low and reasonably constant. The flow of supply air is mainly determined by the quality demands on the air. If the hygienic air flow is not sufficient to remove the generated heat then products for waterborne cooling can be added. CAV systems are often designed according to the branching principle with an adjustment damper in each branch. The pressure drops across the terminals is chosen so that these, together with the pressure drops across the dampers, give the correct flow distribution.

The disadvantage of this principle is that the system can easily become unbalanced because of the effects of thermal lift, changes in damper settings etc.

Another disadvantage is the relatively high pressure drop across dampers and terminals that is required to ensure that the flow variations are not too large. In turn, this means that sound problems can be difficult to deal with and power consumption unnecessarily high. Lowering the fan speed, to lower power consumption during certain periods, means that the flow distribution cannot be maintained because the pressure drops across terminals and dampers is reduced.

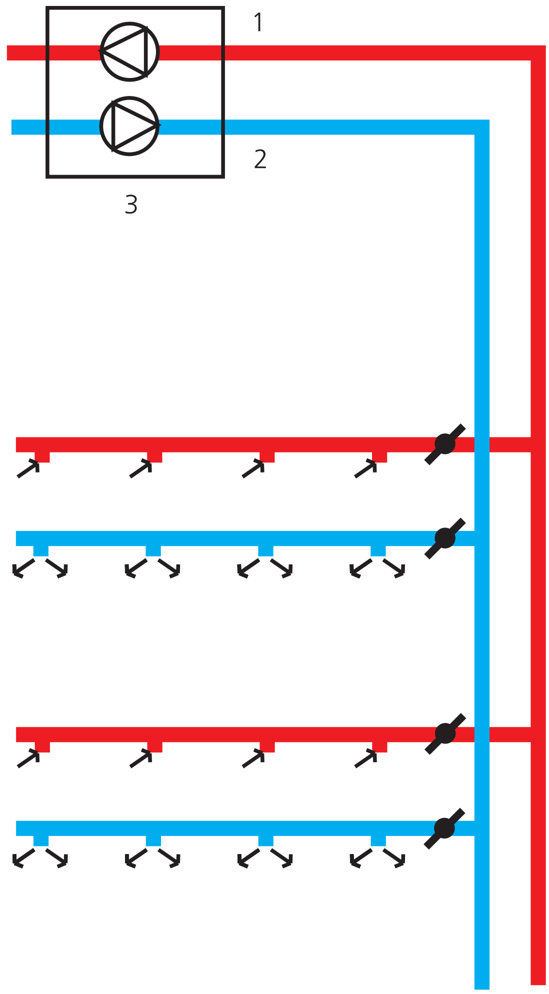


Figure 29. CAV systems, schematic diagram.
1. Exhaust air
2. Supply air
3. Ventilation equipment (FTX)

VAV/DCV-systems

VAV/DCV systems are used for variable occupancy rates. Heating is best achieved by using radiators. Cooling the room can be achieved by varying the air flow.

VAV/DCV systems differ from CAV systems in a number of ways, one of which is that pressure is regulated in the main ducts for supply and exhaust air. This is necessary from both a power and a sound point of view.

Another difference is that in the immediate vicinity of the supply air terminal there are dampers that regulate the air flow through the terminals. A fundamental problem with this is that when air flows are reduced the pressure drops increase. This can have serious consequences. Increased pressure drops lead, in general, to higher sound levels. The pressure in the main duct must always be sufficient to guarantee that requisite volumes of air can be delivered to the remotest branches.

If the flow distribution in the system should cause a temporary lower pressure, the setpoint value must still be maintained. This has a negative effect on the running costs of the system.

Demand controlled ventilation

It is generally accepted that if we, as users of an installation, can easily adjust the system, we perceive the system as better. In homes, for example, this means that residents can easily adjust air volumes to meet individual needs. In terms of the traditional FTX-system (fan controlled supply and exhaust air), we are not accustomed to the luxury of controlling the various air volumes in different spaces. Instead there has been an attempt to keep the air volumes as constant as possible. However, we are convinced that there is a clear advantage if residents can adjust air flows in individual rooms (within reasonable limits). This should be possible without altering the air volumes in other rooms. The minimum air volume in each space must be guaranteed.

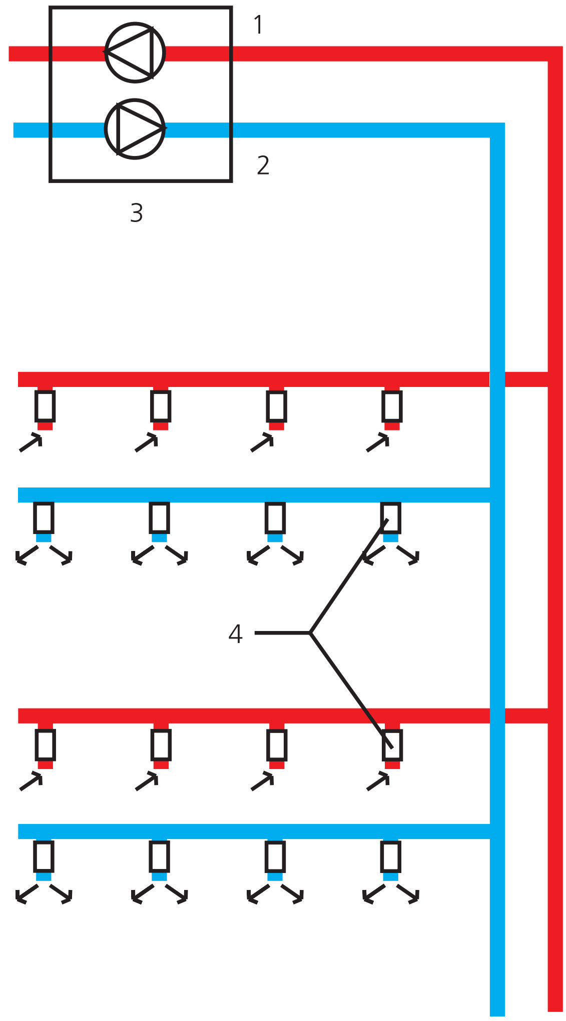


Figure 30. VAV system, schematic diagram.
1. Exhaust air
2. Supply air
3. Ventilation equipment
4. VAV unit

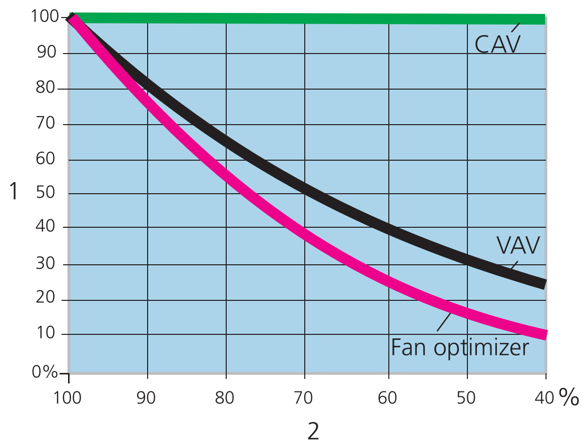


Figure 31. Relative power consumption depending on type of system.
1. Relative power consumption fan (%)
2. Relative air flow requirement (%)

General

Mixing ventilation can generally be used in comfort ventilation, i.e., regardless of whether the ventilation air is used for cooling or heating.

In the chapter "Ventilation principles", the flow forms for the different supply air alternatives were discussed. The design must take into consideration the following:

- degree of activity/room type
- room dimensions
- air volumes etc.
- possible cooling requirement
- resulting air velocity in the room
- resulting sound level

1. It is important to determine the degree of activity in order to decide what limits to apply for comfort.
2. The room dimensions influence the flow pattern and thus the comfort of the room. It is essential during the planning stage to correct throw data to ensure compliance with current desig regulations.
3. The least possible air volume is based on the hygiene requirements. For general ventilation in offices, 12 to 15 l/s, person can be considered to be the minimum outside air volume.
4. A calculation in which the internal and external loads as energy accumulation in the building are taken into consideration, must form the basis for calculating the required cooling. Together with the comfort requirements, this forms the foundation for the selection of the system solution and a suitable supply air flow.
5. The diffusers are described with a throw with a final velocity of 0.20 m/s. In different operating situations, this final velocity can be corrected so that the correct flow can be attained without draught problems. A description of this procedure is given in this section.
6. A calculation should always be made of the resulting sound from air terminals and duct systems in relation to the actual sound absorption in the room. The procedure for this calculation is given in the "Acoustics" section.

Other aspects which must be considered include vertical spread pattern:

When cooled or heated air is supplied vertically to a room or with a certain vertical injection angle, the heated air shortens and the cool air lengthens the throw depending on the density of the supply air. These conditions can be calculated and Swegon have developed a special computer program for this type of operation. The air flow, temperature variations between supply air and room air and the injection angle are given in this program.

Alternative mounting

The stated throw for slot diffusers, cone diffusers and perforated diffusers relates to ceiling-mounted units. If the supply air terminal is mounted in a freely suspended position and the jet is directed so that it adheres to the ceiling, the throw is reduced as a result of the induction at both sides of the supply air jet. The following conditions exist:

$l_{0,2}$ free suspended = $k_y \cdot l_{0,2}$

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

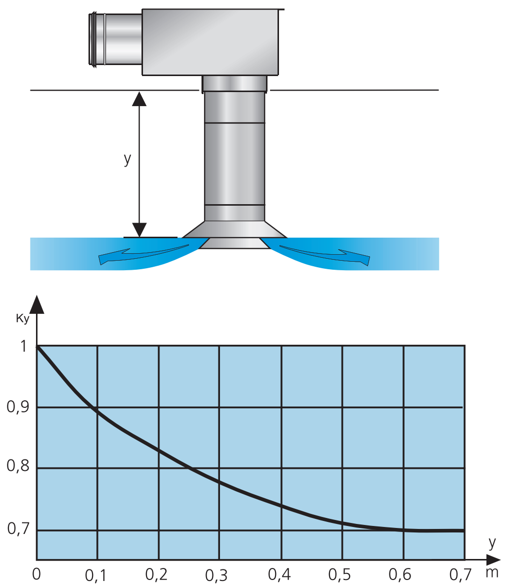


Figure 32. Correction factor ky, as a function of the distance y, between the device and ceiling.

The information applies to wall jets, i.e. with adherence to the ceiling. If the grille is mounted more than 0,2 m from the ceiling, the throw decreases in accordance with the formula:

$l_{0.2}$up to ceiling = $k_y \cdot l_{0.2}$

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

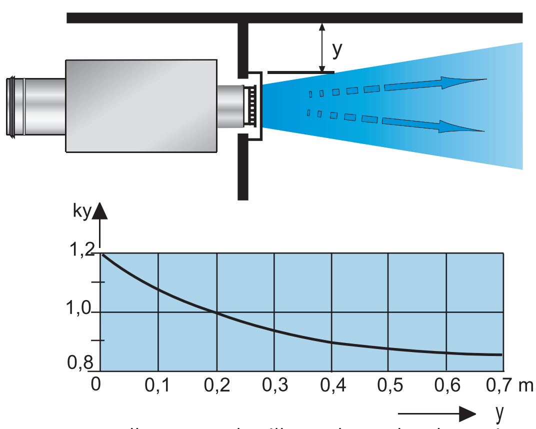


Figure 33. For wall-mounted grilles, where the throw is measured with the diffuser mounted 0.2 m from the ceiling, the above graph illustrates ($l_{0,2}$) for other distances between the grille and the ceiling.

Combining supply air jets

When two or more supply air terminals are positioned so closely to each other that their jets combine, the throw is lengthened. To calculate the extended throw, please refer to our calculation program ProAir, which can be accessed on our website or at your nearest sales office.

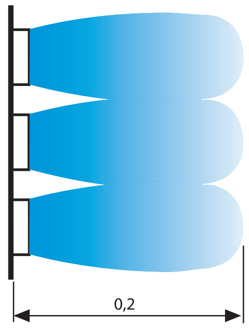


Figure 34. Combining of supply air jets.

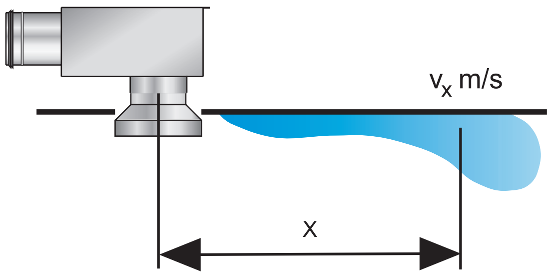
Throw

General

The throw should be stated with a terminal velocity of 0.2 m/s in accordance with VVS-AMA. For calculations using other terminal velocities, please refer to the ProAir calculation program.

Conversion of throw

For different reasons, a higher air velocity can be accepted when a supply air jet reaches the occupied zone or meets an obstruction, such as a wall. In a limited area the air velocity can be calculated according to the figure below.



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

Figure 35. Calculations of air velocity at the distance x from diffuser.

x = distance in m from the air terminal to the point in the jet where the air velocity is vm/s.

V_x air velocity at distance x from the air terminal device.

Example:
An air terminal has a throw of $l_{0.2}$ = 3 m. The throw $l_{0.3}$ then becomes:

$$l_{0.3} = 3 \cdot \frac{0.2}{0.3} = 2 \text{ m}$$

The minimum distance between supply air terminals

The minimum distance between two supply air terminals, which have their jets directed at each other, can be reduced. This is because the velocity of the core jets can be higher at the mixing point, without the combined velocity of the jets in the occupied zone exceeding 0.2 m/s. This is a result of the strong mixing effect of the two jets, which retards their velocities. The following relationship is applicable:

L_m = kv ($l_{0.2}$ unit 1 + $l_{0.2}$ unit 2)

L_m = The minimum distance between the supply air terminals.
k_v = Correction factor, see Figure 37.

Example:
Two supply air terminals, each with a throw of $l_{0.2}$ = 5.0 m have a minimum distance at
Δt = 6°C of L_m = 0.72 (5.0 + 5.0) = 7.2 m.

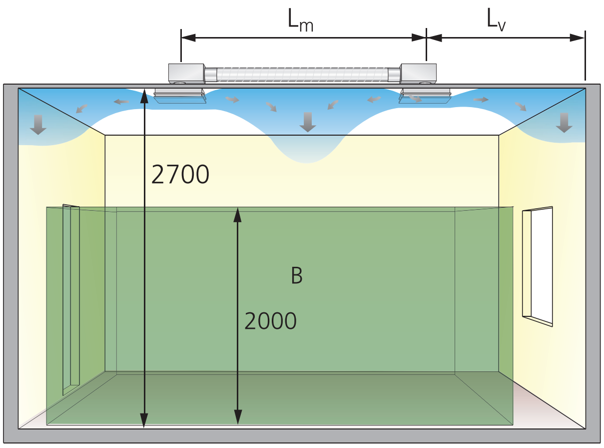


Figure 36. The minimum distance L_m between supply air terminals.
B = Occupied zone

The minimum distance between the supply air terminals and a wall

A jet which strikes a wall is permitted to have a higher terminal velocity than 0.2 m/s due to the retarding effect and the deflection which occurs.

The following relationship applies:

$L_v = k_v \cdot l_{0.2}$ K_v is obtained from Figure 37. Note that the formula above does not generally apply to outer walls, where convection flow or a cooling effect can occur.

Example:
A supply air terminal with a throw of $l_{0.2}$ = 5.0 m and Δt = 4°C can be placed L_v = 0.67 · 5 = 3.35 m from the wall.

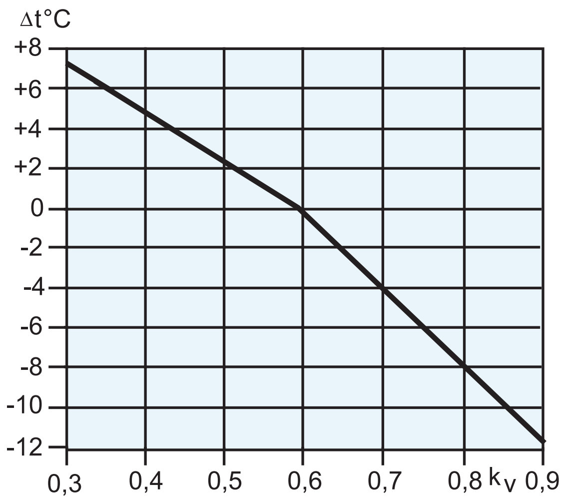


Figure 37. The relationship between correction factor k_v and the temperature difference Δt° C (t_{supply} - $t_{exhaust}$)

The minimum distance between the supply air terminals with increased ceiling height

The stated throw applies to a normal ceiling height of 2.7 m. With higher ceilings, the distance between the ceiling and the occupied zone is considered to be a retardation distance for the air jet. The figure illustrates the relationship between the distance between two supply air terminals and the distance to the occupied zone.

$L_{mA} = L_m - A$

Example:
Two supply air terminals, each with a throw of $l_{0.2}$ = 5.0 m and Δt = -6°C and mounted in a ceiling with a height of 4.5 m, can have a distance of 5.4 m. L_m = (5.0 + 5..0)· 0.72 = 7.2 m. Calculation of the distance L_{mA} = 7.2 - (4.5 - 2.7) = 5.4 m, i.e., the devices can be placed with 5.4 m between the inner edges.

The figure shows the relationship for the distance between the supply air terminal and the wall, which can also be corrected because the air jet has a longer retardation distance.

$L_{VA} = L_V - A$

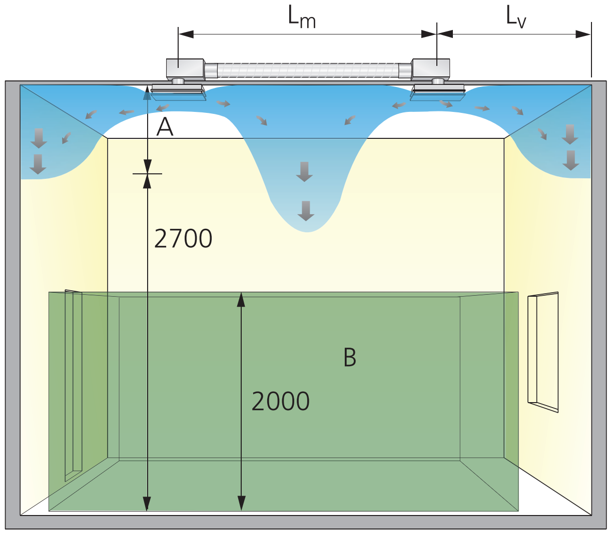


Figure 38. Minimum distance between supply air terminals with increased distance A + 2700.
B = Occupied zone

Minimum distance between the supply air terminals and obstructions in the jet

Hanging obstructions, such as light fittings, should not be placed near the supply air terminal.

Various alternatives for this situation:

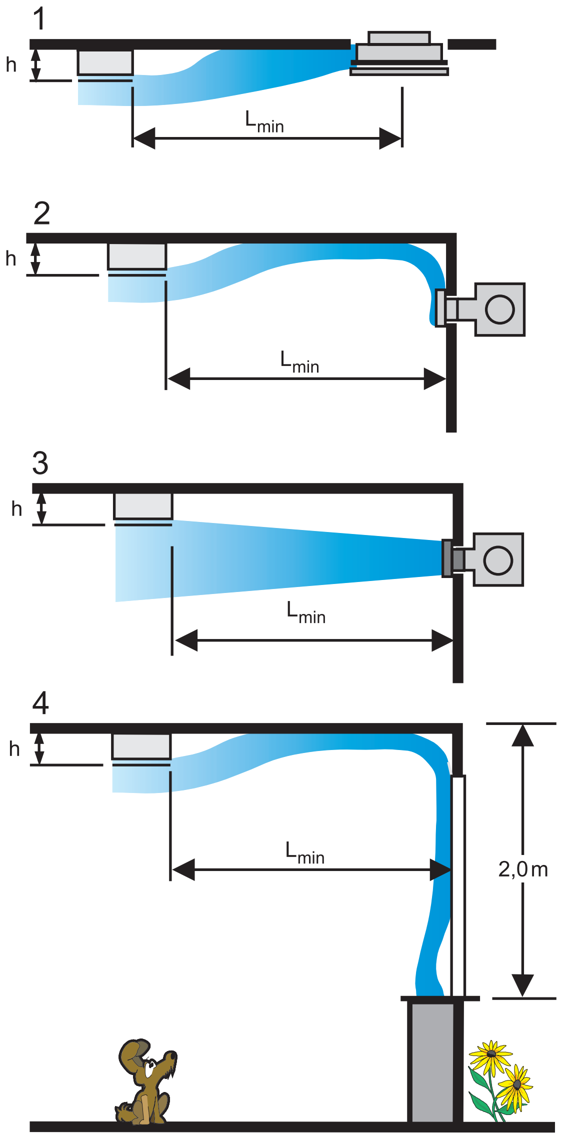


Figure 39. Different alternatives for obstructions in ceilings.
1 = Ceiling terminal
2 = Side wall terminal
3 = Wall grille
4 = Front, via window sill

For all alternatives, the minimum distance, L_{min}, depends on the spread pattern of the device and the supply air temperature. The shape of the obstruction is also important; an object with rounded or angled edges causes less distortion than an object with straight sides.

The following guideline values can be given:

Alternatives 1 and 2:
For air with a maximum of 6°C undertemperature, the following formula applies:

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

For higher undertemperatures, 50% higher values are recommended.

Alternative 3:
Grilles must be mounted at a distance from the ceiling ≥ 2h. If the hydraulic diameter of the grille is greater than 1.4 x h, there is no risk of a downdraught caused by the obstruction.

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

Alternative 4:
For this situation, the height of the obstruction must be restricted for different L_{min} according to the graph below.

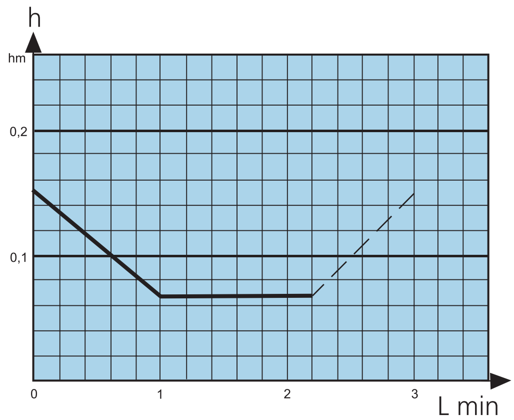


Figure 40. Critical height for a ceiling obstruction with a jet via a window sill, as a function of the distance from the sill.

Ventilate using the correct type of diffuser

The characteristics which are normally regarded as being important for supply air diffusers using mixing ventilation are as follows:

1. High induction of room air so that low supply air temperatures can be used.
2. Short throws for ceiling and wall diffusers, without the air slipping from the ceiling too early and thus causing draughts in the occupied area.
3. The capacity to supply large airflows without the throw being too long.

One method of ensuring the first two of these characteristics are fulfilled is to make sure that:

- The exit velocity of the air from the diffuser is high, which means that the exit area of the unit (A_0) needs to be small.
- The diffuser constant (k) should be low.

The same principles apply for item 3 above, with the proviso that there is a conflict between small A_0 and large air flow. Since the throw is proportional as in the formula:

$k / \sqrt{A_0}$

we can see that the diffuser constant should be small in order to achieve a short throw. At the same time, the exit area should be as large as possible. There arises a conflict here with the wish to create high induction, which is proportional to the expression:

$x/k \cdot \sqrt{A_0}$

x = distance from diffuser

The type of diffusers named rotation units are designed exactly to achieve a low diffuser constant and a relatively high exit velocity. One characteristic for typical rotation units is just their limited capacity in comparison with perforated units, for example.

In order to obtain as low as possible diffuser constant, one requirement is that the supply air is provided through ceiling diffusers and that it is spread evenly over the entire exit area of the diffuser. The spread angle should be 360° for the lowest possible diffuser constant.

Traditional rotation diffusers are designed to provide air through several long, rectangular slots, which are often radially oriented in a circular shape.

The principle of supplying air through a number of slots means that the exit area is limited and a high exit area can be maintained. One disadvantage with slots is that the flexibility is limited with respect to the possibility of achieving variations in the spread pattern.

The method which developed by Swegon, using infinitely adjustable discs, gives a far greater flexibility in this matter. Different characteristics and also possibilities are obtained with different numbers of discs. The size of the discs affects the possibility of varying the ventilation effects. The greater the number of smaller discs we have in a diffuser, the more become the possible variations.

The spread patterns which can easily be achieved with disc diffusers can be infinitely varied. The following variations can thus be obtained simply:

- All-round spread pattern
- 1-, 2-, 3-, and 4-way spread patterns
- Tangential spread patterns
- Vertical spread pattern
- Simultaneous vertical and horizontal spread patterns

The tangential spread pattern can be achieved in a number of ways. The most common of these is that the all the discs are aimed in the same direction, i.e. either clockwise or anticlockwise. This configuration gives the greatest induction. If a short throw is given priority, the discs can be set so that a negative induction is obtained. If the discs are arranged in rings, every second ring of discs can be oriented clockwise and every other ring anticlockwise. This gives rise to a powerful loss of momentum and thereby shorter throw.

Other factors which are seldom discussed also have some importance in this respect. One of these is the direction of flow when the air is just leaving the diffuser. To minimise the loss of momentum, the air must have an exit direction which is parallel to with the ceiling, see Figure 41. This is especially important in situations where large under-temperatures are involved. The use of rotation diffusers is especially advantageous here because they are well-suited to large under-temperatures. If the disc or slot does not fulfil these demands, there is a risk that if there are large under-temperatures the thermal forces will exceed the coanda effect. The result may well be undesirable flow patterns. A comparison of Swegon's disc diffusers and other slot or disc diffusers on the European market shows ours to have clear advantages.

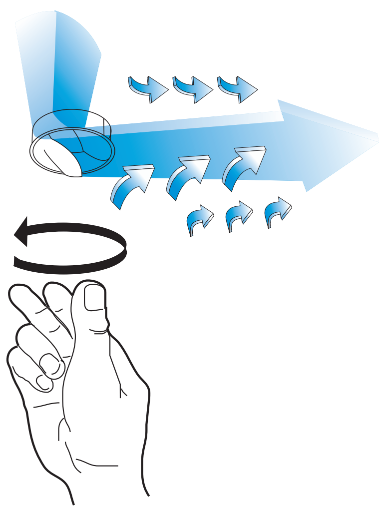


Figure 41. The working principle of the Swegon disc.

The table below gives a brief summary of characteristics of the most common types of diffusers:

Diffuser type	Characteristics
Disc diffuser ceiling mounting	Flexibility in settings, long and short throw possible. Vertical and horizontal spread patterns can be obtained. Large under-temperatures can be utilised. Wide variation of air flow without dumping. Handles large under temperature.
Disc diffuser, wall mounted	Flexibility in settings. Lateral and/or forward spread patterns. Long or short throw.
Guide vane perforated ceiling diffusers LOCKZONE	Large under-temperatures can be utilised. Fixed spread pattern. Flush mounted in ceiling. Capacity in between disc diffusers and perforated diffusers.
Guide vane perforated wall diffusers LOCKZONE	Large under-temperatures can be utilised. Fixed spread pattern. Capacity in between disc diffusers and perforated diffusers.
Perforated ceiling diffusers	Relatively large under-temperatures can be obtained, though not as large as with disc diffusers. Short throw lengths due to loss of momentum though the perforations. Different spread patterns possible by using mechanical devices. Suitable for large air flows.
Perforated wall diffusers	Relatively large under-temperature can wall be obtained, though not as large as with disc diffusers. Short throw lengths due to loss of momentum through the perforations. Different spread patterns possible by using mechanical devices.
Linear slot diffusers in ceiling	Limited flexibility in spread patterns. Long throw in spite of high induction.
Circular slot diffusers in ceiling (conical diffusers)	High induction with small slots due to a favourable proportion between the diffuser constant and the exit area. Relatively large under-temperatures can be utilised. Limited flexibility in spread patterns.

Table 7. Characteristics of various diffuser types.

General

Figure 42 shows a displacement system. Air is supplied with under-temperature at floor level and at ceiling level. The supply air spreads across the floor and gradually rises when it comes into contact with heat sources (e.g. people), which create convection streams.

The heat source in the figure represents a contaminant, in these case heated air, which is lighter than the surrounding air. The contaminant rises to the ceiling, and more air is drawn into the plume.

If the volume flow of air in the contaminated plume is greater than the ventilation air flow, when it reaches the ceiling, the contaminated flow may not be extracted directly with the ventilation air. Some of the polluted air is therefore recirculated downwards into the room. A front with polluted air then forms, which begins to move downwards.

The front stops at that level where the volume flow of the rising contaminated plume is equal to the ventilation flow.

Two zones are then created in the room, the upper zone with contaminated air and the lower zone with "clean" air. In rooms with high ceilings and where active work is carried out, it is desirable to make the clean zone as high as possible, and preferably over the breathing zone.

The required air flows are determined on the basis of applicable standards and hygiene limits.

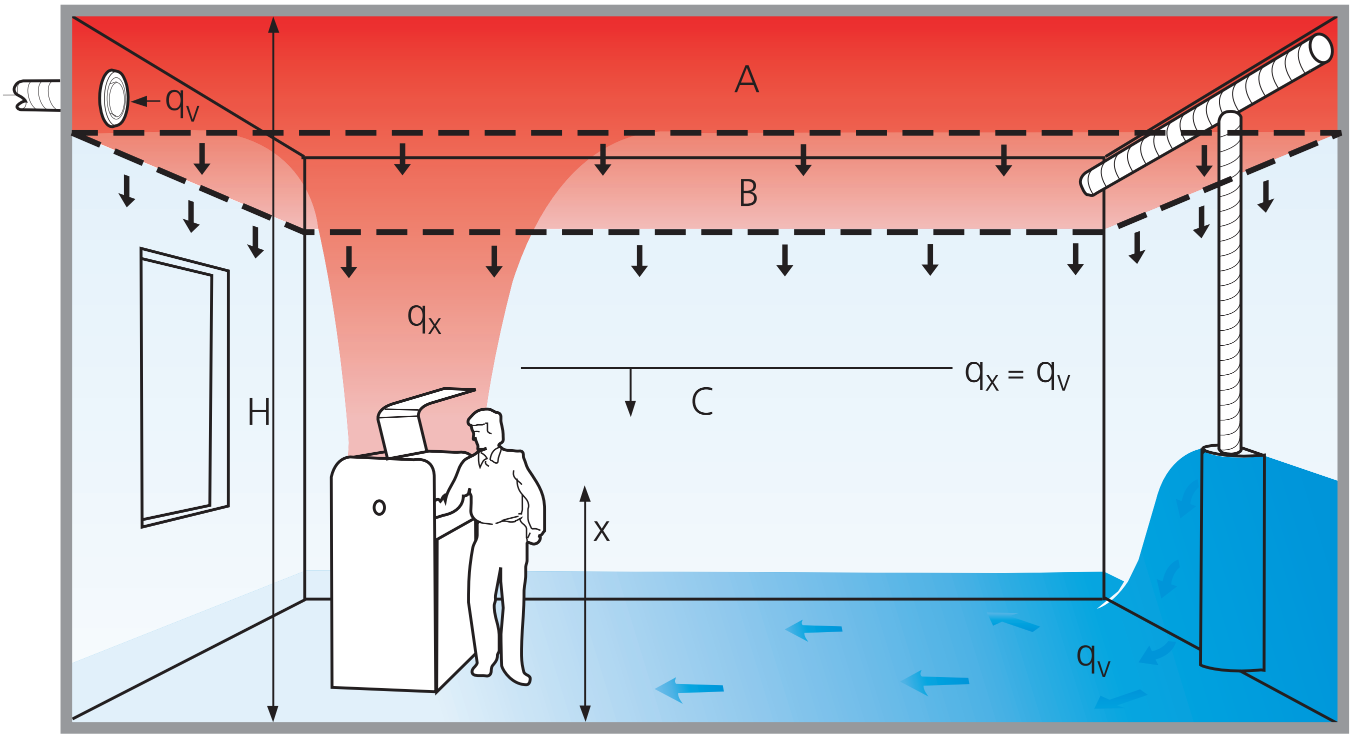


Figure 42. Displacement ventilation.
The air is supplied at under-temperature at floor level and is extracted at ceiling level. The supply air (q_v) is distributed across the floor and gradually rises when it comes into contact with heat sources.

q_v = ventilation flow l/s
q_x = convection flow in the polluted plume on the level x, l/s.

A = Contaminated zone
B = Mobile front with contaminated air
C = Clean area

Design

When using ventilation based on the displacement principle, the air is supplied at a very low velocity. This means that the spread profile in the room is controlled by the density differences of the air. The spread profile can be said to be thermally controlled. This means that other aspects must be considered than are normally observed with mixing ventilation.

It is therefore important to analyse the elements that shape the final conditions in the room in the design work. The following elements can be distinguished.

1. **Process Analysis**
   - Activity type/type of locality
   - Activity level
   - Convection streams
   - Room dimensions
   - Room layout

2. **Calculation**
   - Air flow
   - Energy balance calculation
   - Convection air flows
   - Resulting sound level
   - Affected zone

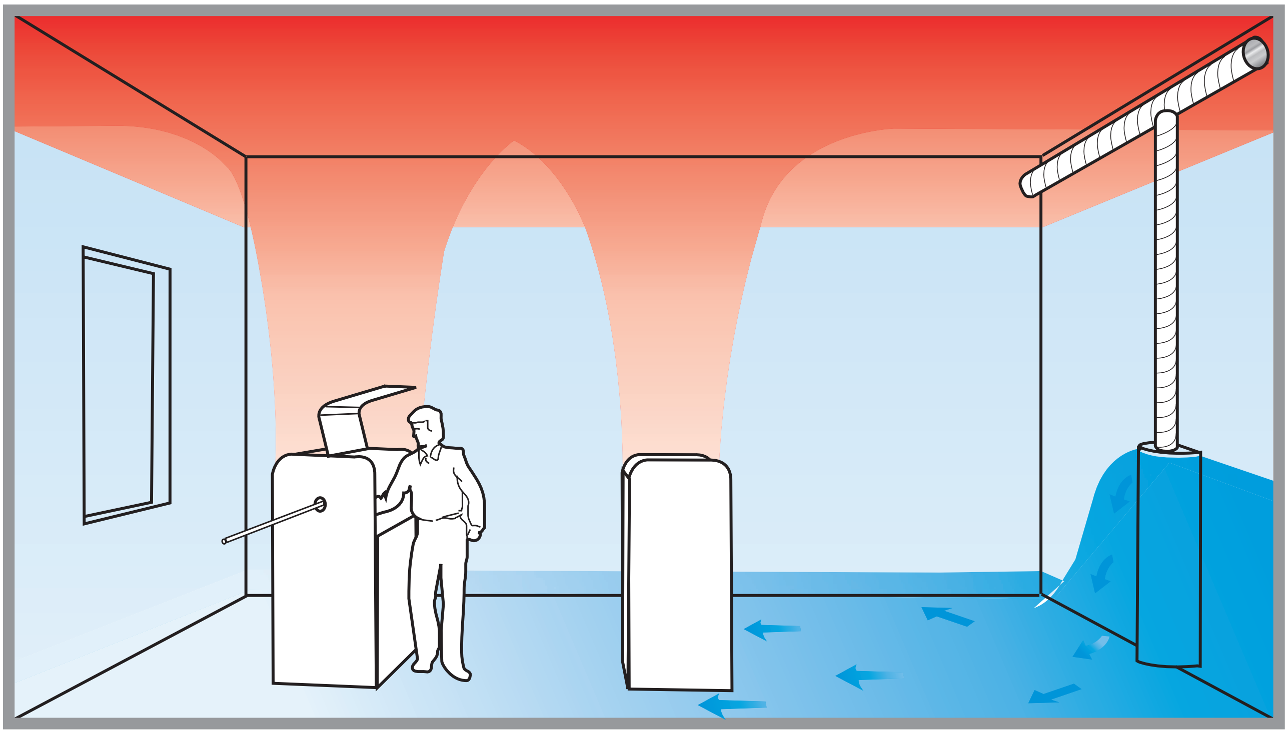


Figure 43. Displacement ventilation.

Process analyses

Type of activity/type of locality
The use of displacement systems in contaminated rooms is very suitable, since the terminals have a very small induction rate.

Activity level
In rooms where a high level of comfort is required, it is essential to determine the comfort requirements based on the activity level in the room.

Convection currents
The size and position of prevailing convection streams in any locality determine the air movements and consequently the efficiency of the ventilation. It is therefore critical, when designing a system, to carry out a careful analysis of machine size and their convection heat, the number of people and their activity levels, the effect of the sun and the heating or cooling effect from walls, radiators etc. If the design is to be based on the generation of contamination in the room and a maximum pollution concentration in the occupied zone, the air flow must be designed according to current norms and regulations regarding hygiene limit values. See "Calculations" for more information.

Room dimensions
The height of the room greatly affects the air exchange and ventilation efficiencies obtained.

A high ceiling gives more space for contaminated air, while with a low ceiling, the opposite is the case.

Room layout / General
Since the location and size of heat sources have a considerable influence on the final ventilation result, it is important to know where they will be placed.

A careful analysis of workshop machinery positions for example, is necessary during the design stage.

In comfort ventilation systems, it is necessary to know how the room will be furnished in order to choose the best location for the supply air terminals. Guidelines for this are illustrated by the following figures.

Room layout / Open-plan office
Try to find places where people do not sit regularly. Often there are distinct "walkways" in the room, and the terminals can be placed near them.

Flat or half-round terminals beside columns, or circular terminals in an office reference area or near a copying machine.

Other places which can be suitable for terminals in an office are a reference area or near a copying machine.

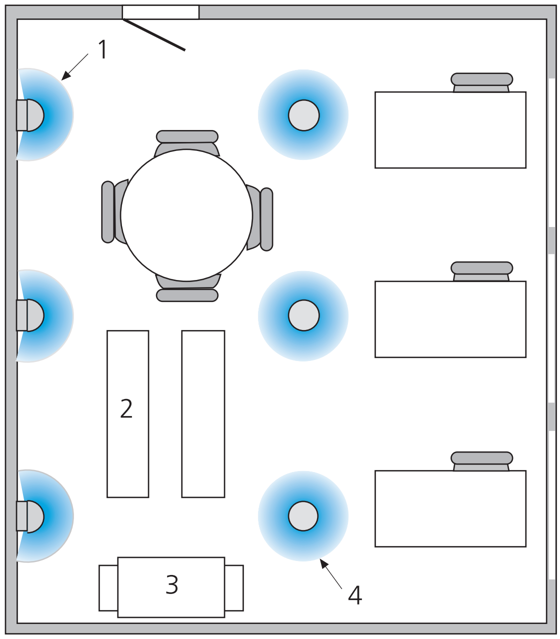


Figure 44. Terminals, placed in an open-plan office.
1 = Concentration of terminals far from work spaces
2 = Bookshelves
3 = Copier
4 = Alt. supply air terminals in the form of false columns

Room layout / Cell offices

In normal cell offices, the depth of the cell is often larger than the width. Try to place the device in or on a wall between the room and the corridor. This provides a suitable distance between the terminal and the person working at the desk.

Notes to Figure 45.

A. The terminal can often be placed in a recess close to the door. It is usual that no fumiture is located along the door wall. since the light switch is normally located there.

B. A terminal placed behind a door is not very suitable. The affected zone will be influenced and increased air velocities are created along the wall (in this case where a visitor is sitting).

C. If the device is placed near an outer wall, it should be moved to one side or the other depending on the location of the desk. The occasional visitor is not as sensitive to draughts as the office's occupant, who sits in the office the whole day.

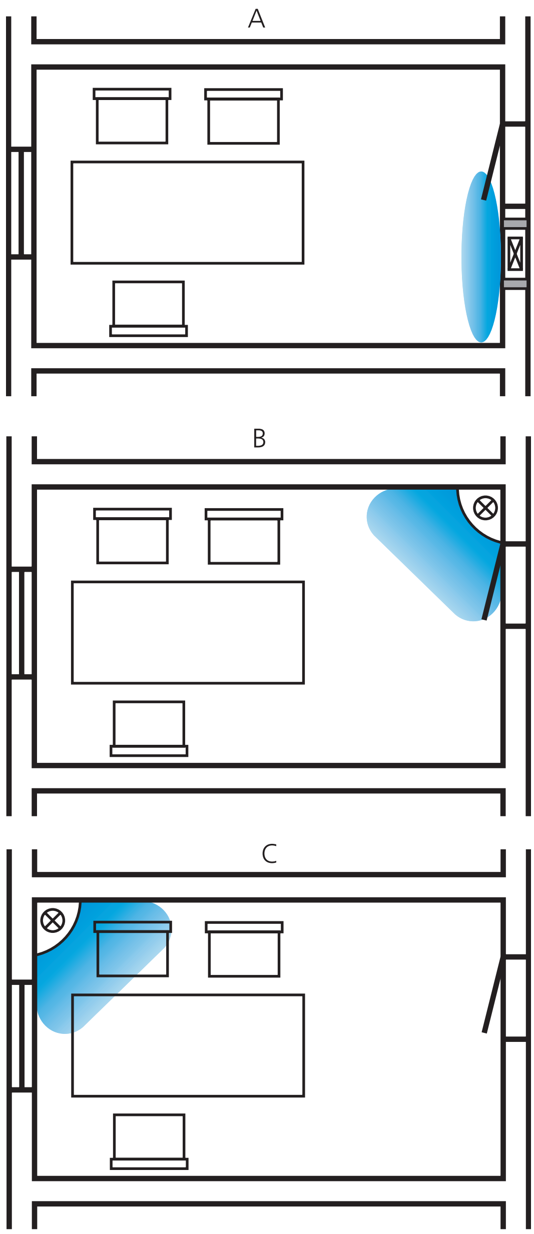


Figure 45. Terminals, placed in module offices.

Flexible spread patterns, affected zones

It is in most cases a great advantage if the diffuser spread pattern can be modified to prevent seating locations from being inside the affected zone.

Swegon displacement diffusers are equipped with the VariZone system. Every diffuser has a number of circular rotatable air deflectors behind the perforated front plate. The spread pattern can be modified by turning these in various ways.

This represents a clear benefit when the layout or function of the room changes with time.

One rule of thumb is that the spread of the affected zone over the floor area (m^2) which is stated in the catalogue can be changed from the standard setting to:

- Affected zone to the right
- Affected zone to the left
- Long and narrow affected zone

All these alternatives have one factor in common, namely that the spread area is the same as the standard setting.

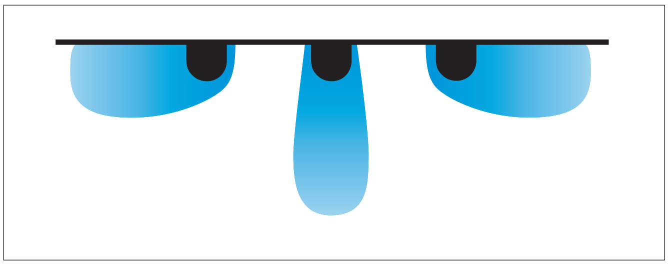


Figure 46. Examples of alternative settings.

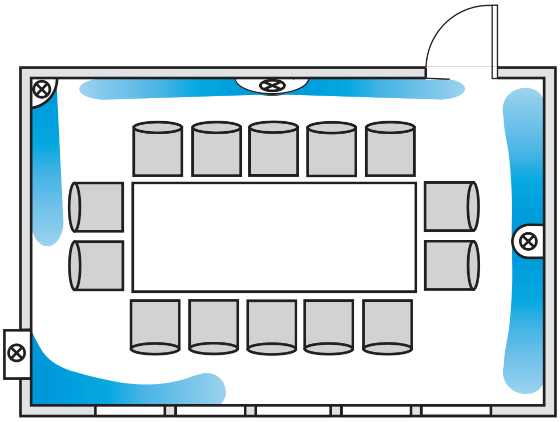


Figure 47. Diffusers whose affected zones have been adapted to suit the room furnishings.

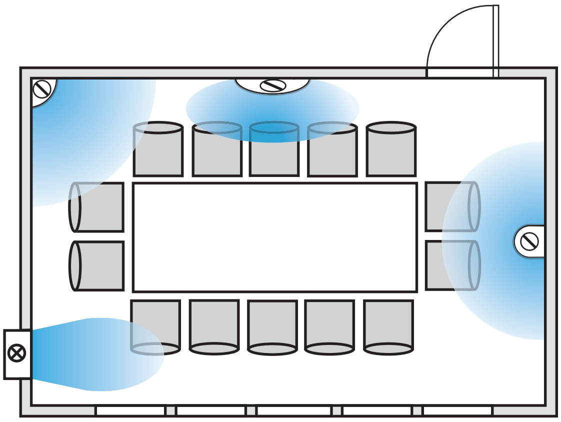


Figure 48. Diffusers without any changes in their spread pattern.

Calculations

Air volume

For industrial systems, where the ventilation system must be designed for a specific, maximum pollution concentration in the occupied zone, it is important that the supply air flows are determined according to the quantity of pollution released m (mg/s) in the room and the permissible pollution concentration c_{in} (mg/m^3).

The air flow q is obtained by:

$$q = \frac{\dot{m}}{C_{in}} \ (m^3/s)$$

According to this equation, the concentration c must always be less than the hygiene limit described in current regulations. If the pollution in question is also present in the supply air, the air flow q is calculated by:

$$q = \frac{\dot{m}}{C_{in} - C_t} \ (m^3/s)$$

where c_{in} = pollution concentration in the supply air in mg/m^3.
In comfort systems the minimum air flow is generally 0.35 l/s per m^2 floor area. For cell offices, however, the outside air flow should not fall below 12 to 15 l/s per person.

Energy balance calculation

A calculation of the internal and external heat loads, where consideration is given to heat accumulation in the building, must be the foundation for calculation of the required cooling load. Together with the comfort requirements, this provides a suitable basis for the selection of a proper system and supply air flows. Calculation of the ability of the displacing system to supply cooling power is included in these guidelines.

Convection air flows

The convection air flow for different heat sources in industrial systems does not need to be determined if the air flows are calculated according to the instructions given above. Even in comfort systems, the size of the convection streams should be ignored when selecting air flows.

Resultant sound level

The supply air terminal usually generates very little sound. The sound attenuation is also normally quite low, and so a careful sound calculation must be made for the duct system. Correction for any room absorption is made according to the chapter "Acoustics".

Affected zone

It should be pointed out that the affected zone size should be considered during the design phase. No persons should be expected to regularly occupy the affected zone. This is particularly important to remember in high-occupancy rooms. Selection of the device should be based first on the size and shape of the affected zone, and then on the sound generation of the device.

Figure 49. Calculations.

- Min. flow according to norms and regulations
- Energy balance calculation
- Resultant sound levels
- Affected zone

The relationship between air flow rate, temperature gradient and heat loads in displacement ventilation

One method of calculating the airflow required to limit the vertical temperature gradients at various heat loads is illustrated below. The method is taken from:

Memorandum 16 from Installation Techniques KTH, March 1991.

Symbols:

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

The temperature gradient must be limited so that it does not exceed the limit values which are given in "Requirements for interior climates".

The requisite lowest ventilation airflow for a certain maximum temperature gradient is obtained from Figure 51.

The temperature difference between 1.1 m level and supply air can be determined from Figure 52.

The air temperature at floor level (t_{lg}) is checked with the aid of Figure 50. This check is important for comfort ventilation - t_{lg} must not fall below 20°C.

Another practical guideline is that the supply air temperature must not fall below 18°C.

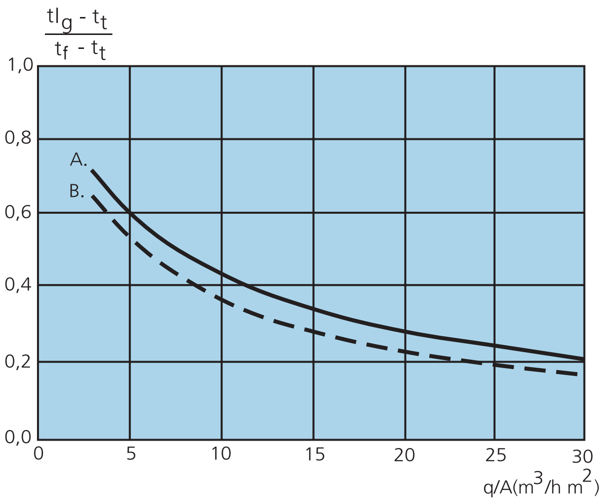


Figure 50. Non-dimensioned temperature difference at floor level for various air flows.

A = α_{kg} = 5 W/(m² · K) (heat transition coefficient due to convection at floor surface)

B = α_{kg} = 3 W(m² · K)

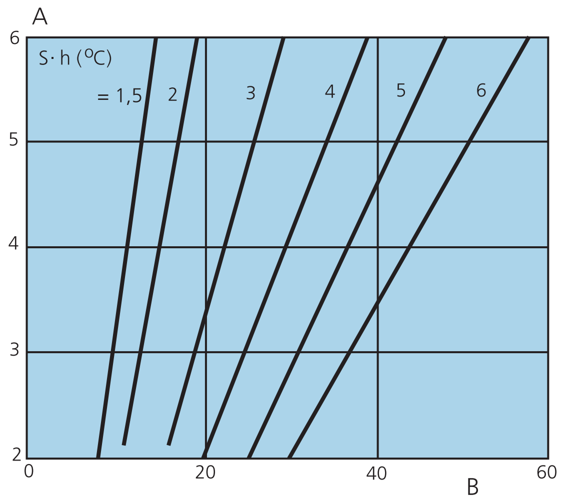


Figure 51. Requisite ventilation flow as a function of the cooling effect of the various products of gradient and room height.
A = Air flow (l/s,m²)
B = Cooling effect (W,m²)

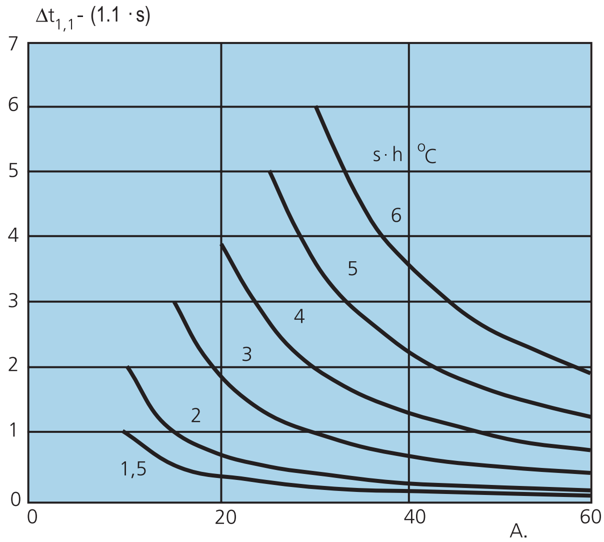


Figure 52. Temperature differences between air at floor level and supply air as a function of the cooling effect of the various products of gradient and room height.
A = Cooling effect (W/m²)

The method is described with the help of the following examples:

An office with a height of 2.7 m has a cooling requirement of 25 W/m^2.

The vertical temperature gradient must be limited to 1.7 °C/m.

Calculate the required air flow and the temperature at 1.1 m above floor level as well as at floor level.

Solution:

$s \times h = 1.7 \cdot 2.7 = 4.6$ °C

Figure 51 gives q/A = 2.8 l/s,m^2

Figure 52 gives $\Delta t_{1.1} - (1.1 \cdot s) = 3.6$

i.e. $\Delta t_{1.1} = 3.6 + (1.1 \cdot 1.7) = 5.5$°C

Set the supply air temperature $t_t = 18$ °C

which gives $t_{1.1} = 18 + 5.5 = 23.5$ °C

Figure 50 gives:

$$\frac{t_{lg} - t_t}{t_f - t_t} = 0{,}4$$

($q/_A$ = 2.8 l/s,m^2 or 10 m^3/h,m^2)

Since:

$$t_f - t_t = \frac{25}{2{,}8 \cdot 1{,}2} = 7{,}5^\circ\mathrm{C}$$

giving $t_{lg} = 0.4 \cdot 7.5 + 18 = 21$°C

i.e. required air flow rate = 2.8 l/s,m^2
air temperature at 1.1 m level = 23.5°C
air temperature at floor level = 21.0°C

General

Badly adjusted ventilation systems are often the reason why designs do not fulfil their intended purpose. As a rule, this is because technical installation instructions and details have not been fully carried out, and due to lack of knowledge the adjustment and measurement techniques.

Commissioning must be carefully considered at the design stage. It is also necessary to take into account any changes which may take place during construction. Commissioning can therefore be very time-consuming if it is not planned correctly.

It is important to allocate sufficient resources for this work. This should be seen as a safeguard to ensure that the installation functions as it should on completion.

Recommendations for the measurement of air flows and how this should be done are given in the report T22:1998 "Methods for the measurement of air flows in ventilation installations", which can be ordered from the Swedish Building Service.

Measurements

In a ventilation system, the deviation from prescribed values for an air flow must not exceed 15%, including measurement error. This requirement is stated in VVS AMA, Sweden.

Instruments must be calibrated using a recognised method with an acceptably small error. Calibration curves should be used where the correction is given as a function of the observed value.

Exact measurements can never be made. The measurement of air flows requires the following sources of error to be observed.

- Instrument error - denoted m_1, which could be caused by friction in the instrument or other similar problems, or it could be residual system faults from calibration.
- Method error - denoted m_2. For example, this could be a result of faulty measurement locations. This error is obtained for the recommended methods given in the report mentioned above.
- Reading error - denoted m_3. Caused by problems such as poor resolution on a scale.

With these errors as a starting point, a probable measurement error, m, in accordance with VVS AMA regulations, can be calculated:

$$m = \sqrt{m_1^2 + m_2^2 + m_3^2}\ \%$$

Measurement in ducts

There are three basic methods for measurements in ventilation ducts:

A1 - measurement with a Prandtl tube
A2 - measurement with a fixed measuring nozzle
A3 - measurement with a tracer gas

Method	Designation	Method error m_2
Traversing with a Prandtl tube in a duct with:	A1	
Circular cross section	A11	4-5% recommended 7% alternative measurement plan
Rectangular cross section	A12	4% recommended 7% alternative measurement plan
Fixed measuring unit	A2	See in built size error limits 5 and 10%
Deciding air flow with the aid of tracer gas	A3	5 and 10%

Table 8. Method for measuring the air flow in duct systems.

To reduce the time needed and to increase accuracy when commissioning, Swegon recommends that permanent measuring devices (method A2) should be used in as many cases as possible. Swegon has various types of measuring units; the most recent being CIM, intended for ducts that need to be accessible for cleaning. CIM can be easily removed from the duct during cleaning.

There is also the adjustable measuring unit, CRM, where the measurement function has been complemented with a damper for easy adjustment of the air flow.

The air flow is decided by measuring the pressure drop difference over the central body. The flow can then be produced from a special adjustment diagram, (MIS).

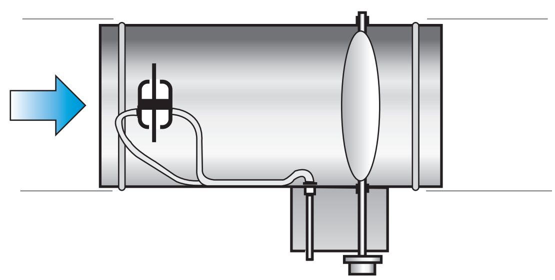


Figure 53. Adjustable measuring unit.

Supply air terminal

Three methods are recommended for measuring the air flow in supply air terminals.

Method	Designation	Method error
Pressure drop with a permanent measuring device	C2	5%
Bag method	C5	3%
Measurement with a conventional anemometer, with an extension sleeve	C3	5%

Table 9. Methods of measuring air flow in supply air terminals.

Swegon air terminals are supplied with a fixed measuring device in accordance with method C2. The measuring device is either placed directly in the terminal or in the multi-function plenum box, which is available as an accessory for the terminal. A manometer can be attached to the device and a characteristic pressure measurement can be made. The air flow is a function of the characteristic pressure difference.

In the plenum box there is an adjustment damper which is easily accessible through the device. The design of the box provides good sound attenuation and an even distribution of air through the terminal, which guarantees an even spread pattern.

In all our plenum boxes, the damper is easily removable for cleaning.

There are either one or two measurement outlets on diffusers, depending on their type and the plenum box used with them.

Balancing factor (k-factor)

For every terminal device or plenum box there is a balancing factor (k-factor), which has been specially determined for each product. The air flow is calculated using the following formula:

$$q = k \cdot \sqrt{\Delta pi} \text{ (l/s)}$$

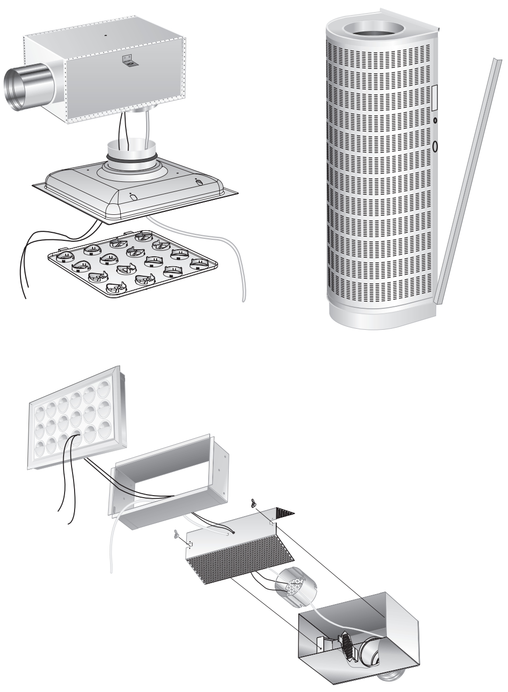
where q = air volume in l/s

k = balancing factor

Δpi = measured pressure difference (balancing pressure) in Pa

This procedure for determining the air flow applies to both A2 and C2 methods.

Examples of diffusers with one or two measurement outlets.



Figue 54. One measurement outlet.

Figure 55. Two measurement outlets.

Continual checking of the commissioning data is carried out at the Swegon laboratory.

Swegon has issued a special catalogue with instructions relating to the mounting, commissioning and maintenance of the product. These instructions are called MIS-instructions.

To simplify the commissioning procedure, the measuring contacts are lengthened using a plastic tube. The adjustment damper is supplied with a cord regulator (the white cord opens and the brown cord closes the damper), so that the adjustment can be carried out without dismantling the ceiling panels or terminal.

Commissioning of displacement terminal unit.

When dimensioning supply air diffusers for very low sound level or short throw, a low presure drop is obtained. This in turn means that the commissioning pressure Δp_i is low and there can be problems measuring the pressure.

Commissioning of supply air terminal with regulating cords and plastic measurement tube.

A working model for ventilation noise in mechanical ventilations systems

Background

One of the most common causes of complaint these days in the context of interior climates both at work and at home is excessive noise from ventilation systems, and this is also the case regarding the environment outside buildings. There are minimum requirements from SBN 75 which are in many respects sufficient, but unfortunately these are not always put into practice and neither are the requirements correctly formulated. Swegon, as one of the leading manufacturers of components for ventilation systems, has a our quality goal that all products which we manufacture must be able to contribute to a well -functioning, quiet ventilation system with a high comfort level if the components are used in the correct way. To match this ambition to provide quiet ventilation systems, we also describe a working method to apply in the planning stages. Needing a minimum of work, the method gives correct results with great precision.

Overview

All Swegon products are designed bearing in mind their influence on the four areas of climate which primarily determine the comfort in the rooms concerned. In addition to this, the products' energy efficiency and economic aspects are woven into the design. The four areas of climate are:

- Air hygiene
- Thermal climate
- Acoustic climate
- Visual climate

Measures taken regarding the ventilation system directly influence the first three of these areas, and work carried out in the room itself affects the fourth. This method of working means that acoustic factors, i.e. noise from the ventilation system and other acoustic effects, are given the same weighting in calculations as the primary functions of the ventilation system.

Stability and variations

Ventilation systems are in themselves unstable. There are many factors which affect the running conditions and which cause variations in flow, sound etc. The reasons for these variations can be many: degree of fouling of filters, running and maintenance conditions, the number of diffusers in operation, influence of outside wind etc. With regard to noise calculations, the point of departure is the worst case of operating conditions which can be continuously handled. This could mean that it is not necessary to take into consideration certain forced flows if they only take place during breaks and so on. Swegon have developed products and technology to create stability in systems even when the basic operating conditions vary. With respect to noise, the maximum sound level for these products needs to be calculated from the standpoint of the basic conditions which the ventilation system must handle. This applies to products such as in the e.r.i.c.-system.

Effects of the ventilation system on the acoustic climate

The ventilation system in a building affects its acoustic properties through sound generation, sound transmission in the duct systems, leakage in holes cut and an increased room damping by diffuser openings in the rooms. On top of these things there is also vibration from the fans. All of these aspects must be considered in the planning stages.

Sound generation:

Fans

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

Dampers

Sound power levels in dB are usually quoted by the manufacturer in octave bands. Swegon quotes octave bands between 63 Hz and 8000 Hz.

Diffusers

Inherent sound is often stated as a sound level in dB(A) related to room damping 10 m^2 Sabine, which means that the sound level applies at a distance from the diffuser in a room with a 10 m^2 absorption area. If the occupied area in the room extends as far as the diffuser, the fact that the sound level in the proximity of the diffuser is much higher than the sound level in the centre of the room must also be taken into account. See Fact Box 1.Air flow The airflow in a duct creates turbulence at points of unevenness, joints, connections etc., and thereby creates extra sound. Approximate levels of sound generated in ducts by airflow are given in Fact Box 2. Swegon's diffusers are in most cases designed with sufficient inlet damping to handle sound generation from an air velocity of 8 m/s in main ducts and a maximum of 4 m/s in branch ducts. Variations can occur, especially in rooms which demand low background sound levels. See Fact Box 2.

Damping

Fan unit attenuator

Fan units are usually equipped with unit attenuators. These are seldom sufficient, which is why extra attenuators are normally required near to the fan unit.

Resistive attenuators

The most common type are those where the sound and air flow alongside a sound absorbing material. The longer the attenuator, the more acoustic damping takes place. Resistive attenuators provide better attenuation at higher frequencies. The attenuation is stated in dB in octave bands and corresponds to the attenuation which would be obtained if the same length of duct was replaced by an attenuator. Angle attenuators give more efficient damping.

Reactive attenuators

A reactive attenuator can provide good damping even at low frequencies if the volume is sufficiently large. An example of a reactive attenuator is a plenum box which is lined on the interior with acoustically absorbent material.

Sound energy is assumed to be equally distributed over the total area and the attenuation effect is proportional to the ratio of the openings to the total interior area. Inlet and outlet orifices should preferably not be located opposite each other as high frequency sound can then pass directly through the attenuator. Fact Box 3 gives guidelines to help estimate the degree of damping. See Fact Box 3.

In normal circumstances sound energy is distributed in the various branches of the duct system in proportion to the area size. One rough method is to assume distribution of the sound in proportion to the air distribution in the system. There may be deviations which cause more sound at certain frequencies than this approximate method can cover, for example branch ducts in direct connections near to fans. However, with some caution this method can give a useful estimate of the sound attenuation characteristics of a ventilation system. See Fact Box 4.

Fact Box 1

Sound power level and sound pressure level

The calculation of the difference between sound power level (L_w) and sound pressure level (L_p) as a function of the room constant (A) and the distance from the source of the sound (r) with different positions of diffusers (Q) is done according to the following formula:

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

where Q = direction factor

r = distance from sound source (m)

A = equivalent absorption area

α_m = mean absorption factor for the total limiting area

Q =	1 Middle of room. free supply air
Q =	4 Wall near ceiling
Q =	2 Wall or ceiling
Q =	8 Corner position

With the help of the graph in Figure 57, the difference between sound power level and sound pressure level at different distances, r, and equivalent room absorption area values, A, can easily be determined.

Example:
In a room with 50 m² equivalent room absorption area, the distance between the occupied area and a supply air diffuser is 2 m. The diffuser is mounted in the ceiling (Q = 2). The sound level from the supply air diffuser and the duct system in according to catalogue information 43 dB(A). L_p - L_w will be according to the graph in Figure 57 (9 dB - 4 dB) 5 dB.
Sound pressure level is then L_p = 43 - 5, that is 38 dB(A).

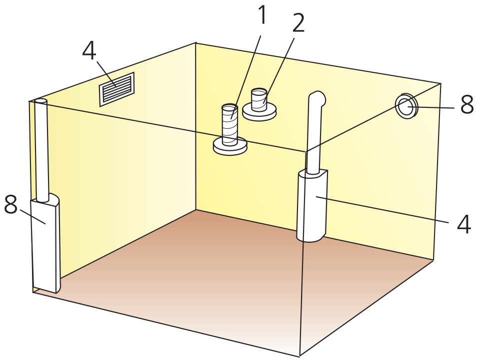


Figure 56. Directional factors for various diffuser positions.

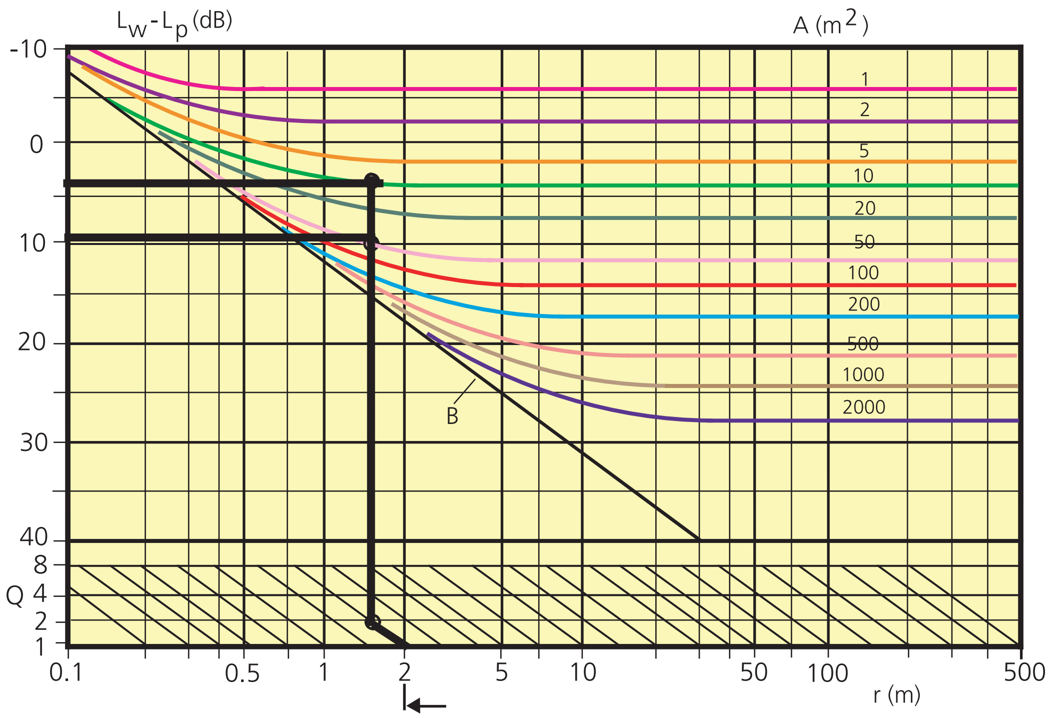


Figure 57. The difference between sound power level and sound pressure level.
Difference between sound power level and sound pressure level L_p - L_w, in dB
A = Equivalent room absorption area A in (m²)
B = Free field
r = Distance from the sound source, (m)

Fact Box 2

Sound generation in straight ducts

The sound power level for sound generated in a straight duct can be obtained from the equation:

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

v = air velocity in the duct in m/s
s = cross-sectional area of the duct in m^2

The distribution of the total sound pressure level in octave bands is obtained approximately from the graph below:

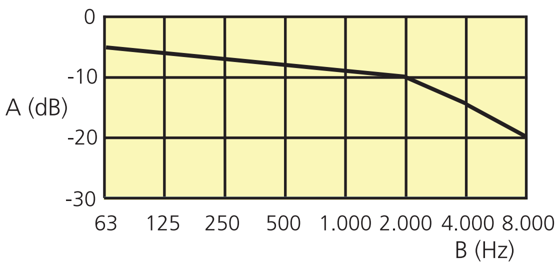


Figure 58.Distribution of the total sound pressure level in octave bands (straight ducts).
A = Relative sound power level in dB/octave (dB, over 1 pW)
B = Mean frequency in octave bands (Hz)

Example:

Air velocity, v = 10 m/s

Duct area, s = 0.5 m

Sound power level, L, will then be 57 dB.

The total sound power level is distributed in the various octave bands as below:

	Mid-frequency Hz							
	63	125	250	500	1000	2000	4000	8000
Exit level dB	57	57	57	57	57	57	57	57
Correction as fig. 58	-5	-6	-7	-8	-9	-10	-15	-20
Octave band level	52	51	50	49	48	47	42	37

Fact Box 3

Attenuation in interior-lined suction or pressure chambers

If the sound from a fan travelling towards a room passes through a chamber which is lined with sound absorbent material, the sound will be attenuated in proportion to the difficulty with which it escapes the chamber.
The graph to the right can be used to determine the attenuation obtained in a chamber lined with 10 cm mineral wool.

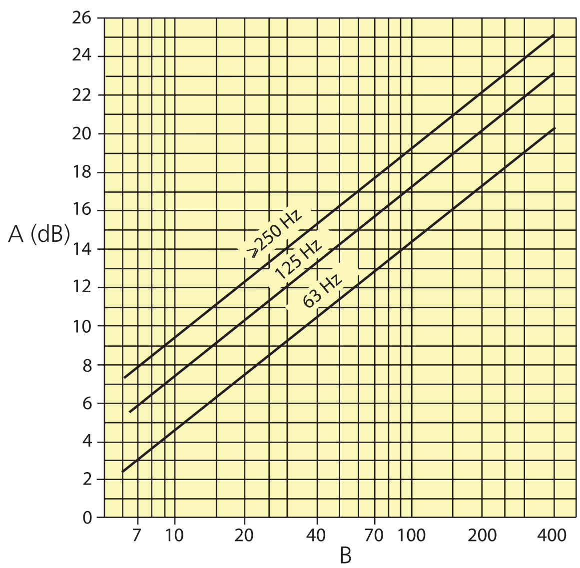


Figure 59. Attenuation obtained in a suction or pressure chamber lined with 10 cm mineral wool. (N.B. Inlet and exhaust ducts must not be placed opposite each other.)
A = Attenuation in (dB)
B = Lined area divided by exit area

Fact Box 4

Branch attenuation

At branch ducts the sound effect is divided proportionally in relation to the various duct areas, i.e., (A_1/A_2). In cases where the air velocity in all ducts is relatively equal, the sound effect will distribute itself in the same way as the air volume. A branch duct which transports 10% of the total air quantity will also contain 10% of the sound effect.

High frequencies can be compared to light rays; using this comparison, we see that only a minor part of the high-frequency sound is propagated in the branch (see figure). In this case, we reduce the attenuation for high frequencies (> 500 Hz). In a T-pipe, however, the sound energy will distribute itself according to the same relationship described for the duct areas.

The graph can be used when one considering the relationship between air volume and area.

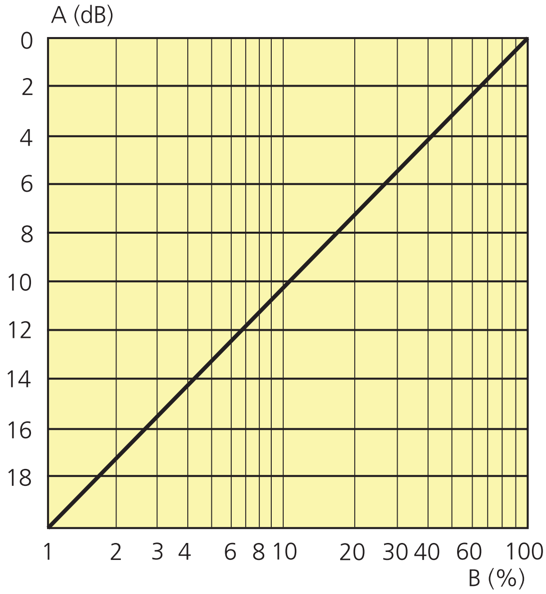


Figure 61. Relationship between % branch air and attenuation.
A = Attenuation (dB)
B = Air quantity to room per terminal (%)

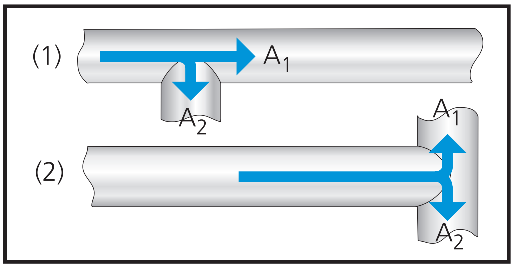


Figure 60. Propagation of sound in branches.

Duct attenuation
There is very little sound attenuation in duct systems with the exception of bends, curves and changes in dimensions where reactive attenuation is obtained. These effects are shown in Fact Box 5.

Inlet attenuation
Attenuation in a diffuser is stated in dB per octave band and normally refers to the attenuation from sound power in the duct to the sound level in a room with 10 m^2 Sabine room damping. Other references also exist, so be aware of possible misunderstandings. Note too that some diffusers act as sound amplifiers, i.e. the inlet attenuation can be negative.

Swegon normally states the attenuation for dampers at 10 m^2 room damping. Larger units designed for industrial applications are quoted for 150 m^2 room damping. Fact Box 1 provides the additions which are necessary to compensate for the near zone of the unit and commissioning in a room with damping which deviates from the 10 m^2 Sabine reference. See Fact Box 5.

Acoustic comfort requirements
We at Swegon recommend that requirements are stated in both dB(A) and dB(C) to ensure that sound levels are both sufficiently low and at the same time have an acceptable content without any dominating rumbling. Swegon has for many years designed and investigated the effects of products with respect to low frequencies. Values are stated in the product descriptions down to 63 Hz. For information on even lower frequencies, 31,5 Hz, please contact any sales office.

Examples of required values:
Swegon recommends that the dB(C) demand is at the most 15 dB higher than the dB(A) requirement, which provides good acoustic comfort in the room.

Housing	30 dB(A)	45 dB(C)
Offices	35 dB(A)	50 dB(C)
Classrooms	30 dB(A)	45 dB(C)
Conference rooms	30 dB(A)	45 dB(C)
Health care rooms	30 dB(A)	45 dB(C)
Cleaning rooms	45 dB(A)	60 dB(C)

It is important that all areas used by people have clear requirements for sound levels. Values which are higher than 45 dB(A) 60 dB(C), should not be accepted in rooms other than fan rooms, machine rooms and similar with internal sources of noise. The sound generated by ventilation systems should not be the dominating component in any room where there are demands placed on the sound level.

Why dB(C)?
Historically, the three weighting curves A, B, and C were established with the idea that the A-weighting would be used to measure the weakest sounds, B for the higher sounds and C for the most powerful sounds. In all contexts, the acoustic level was intended to match that of the human ear. Since our hearing does not have the same sensitivity at different frequencies, and in addition that sensitivity varies with the strength of the sound, several different weighting curves were necessary. International comparisons showed that none of the value measurements were superior to any of the others. To indicate the perceived strength of sound, measurements were simplified to only A-weighted, which in many cases gives a reasonable indication of the heard sound. Sound generated by sources with much low frequency noise are often evaluated incorrectly as the A-weighting obviously reduces the low frequencies far too much. This is especially true when the evaluation is not only based on audible sound but on the disturbance of people's activities caused by sound.

In our presentation of products, we always consider their low-frequency characteristics. For attenuators, throttle dampers, inlet attenuation of diffusers and so on, information on sound is stated in octave bands down to 63 Hz, which is in most cases sufficient to assess sound production and transmission in dB(C) from fans and other components. Inherent sound generation in diffusers is normally only stated in dB(A) for the simple reason that this type of sound is dominated by medium and high frequencies and does not therefore contribute much to the C value in the room. Background values in the different octave bands are available as corrections to the dB(A) values, so that it is always possible to calculate the C-weighted values.

Working method
Principle
This method follows the traditional planning procedure, with preliminary drawings, system documentation and construction documentation. The method entails:

1. In the initial stage, determining the measures required to attenuate the sounds emitted from fans that reach rooms and other surroundings, as well as designing the system in principle.
2. When the building construction planning stage is complete, carrying out detailed dimensioning and specifying the preliminary overall measures more precisely.

During the first conceptual stage, one must, with the architect:

- Decide on the geometry and volume of the building.
- Propose the number of air handling units and necessary rooms for these, taking into consideration the form and function of the building.
- Decide on the size of the rooms for the air handling units.
  N.B. The size here has more dimensions than just L · W · H - e.g. sound level, dynamic forces on joists, power requirements etc.

The natural steps

A

The size of the air handling unit can be stated when the volume to be ventilated is known. The overall sound power level of the unit can then be determined followed by the necessary measures to reduce this level to one which is acceptable both inside the building and outside. Fact Box 6 shows an approximation of fan-generated sound.

The dimensions of the air handling unit room are stated, i.e.:

- the required area in m^2 and
- the sound level required and
- vibration damping for joist requirements

The size of the room is determined by the size of the necessary sound attenuation materials which must reduce the noise transmission to the surroundings and the sound power level in ventilation shafts to less than 60 dB(C) in ducts. All low frequency attenuation must take place in the air handling unit room.

All sound traps which contain porous materials must be accessible. The sound traps must be opened able and the porous material replaced.

Vibration damping of the unit puts demands on the floor structure and the height of the room. This must be decided at an early stage so that the correct floor structure or supplementary measures can be dimensioned. See Fact Box 6.

B

All requirements for sound levels must be summarised in one document. These shall include:

- maximum dimensioned sound level dB(A)/dB(C) in the air handling unit room, designated by V
- maximum dimensioned sound level dB(A)/dB(C) in the ventilation shaft, designated by V
- maximum dimensioned sound level dB(A)/dB(C) in corridor cielings etc., designated by V
- sound level in room dB(A)/dB(C), designated by Acoustic or A
- acoustic insulation in dividing walls and floor structures R_wdB, designated by Acoustic or A
- noise emission in surroundings, L_w sound power in exhaust air and waste air openings, designated by Acoustic or V.

C

Design principle documents are drawn up indicating the size of low-frequency attenuators in the air handling unit room and attenuators used in the duct system, including their distribution as required next to rooms.

The following should also be shown: sound attenuators used after throttle dampers next to rooms in office buildings and similar. In school buildings and next to conference rooms with higher demands on transmitted sound this will normally mean a 1000 mm attenuator between the throttle damper and the room. The different attenuators are usually given the designations LD 1, 2, 3 and so on, and are dimensioned at the construction measures stage. The duct installation should be so designed that there is no need to cut through partition walls with high demands on acoustic insulation.

D

All duct installation shall be designed so that it can be carried out with high precision and good sealing. It is important that the construction plans clearly show the design at all through-connections so that they can be made and sealed in a satisfactory way. The planner has responsibility for these design details. In addition, the design must be such that the installation of the ducts can take place without risk of injury to the fitter, this also being the responsibility of the planner.

At the end of the planning stage, the sound conditions at some of the critical system branches are calculated. It is seldom that the sound levels for the entire system need to be calculated. The sound attenuators for the air handling unit and duct system are specified in detail at the same time as the above calculations are made. The designations for the various sound attenuators are translated into proposals for suitable models from the Swegon range.

In the detailed specifications, the sound level for each system is determined with a 5 dB margin. This means that when the supply and exhaust systems are in operation simultaneously the error margin will be 2 dB, which is necessary when considering variations in workmanship and the inherent margin of error in measurements made. If the building contains other systems such as cooling, heating and so on which can be expected to affect the sound level in the room, the margin should be increased by another 2 dB per additional system. For example, in the case of three systems all producing sound, the total sound shall be calculated for a requirement which lies 7 dB under the prescribed limit. If they contribute in equal amounts to the sound level in the room, this means that they together generate a sound level which is 5 dB over that dimensioned for each system separately, i.e. the margin will be 2 dB under the requirement. See Fact Box 7 regarding the aggregation of sound sources. In practical terms, the supply and exhaust air system in a building with three systems including the heating system should be dimensioned to manage 23 dB(A) in for example classrooms and residential rooms. See Fact Box 7.

E

A further document should be added to the construction plans which describes holes to be cut, through-connections and methods of sealing and later repairs with respect to acoustic and fire requirements as well as flexibility. The degree of sealing is critical if the acoustic requirements in the building are to be fulfilled. Swegons products are designed from the start to be easily recessed in walls, joist structures etc, and to be easily and securely sealed.

The construction phase

During this phase, all the usual shortcomings in workmanship and design which can cause undesirable noise must be continuously checked. These include:

Changes in materials

Equivalence in acoustic properties must be checked. Demands should be made for documentation of equivalence for all the aspects stated below.

Fan mounting

The planned vibration damping suspension must be checked. The fan unit must be suspended in a flexible mounting and be able to move freely from the point of rest. Even large concrete foundation pieces must be able to be rocked back and forth by hand.

Duct installation, sealing and pressure drop

Ducts shall be connected without sharp inside edges. Leaks cause sound. Unnecessary pressure drops cause sound and deteriorated operating conditions for the fans. Check that the duct installation is even, with smooth bends and changeovers in dimensions.

Final pressure drop

It is wrong to have too large a pressure drop from diffusers which results in increased sound levels. Most diffusers can be adjusted and this possibility should be utilised to even out the flow between rooms in series, but not to cut down the flow to the distribution of the whole system.

Commissioning

This must be carried out so that the diffuser does not produce too much self-generated sound in the room.

Checks

The sound levels must be checked during the commissioning procedure. Sound on the outside of the building is also measured.

Fact Box 5

Attenuation in bends

When sound in a duct hits a bend, part of the sound is reflected back along the duct. The extent of the reflected sound depends on the dimensions of the duct and the wavelength of the sound.

Attenuation in rectangular bends is considerable larger than circular section bends. The following table can be used to determine the attenuation effect. The attenuation for rectangular bends, see Table 10.

The lining material shall be of the same length as twice the duct width;
The thickness of the lining material t, shall be minimum of 0.1 · B.

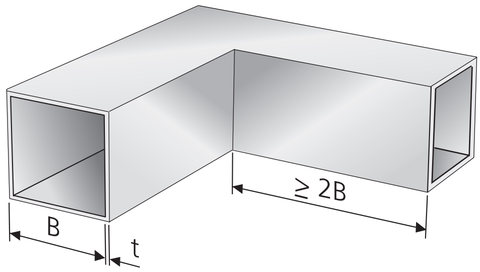


Figure 62. Interior lining in bend with attenuation.

	Duct width	Attenuation in dB at octave band mid-freqency (Hz)						
	mm	125	250	500	1000	2000	4000	8000
Without acoustic 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 bend (1)	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 bend (1)	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 bend (1)	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

(1) See figure 62.

Table 10. Attenuation in rectangular bends with and without acoustic lining.

Fact Box 6

Calculation of sound power level of fans

In those cases where the sound data for a fan is not available, the sound power level can be calculated with sufficient accuracy for most applications.

The following formula can be applied:

$L_{Wtot} = 40 + 10 \log q + 20 \log p_t$ dB

Air flow q is stated in m³/s
Pressure setting p_t stated in Pa

The above equation is illustrated in the graph below.

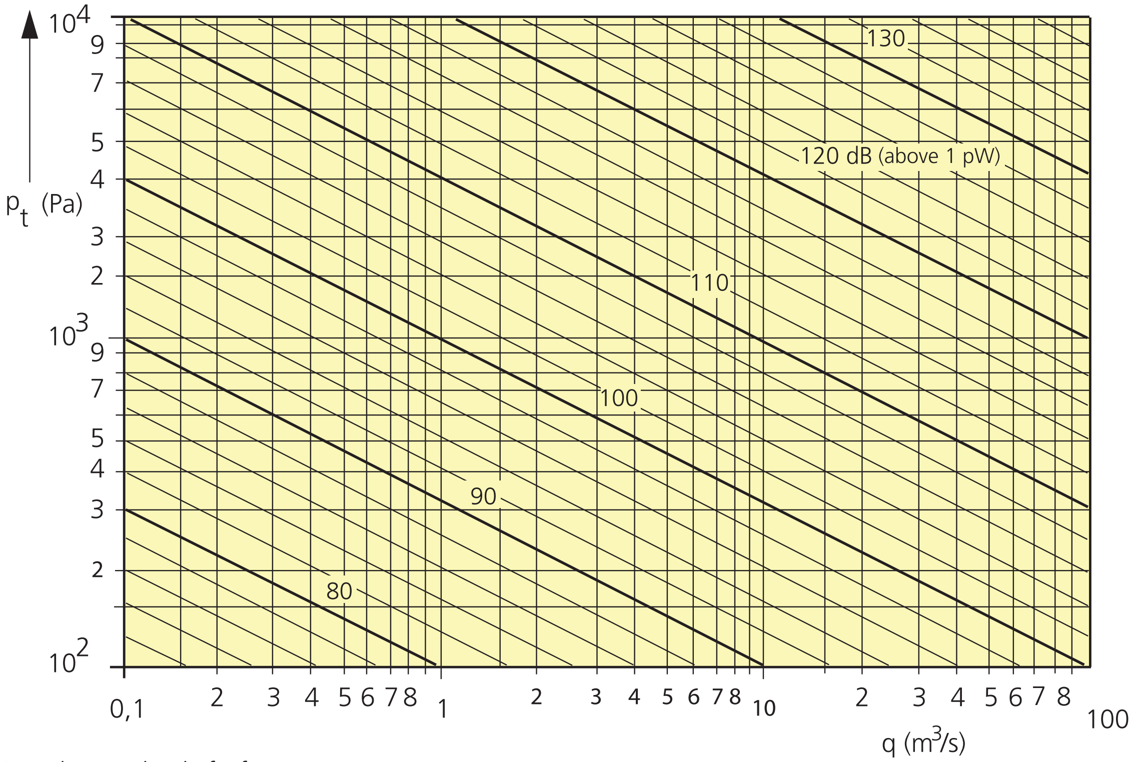


Figure 63. Sound power level of a fan.

Fact Box 7

Adding sound levels together

All of the sound sources in a room can be added together logarithmically. The process of addition can be done with the aid of graphs for a number of identical or different sound sources.

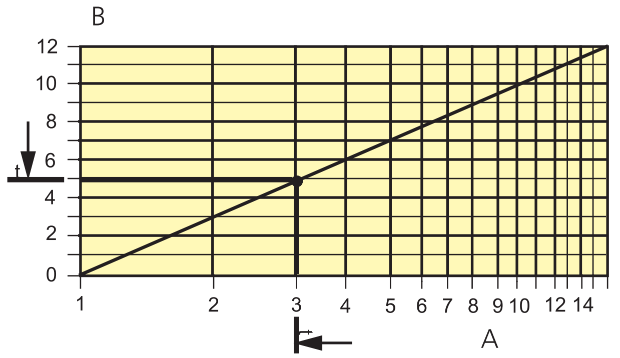


Figure 64. Logarithmic addition of several different levels.
A = Number of sound sources.
B = Increase, to be added to level from sound sources (dB)

Example:
There are three exhaust air terminals in a room, each producing 25 dB(A).
They produce a total sound level of 25 + 5 = 30 dB(A).

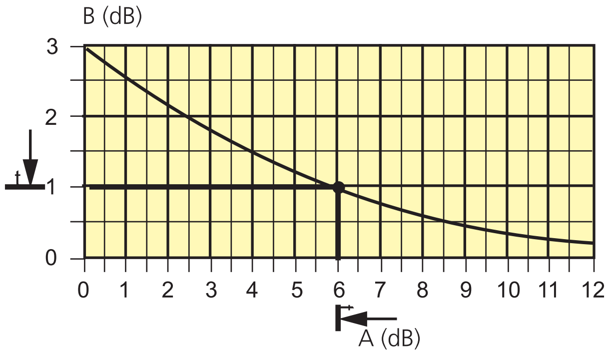


Figure 65. Logarithmic addition of two different levels.
A = Difference between levels which is to be added (dB)
B = Increase, to be added to level from sound sources (dB)

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

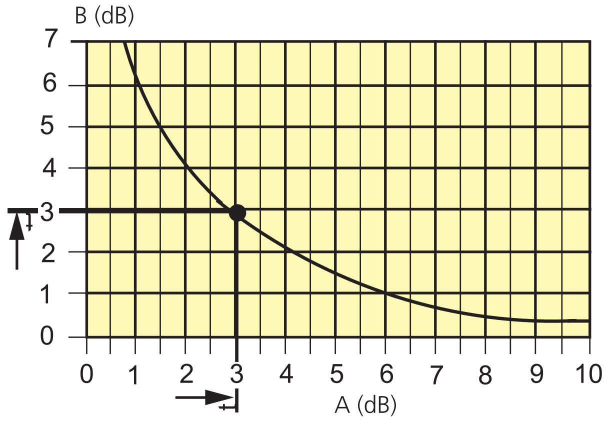


Figure 66. Logarithmic subtraction of two different levels.
A = Difference between sum of levels and level from sound source 1 (dB)
B = Reduction, to be subtracted fro the total sound level (dB)

Example:
There is a total sound level of 35 dB(A) in a room with both supply and exhaust air systems. The supply system alone produces 32 dB(A). The difference is 3 dB(A), which means that the exhaust system produces 35 - 3 = 32 dB(A).

Basic facts

The mathematical formulation for logarithmic addition and subtraction, if there is a change of sign.

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

Later repairs

All sealing around ducts in through-connections must be checked. It is often only the visible part of the hole which is sealed, and the acoustic insulation should be measured to check the quality of the seal. The these measurements can be made at an early stage and the results can be useful guidance for the contractor.

Reference literature

Acoustics in Rooms and Buildings, Lennart Karlén, Byggtjänst.

Bahco-Compendium.

K-Konsult Compendium for Noise in Installations.

Acoustics and Noise, Johnny Andersson. Ingenjörsförlaget.

Calculation Aid

Swegon's calculation program for sound in ventilation systems - ProAc.

Acoustics - Basics

To estimate how disturbing a sound is, the frequency analysis of the sound can be compared to normalised noise curves, (so-called NR curves. The NR value is given as the highest number on the highest NR-curve, which is a tangent of the curve for frequency analysis.

A direct conversion of a dB(A) value to an NR value cannot be done. As a guideline we can say that a dB(A) value is roughly five units higher than the NR value. The difference depends completely on the sound frequency distribution.

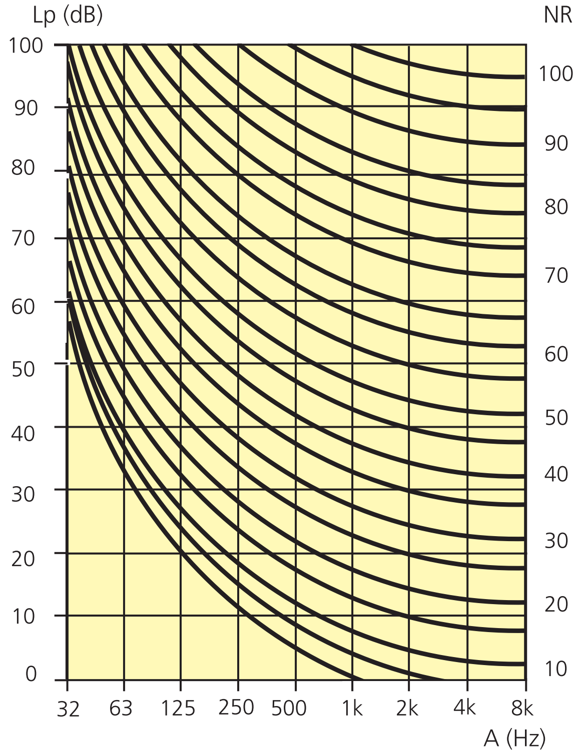


Figure 67. NR-curves.
Lp (dB) = Sound pressure level
A = Mid-frequency (Hz)

Octave band No.	Mid-frequency Hz	Band limits Hz	Wavelengths 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

Table 11. Recommended octave bands according to ISO.

Mid-frequency Octave band Hz	Filter A (dB)	Filter B (dB)	Filter C (dB)
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

Table 12. Weighting filters for sound level measurement.

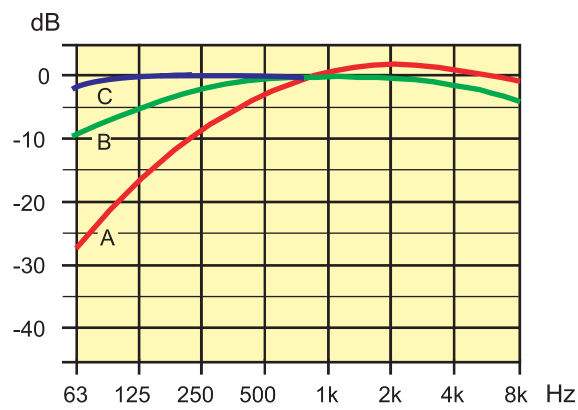


Figure 68. Attenuation for different Weighting filters.

When calculating the dB(A) value, the values measured are corrected with the A-filter approximate values according to the table. The resulting sound levels are then added logarithmically per octave band.

Example:
A sound measurement gives 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) is 41 dB(A).

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

Table 13. Correction for A-filter.

Room absorption

The volume of the room, surface properties and interior details have a considerable effect on the resultant sound level. The table with approximate values for absorption factor a, together with the graph, can be used to calculate the equivalent absorption area of a room. The room constant (A) is generally calculated using the following formula:

$$A = S \cdot \alpha_m$$

where	$S \times a_m$	$= S_1 \cdot \alpha_1 + S_2 \cdot \alpha_2 + \ldots + S_n \cdot \alpha_n$
	S	= total limiting area of the room (m^2)
	$S_1 \ldots S_n$	= partial surface area (m^2)
	$\alpha_1 \ldots \alpha_n$	= partial surface absorption factors
	α_m	= mean absorption factor for the total limiting area

Type of room	Average absorptionfactor
Radio studio, music room	0,30 - 0,45
TV-studio, department store, reading room	0,15 - 0,25
Houses, offices, hotel rooms, conference premises, theatres	0,10 - 0,15
Schools, health care facilities, small churches	0,05 - 0,10
Factory locations, indoor swimming pools, large churches	0,03 - 0,05

Table 14. Reference value for the average absorption factors of different rooms.

A	Highly damped room α_m	= 0,40
B	Damped room α_m	= 0,25
C	Normal room α_m	= 0,15
D	Hard room α_m	= 0,10
E	Very hard room α_m	= 0,05

Example:
A clothing shop with the dimensions 20 x 30 x 4.5 m ($2700\ m^3$) has an average absorption factor of $a_m = 0.40$
The equivalent room absorption is 500 m^2

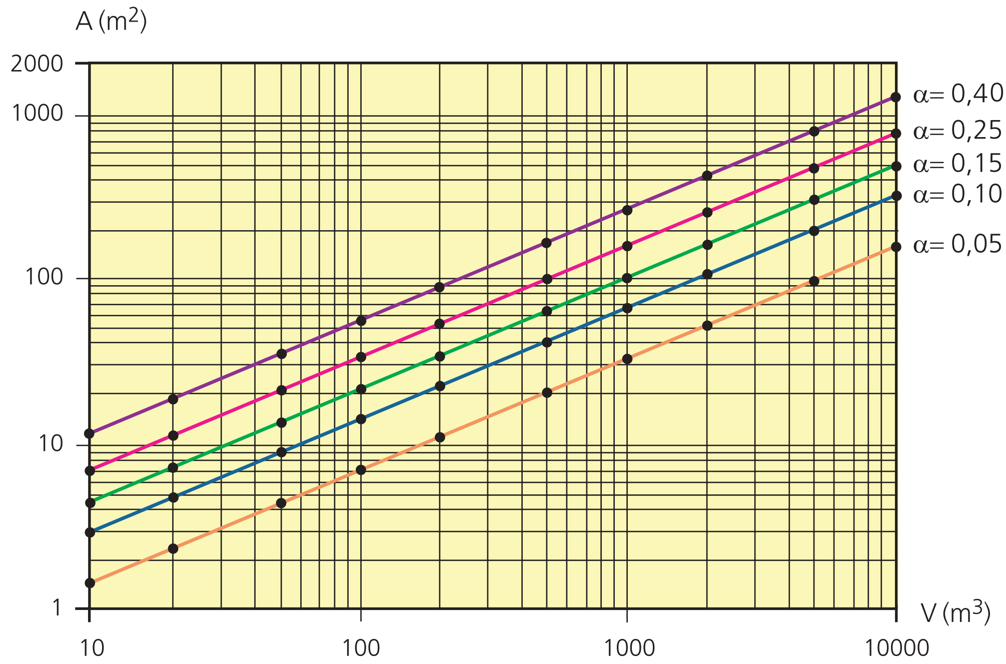


Figure 69. Equivalent sound absorption area.
A = Equivalent sound absorption area A (m^2)
V = Room volume, V (m^3)

Sound attenuation in duct opening

At an opening into the room the sound level from a duct is reduced as a result of the air spread, which normally occurs radially. This reduction is called opening attenuation.

Sound attenuation (ΔL) for each terminal is started in the octave band under "Sound data". Opening attenuation is included in these values. The graph can be used for calculating the opening attenuation of a free duct.

Example:
A rectangular duct has it opening in a room at position 3 in Figure 70 and is cross-section is 0,15 m^2. According to Figure 71 the following opening attenuation.

Octave band, Hz	63	125	250	500
Opening attenuation dB	7	4	1	0

Table 15. Opening attenuation for alt. 3 in Figure 70.

Alt. 1	Middle of room
Alt. 2	Wall or ceiling
Alt. 3	Wall, near ceiling
Alt. 4	Corner

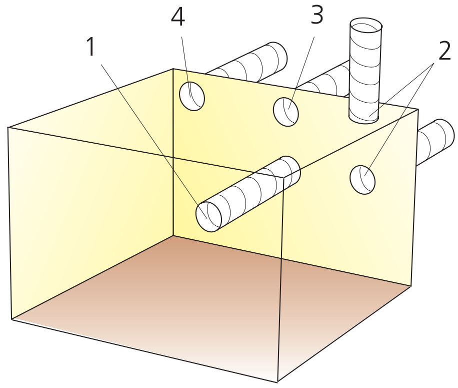


Figure 70. Location of outlet area for ducts.

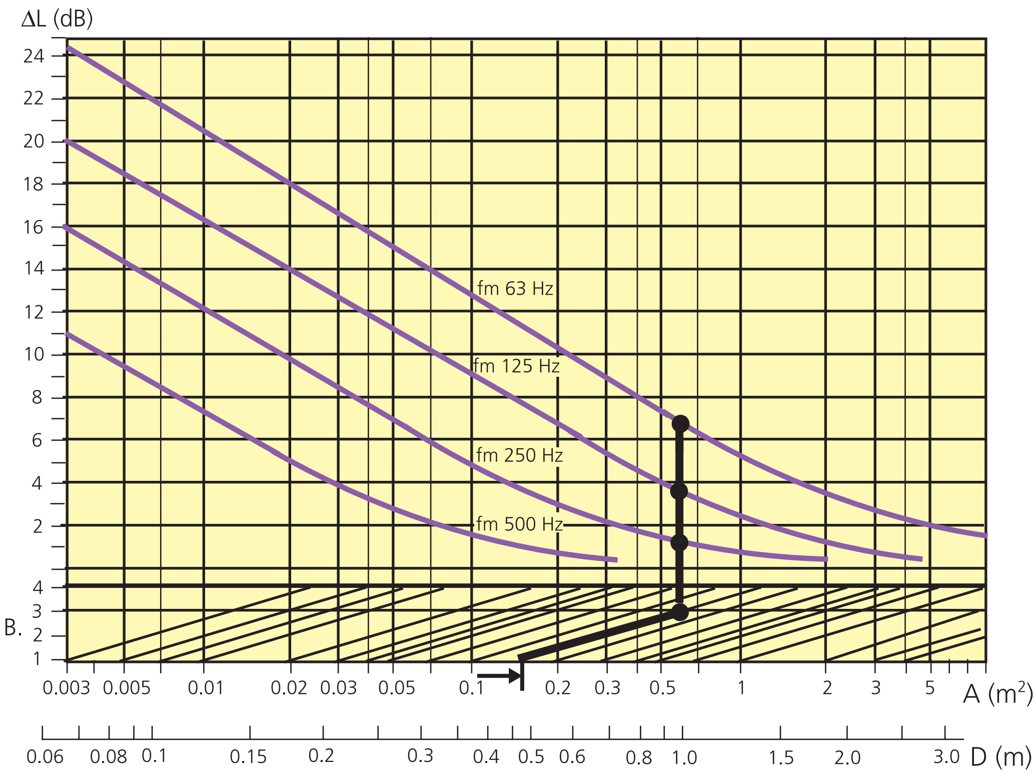


Figure 71. Attenuation.
ΔL = Attenuation (dB)
B = Position of duct opening (see Figur)
A = Cross sectional area for duct with rectangular cross section (m^2)
D = Opening diameter for duct with circular cross section (m)

Duct connection to fan outlet

The connection of the duct to the fan is the first place where excess pressure drops can occur, resulting in excessively high sound levels. When the air is deflected by a bend, consideration must be given to the velocity distribution in the duct before the bend.

An incorrect design, with a 90°-bend directly to the fan, increases the total sound power level by 4 dB. In addition, if the fan is turned "upside down" the sound power level will be 6 dB higher than with a proper layout.

Some examples of a correct and incorrect design are given below:

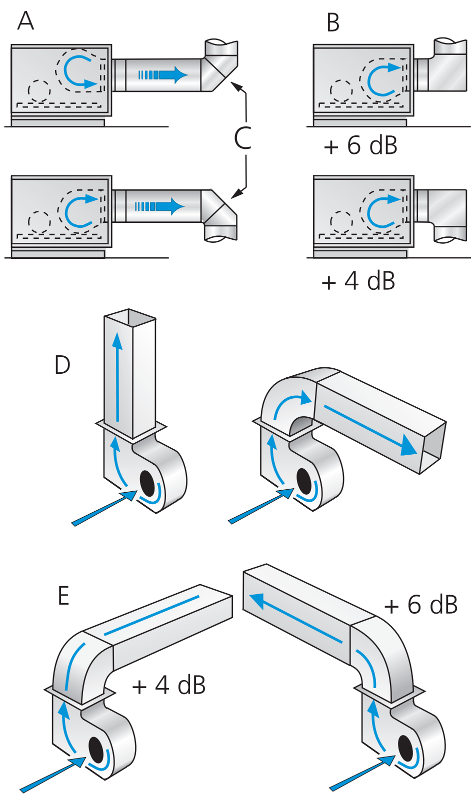


Figure 72. Examples of correct and incorrect duct connections to a fan.
A = Correct design
B = Incorrect design
C = The bend should be chamfered 45° on the outside
D = Correct design - The duct bend should be aimed in the same direction as the fan rotation
E = Incorrect design

Sound attenuator in combination with an exhaust air unit

One way of counteracting transferred sound through a common ventilation duct is to install a sound attenuator between the main duct and the exhaust air unit. By placing the attenuator as shown in the diagram below, resonance effects can be removed and acoustic insulation is considerably improved.

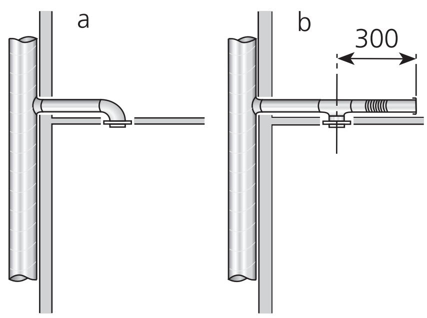


Figure 73. Counteracting sound transfer.

a)	low sound insulation	b)	high sound insulation. with duct lengths of 1-3 m especially high attenuation can be obtained.

The same procedure can be applied to supply air terminals to prevent cross-talk and to attenuate duct noise.

Choice of sound level from unit:
Selecting an air terminal should be done so that the units sound generation is 5 dB lower than the requirement for the room in question.

Distance between the duct and the terminal

Values for pressure drop and sound generation in this catalogue apply when there is an even velocity distribution in the terminal's inlet connection.

A common mistake is to place the unit too close to a branch duct, which creates sound problems.

One recommendation is that the ubnit is placed at a distance of at least 3 times the duct diameter from the branch. See example in Figure 74.

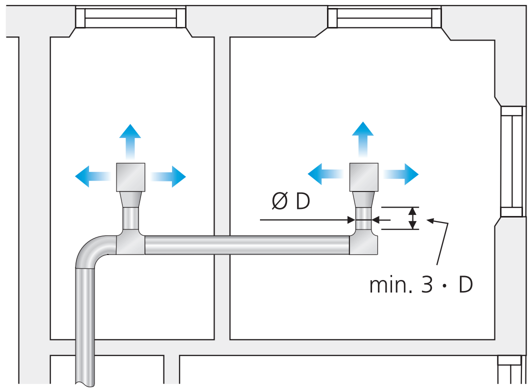


Figure 74. The branch length for the duct should be at least 3 · D.

Duct design for displacement air terminals

The structure of the duct greatly affects sound generation. Bends directly before the terminal can cause considerable increases in sound. See examples in the Figure 75.

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

Table 16. Sound increases (dB) for different duct connections and equivalent sound absorption areas.

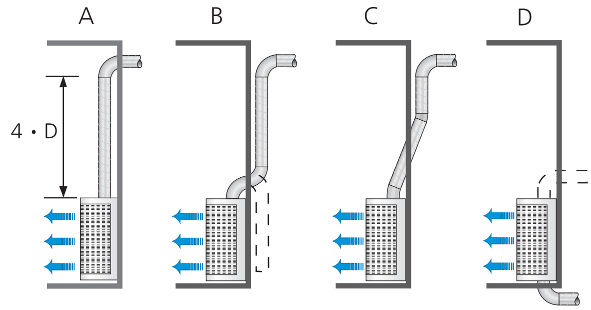


Figure 75. Examples of different connections affect sound generation.

Method for converting dB(A) to dB(C)

The model applied to a PELICAN CSa 315 + ALSc 250-315 diffuser that, at an air flow of 160 l/s and with an open damper, produces 30 dB(A) according to the design graph.

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	5	9	6	6	2	-6	-11	-13	
L	35	39	36	36	32	24	19	17	
Plus C-filter	-0.8	-0.2	0	0	0	-0.2	-0.8	-3	
Result, dB(C)	34.2	38.8	36	36	32	23.8	18.2	14	44
	Room attenuation at 10 m² Sabine								4
	Result dB(C)								40

Result: At 170 l/s with open damper the PELICAN CS a 315, produces 40 dB(C).