An external rotor motor is essentially constructed like a normal non-synchronous motor, with one difference: the stator and the rotor have swapped places. The stator with its windings is at the centre of the motor, while the rotor is located in the casing itself. The motor shaft (connected to the rotor) turns on sealed ball bearings inside the stator, and the impeller or fan blades are fitted to the rotor casing. With this design, the motor and fan form a compact unit at the centre of the air stream.

Because the external rotor motor's unique construction allows it to be cooled by the transported air, the motor speed can be controlled by voltage regulation.

Casing

Most of the fans have an outer casing made of hot-rolled galvanised sheet steel complying with EN 10 142/10 147. The sheet steel has a layer of 20 µm zinc which provides excellent protection against corrosion. The galvanised sheet-metal parts are either spot-welded, screwed or riveted together.

Fans with powder-coated surfaces are well protected against corrosion. The powder coating is at least 40 µm thick and produces a hard and impact-resistant surface. To avoid environmental pollution, no solvents are used at the Systemair powder-coating plant.

Insulation

The material used in our insulated fans is water-repellent, non-capillary mineral wool whose stability is unaffected by steam and moisture. The insulation is classified as non-combustible material which tolerates 200°C.

Motors and impellers

The direction of rotation for three-phase fans is indicated by an arrow on the motor housing. Fans with forward-curved impellers are manufactured from galvanised sheet steel.

Backward-curved impellers have polyamid or galvanised steel plate blades. These blades are mounted on a galvanised steel plate. The impellers are press-fitted directly onto the rotor of the external rotor motor. The motor (complete with impeller) is balanced dynamically in two planes in accordance with DIN ISO 1940.

Bearings

The motor's ball bearings are completely maintenance-free and can be used in any installation position at the maximum indicated temperature for transported air. At a 40°C ambient temperature for transported air, the life expectancy of the bearings is at least 40.000 hours (L10). NB! Low ambient temperature is not a problem for the motor ball bearings if the fan is operating. The reason is the 60- to 90 K temperature increase inside the motor during operation.

Motor protection

Most fans have an integral thermal protection relay which provides the motors with better protection against overheating than an over-current protection relay. This is especially important if the fan is speed-controlled by means of voltage reduction, as it is then impossible to stipulate the precise over-current.

The thermal contacts are fitted in the motor winding. It will open and disconnect the power supply to the fan when the critical temperature is reached. This is 130°C for motors with insulation class B and 155°C for motors with insulation class F.

Integral thermal contact

Fans with integral thermal contacts are reset either automatically or manually by switching off the current and then wait for up to an hour before the fan can be started again.

External leads from thermal contact

Fans with external leads from the thermal contact are supplied with two leads connected to the integral thermal contact (marked TK in wiring diagrams). These leads must always be connected to a motor-protection relay. The S-ET 10 is suitable for single-phase fans (or the AWE-SK if the current is below 0.45 A) and the STDT-16 is suitable for three-phase fans. If the thermal contact has opened, the protection relay must be reset manually.

Thermal contacts that can be reset electrically

If a fan is fitted with a thermal contact that can be reset electrically, one must first switch off the current and then wait for up to an hour before the fan can be started again. KVKF and small KD fans are among those models which require electrical resetting.

Rating

Rated voltage/ Frequency

Maximum permitted voltage variation: +6%, -10% in accordance with DIN IEC 38, plus maximum permitted frequency.

Power rating

Maximum power used by the fan from the mains supply.

Rated current

Rated current means the maximum current used by the fan from the mains supply at nominal mains voltage. When the fan speed is controlled by lowering the voltage, the current in the motor may exceed the specified rated current when the voltage is low. (The recommended speed controllers are designed with this in mind.) The increased current in the motor requires a reduction of the maximum permitted temperature for transported air. In the technical tables, the highest permitted temperature for transported air is shown for both the rated current and for speed control.

Airflow

The air flow is shown for free-blowing conditions (at zero back pressure). Air flow is measured in accordance with DIN 24 163 and BSA BS 848. Assumed air density is 1.2 kg/m^3 at 20°C.

Pressure

The static pressure is shown in the fan diagrams as ps (Pa).

R.p.m.

The tables show the fan's nominal r.p.m. at the rated current.

Capacitor

A capacitor is connected to the single-phase motors. The relevant capacitance is shown in the table for each fan.

Sound pressure – and sound power level

The sound pressure level emitted by duct fans to the surroundings is measured while operating at optimal efficiency in a 20 m^2 equivalent room absorption area (Sabine) at a distance of 3 m.

The sound pressure level emitted by roof fans to the surroundings is measured while operating at optimal efficiency in a free field and at a distance of 4/10 m.

	Duct fan	Roof fan
Room volume	80 m^3	Free field
Room's equiv. absorption area	20 m^2	–
Distance from fan (r)	3 m	4/10 m
Direction factor (Q)	1	1
Difference between sound power (L_W) and sound pressure (L_p)	-7 dB	23/-31 dB

The relationship between the sound pressure level and sound power level is described in the Theory Section on page 509.

General technical information: fans

Adjusted sound values
In this catalogue, all the sound values for fans (both sound power levels and sound pressure levels) have been adjusted to the ear's sensitivity with an A filter.
The sound power levels shown in the diagram are measured at the fan's inlet. Octave band apportioning of the sound pressure level is made at the fan's maximum operating efficiency. The tables show the inlet, outlet and ambient sound.

Speed control

Choosing a speed control method
Both economical and technical aspects should be taken into consideration when selecting speed control. When assessing the most economical option, both the investment cost and the operating cost should be included in the calculations. The most important technical aspects that need to be considered are noise and life-expectancy.
Most of the electrical means for varying a motor's speed cause some degree of noise in the motor, with the exception of transformer-controlled speed. Power dissipation increases when running at lower speeds. This dissipation is transformed into heat in the motor. If the power dissipation is substantial, the operating temperature for the bearings will alter significantly, which will reduce their life-expectancy.
Suitable operating conditions and characteristics of the different speed control methods:

Transformers
No increased motor noise when the speed is regulated. Life-expectancy of the motor bearings can be shortened when operating at low voltages for long periods (voltage steps 1 and 2). Suitable range for speed control: steps 1-5. Several fans can be run via the same transformer without special procedures.
The five curves in the fan diagram show the different voltage outputs from our transformers.

Step (curve)	1	2	3	4	5
Voltage, 1~	80	105	130	160	230
Voltage, 3~	95	145	190	240	400

Single-phase stepless speed control
Can cause noise problems when reducing speed. Should be avoided in noise-sensitive installations. The life-expectancy of the motor bearings will be reduced by operating at lower voltages. Suitable range of adjustment: 60-100% of the rated voltage. Using the same speed control to run several fans increases the levels of noise and electromagnetic interference. Shielded motor cables are recommended for installations with several fans connected to one speed control unit.

Three-phase speed control
There are normally no noise problems associated with speed-controlled operation. The life-expectancy of the motor bearings will be somewhat reduced by operating at lower voltages. Suitable range for speed control: 40-100% of rated voltage. Suitable when using one speed-control unit for several fans. In order to minimise noise and electromagnetic interference, we recommend sound filters and also the use of shielded motor cables when several fans are connected to one speed-control unit.

Explosion-proof fans

The owner of the property and the installation engineer are responsible for ensuring that all equipment that is installed in explosive areas is approved by a recognised testing laboratory and installed correctly. Fans must be installed and protected so that no foreign object can come into contact with the impeller or cause hazardous sparking. Both the motor-protection relay and the transformer must be positioned outside the risk area.

EX series
These fans are fitted with specially-made EX motors. Single-phase fans use a special EX-approved motor capacitor encased in sand which complies with the requirements for Fire Class T5.
The fans casings are cast in silumin alloy, and the impeller is made of aluminium. The certificate of compliance refers to explosion-proof versions in accordance with EN 50014, EN 50017, EN 50019, EN 1127-1 and EN13463-1. Improved safety versions comply with EEx e II T3.
This series must always be connected to an over-current relay which protects the motor against overheating or short-circuiting (for instance with a seized rotor). The motor protection must break the circuit within 15 seconds of a short circuit. The current must be disconnected definitively. This means that the motor-protection relay must require manual resetting. Fans in the EX series are not speed-controllable.

The DKEX and KTEX series
These fans are supplied as 400 V three-phase models. Permissible ambient temperature range: from -20°C to +40°C. The fan casing and impeller are manufactured in galvanised steel plate and the inlet cone is made of copper. The certificate of compliance refers to explosion-proof versions in accordance with EN 50014, EN 50019, EN 1127-1 and EN13463-1. Improved safety versions comply with EEx e II T3.
The fans are fitted with specially-made external rotor motors which allow their speeds to be adjusted from 100% to 15% by lowering the voltage. These motors must be connected to the U-EK230E thermistor motor protection unit.

Step (curve)	1	2	3	4	5
Voltage, 3~	90	140	180	230	400

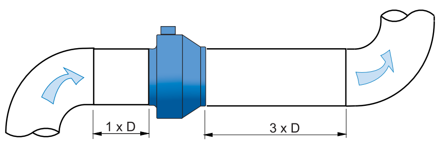
The fan motors have six series-connected thermistors (two per phase winding) whose resistance is determined by the motor temperature. When the motor temperature exceeds the permitted limit, the resistance rises sharply and the connected motor protector is triggered to break the circuit.

DVEX series
These fans can be speed-controlled from 100% to 15% by lowering the voltage. These motors must be connected to the U-EK230E thermistor motor protection unit.
The fan motors have six series-connected thermistors (two per phase winding) whose resistance is determined by the motor temperature. When the motor temperature exceeds the permitted limit, the resistance rises sharply and the connected motor protector is triggered to break the circuit.
DVEX fans are supplied as 400 V three-phase models. Permissible ambient temperature range: from -20°C to +40°C. The casing is manufactured in galvanised steel plate and the impeller is made of aluminium. The inlet cone is made of copper. The certificate of compliance refers to explosion-proof versions in accordance with EN 50014, EN 50019, EN1127-1 and EN 13463-1. Improved safety versions comply with EEx e II T3.

Installation

All fans can be installed in any position, but roof fans should be installed horizontally. Smaller roof fans can be installed on the roof pitch. To avoid the transfer of vibrations to the duct system, we recommend that fans are installed with mounting clips or flexible sleeve couplings. All fans are designed for continuous operation.
Fitting a straight duct or silencer onto the inlet and outlet of the fan will help to prevent pressure-drop and system efficiency losses caused by turbulent air flow. The straight section must have no filter or similar, and its length must be at least 1 x the duct diamater on the fan's inlet side and at least 3 x the duct diameter on the fan's outlet side. (See figure 1.)

Figure 1. Correctly installed duct fan.



For a rectangular duct, the duct diameter is calculated as:

$$D = \sqrt{\frac{4 \cdot H \cdot B}{\pi}}$$

D = duct diameter
H = duct height
B = duct widht

Guarantee
The guarantee period is specified in the relevant terms and conditions for delivery. The guarantee is only valid when the thermal contact motor protector and transformer are correctly installed.

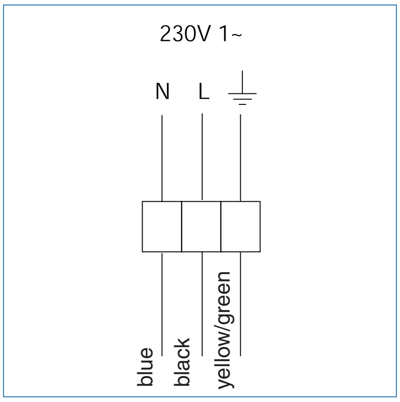
Electrical connection

Fan type	Diagram
AR/AW 200E2-K to 450E4-K	5
AR/AW 315D4-2K to 450D4-K	16
AR/AW 630E6, 710E6	6
AR/AW 450E4 to 560E4	6
AR/AW 450D4 to 710D4	18
AR/AW 630D6 to 1000D6	18
AR/AW 1000D8	18
AW 355D4EX, 420D4EX	19
AW 550D6EX to 650D6EX	19
CE (other models)	6
CE 200	5
CKS single phase	6
CKS three phase	8
CT (other models)	8
CT 200	7
DKEX	11
DVC-S 225	22
DVC-S 315-400	23
DVC-P 225-400	14
DVC-P 450K	28
DVC-S 450K	24
DVC-S 450-630 3~	26
DVC-P 450-630 3~	25
DVEX	11
DVN/DVNI 355DV to 630DS	17
DVN/DVNI 355E4, 400E4	21
DVN/DVNI 630D4 to 900D8	17
DVN/DVNI 710D6	17
DVN/DVNI 800D6-900D6	13b
DVS/DHS/DVSI 190EZ, 225EZ, EV	20
DVS/DHS/DVSI 310ES, 311ES	20
DVS/DHS/DVSI 310EV, 311EV	20
DVS/DHS/DVSI 355DV, 450DV	16
DVS/DHS/DVSI 355E4, 400E4	5
DVS/DHS/DVSI 400DS to 710DS	18
DVS/DHS/DVSI 400DV to 560DV	18
DVS/DHS/DVSI 400E6 to 500E6	6
DVS/DHS/DVSI 450E4	6
DVV 1000 D4-8-P	14
DVV 1000D4-6-P, D6-8, D8-12	15
DVV 1000D6, D8, D4-P, D6-P	13
DVV 400D4 to 630D4, 400D6 to 630D6	13
DVV 400D4-6 to 560D4-6	15
DVV 450D4-8	14
DVV 630D4-6-K, D6-8-K, D4-6, D6-8	15
DVV 630D4-K, 630D4-K	15
DVV 800D4-6-K, D4-6-P, D6-8	15
DVV 800D4-K, D4-M, D4-P, D6-K, D8-K, D6, D8	13
DVV 800D6-12-K	14
DVV-EX 560D4 to 1000D8	13b
DVV-EX 560D4-6 to 800D6-8	15b
EX 140-2	10
EX 140-2C	9
EX 140-4	10
EX 140-4C	9
EX 180-4	10
EX 180-4C	9
K/KV 100M & 125M	1
K/KV 100XL, K125XL to 315L	2
KBR-F 280D2-355DZ-K	17
KBR-F 280D2-4	17

Fan type	Diagram
KBR 315DV, 355DV	17
KBR 315DZ, 355DZ	17
KBR 355DV/K, 355DZ/K	17
KBR 355E4/K, 355E4	21
KBT 160DV to 280DV	17
KBT 160E4 to 250E4	21
KD 200L to 355S	2
KD single phase (other models)	6
KD three phase	8
KDRD 50 to 70	8
KDRE 45 to 65	6
KE (other models)	6
KE 40-20	5
KT (other models)	8
KT 40-20	7
KTEX	11
KVK 125-400	5
KVK 500	3
KVKE	4
KVKF 125-250L	2
KVKF 315M/L	12
KVKF 355-400	6
MUB 025 355DV-A2	16
MUB 025 355E4-A2	5
MUB 042 400E4-A2	6
MUB 042 450DS-A2	18
MUB 042 400DV-A2, 499DV-A2	18
MUB 042 499E4-A2-500E4-A2	6
MUB 042 400DV-K2 to 500DV-K2	17
MUB 042 500DS-A2 to 630DS-A2	17
MUB 042 500DV-A2, 560DV-A2	17
MUB 062 560DV-K2	17
MUB 062 630D4-A2	13
MUB 062 630D4-K2	13
MUB 062 630DV-B2	18
MUB 100 630D4-L	13
MUB 100 710D6-A2	13
MUB 025 315EC-A2 to 400EC-A2	23
MUB 042 450EC-A2-K	24
MUB 042 450EC-A2 to 630EC-A2	26
RS 30-15 to 50-25	2
RS/RSI single phase (other models)	6
RS/RSI three phase 60-35 to 100-50	8
RVF 100M	1
RVF 100XL	2
RVK 100 E2-A1, 125 E2-A1	1
RVK 125 E2L1 to 315E2-L1	2
RVK 315Y4-A1	19
TFSR	2
TFSK	28
TOE	6
TOV	8
WVA/WVI 400D4 to 630D4	13
WVA/WVI 400D4-6 to 630D4-6	15
WVA/WVI 400D6-8 to 1000D6-8	15
WVA/WVI 630D4-6-K	15
WVA/WVI 630D4-8, 630D6-12, 1000D6-12	14
WVA/WVI 630D4-8-K, 630D6-12-K	14
WVA/WVI 630D4-K	13
WVA/WVI 800D6, 800D6-K, 1000D6	13
WVA/WVI 800D6-12, 800D6-12-K	14
WVA/WVI 800D6-8-K	15

General technical information: fans

1



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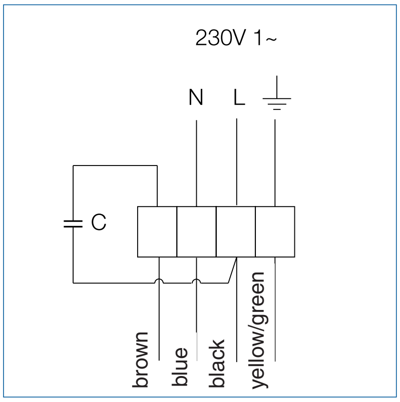

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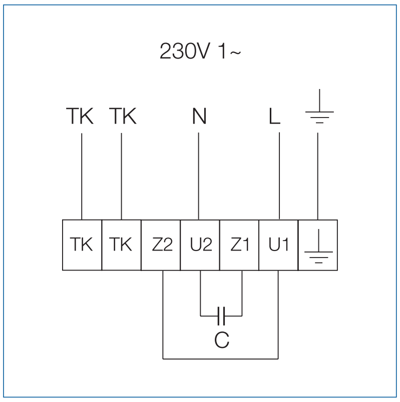

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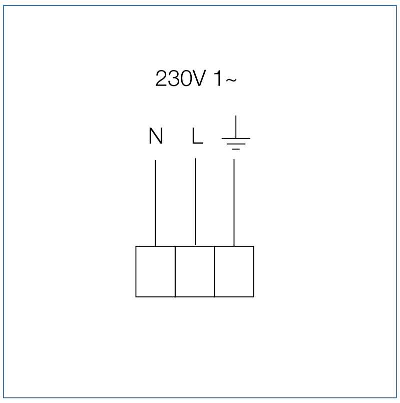

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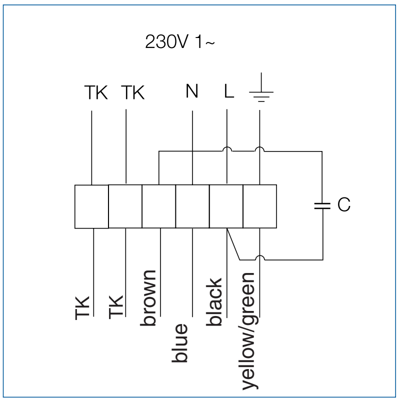

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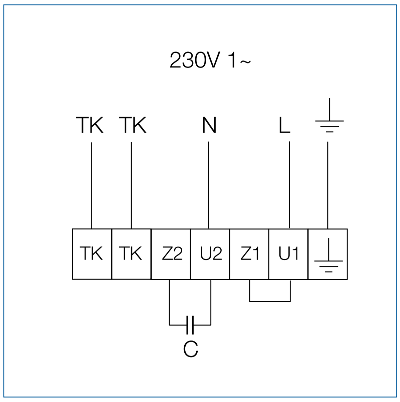

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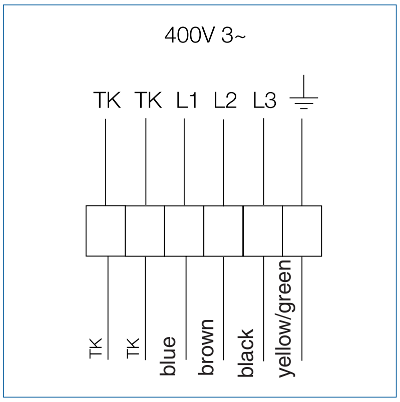

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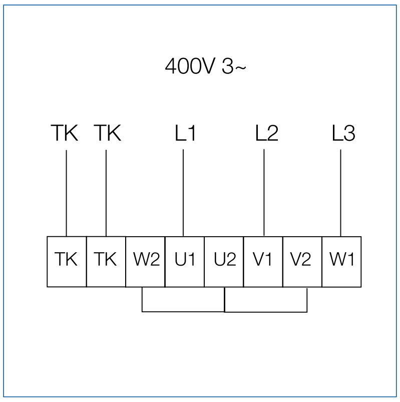

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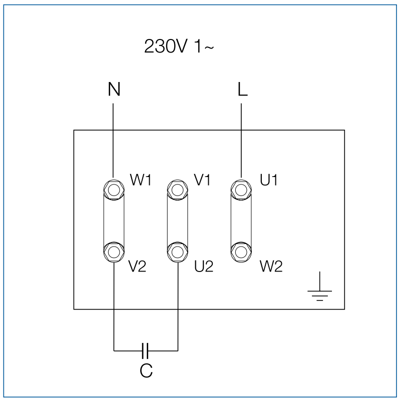

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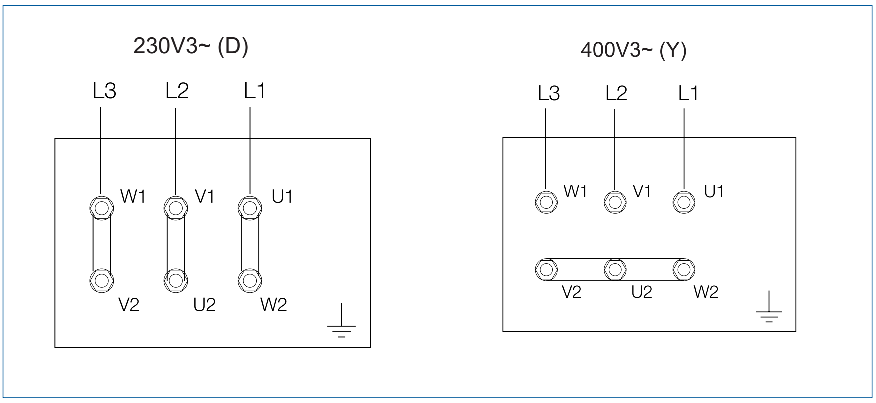

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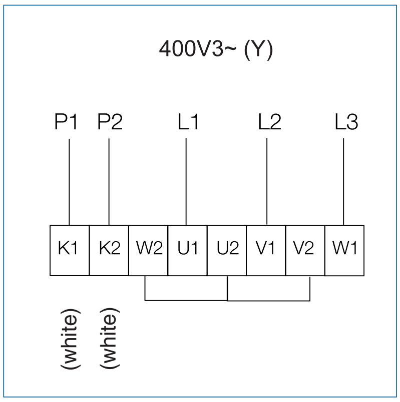

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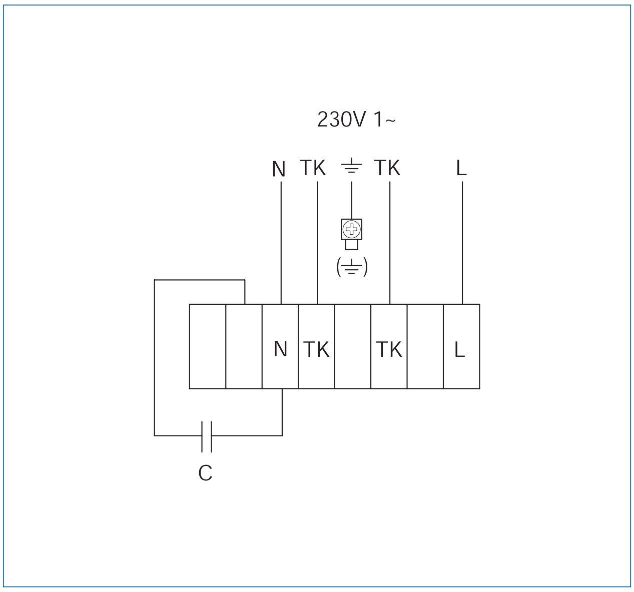

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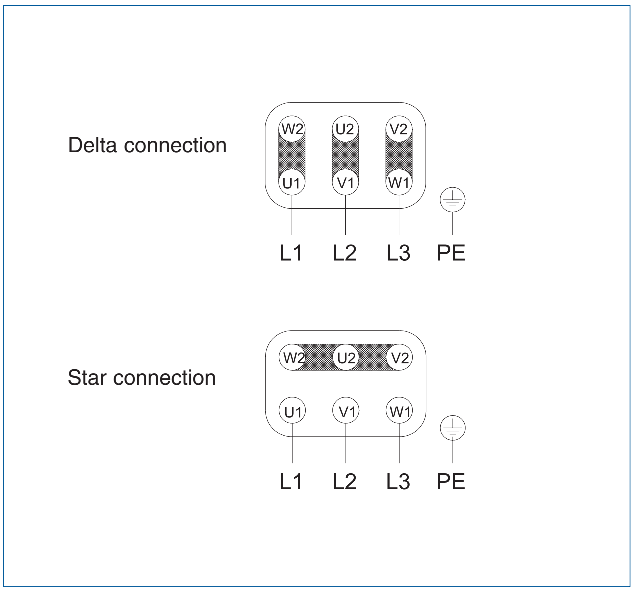

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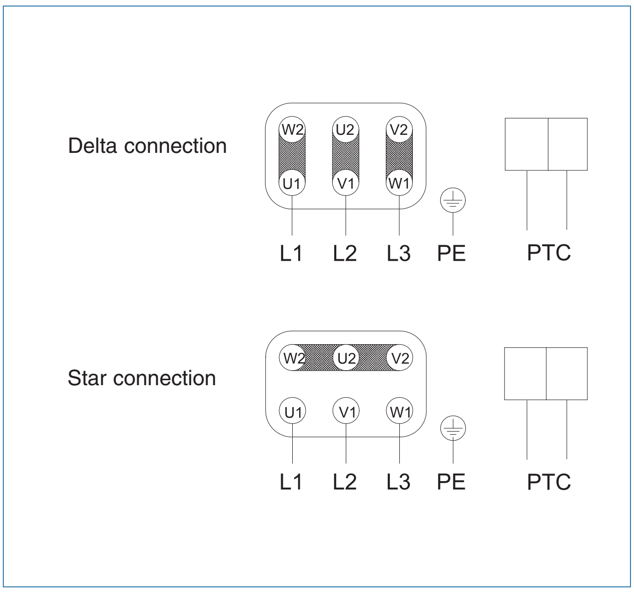

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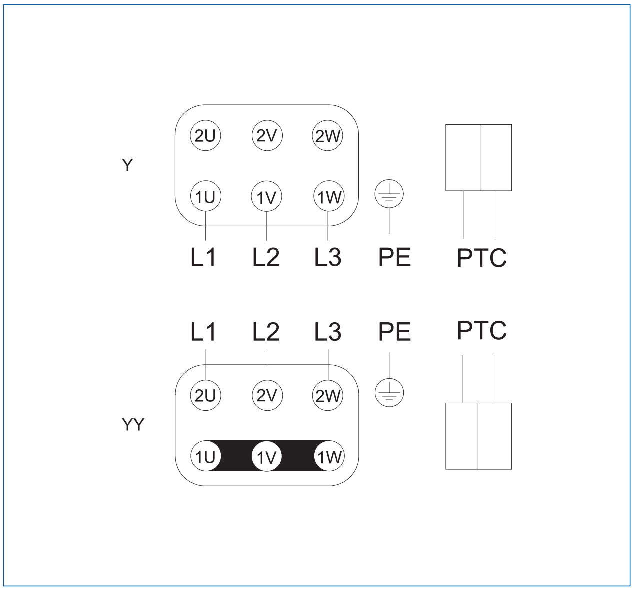

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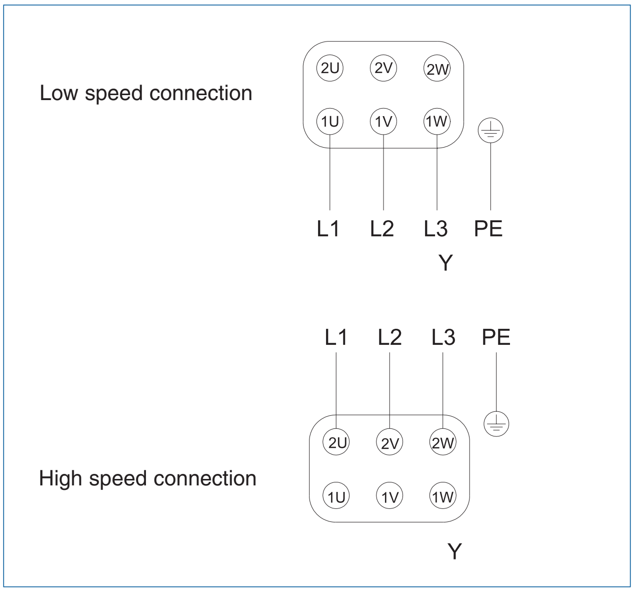

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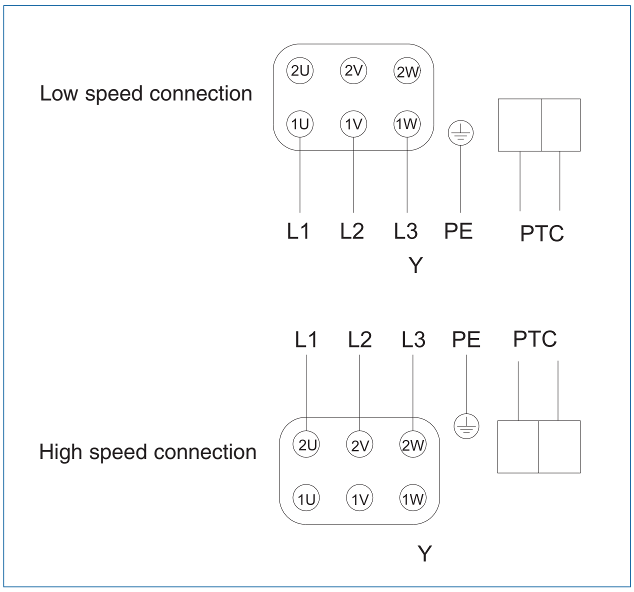

General technical information: fans

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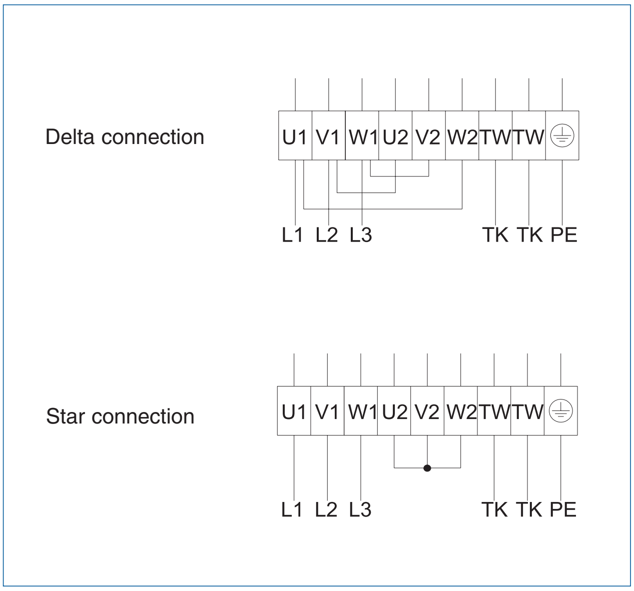

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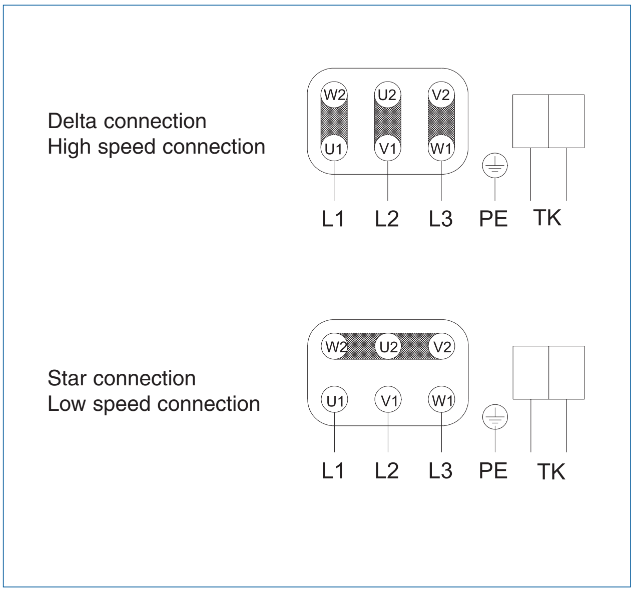

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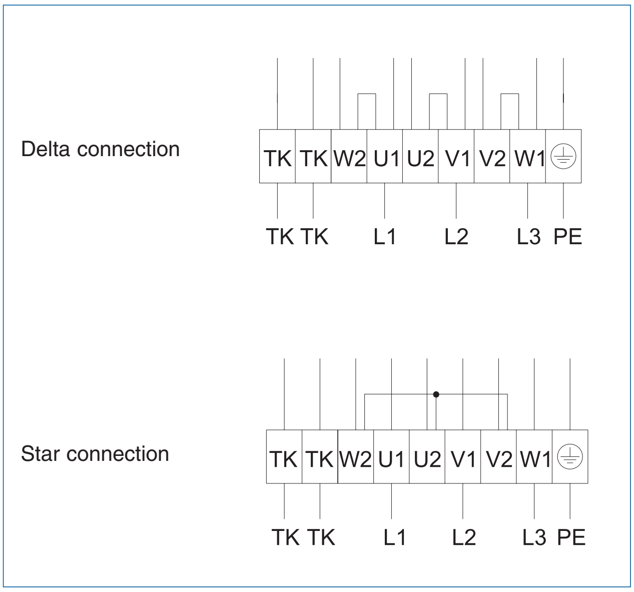

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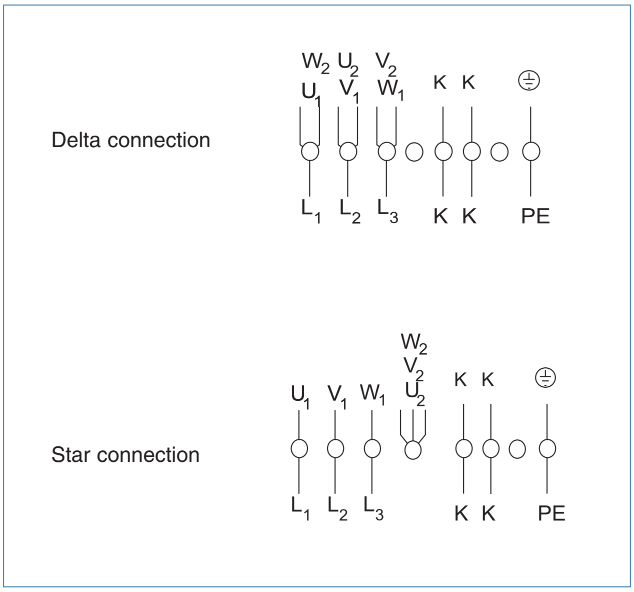

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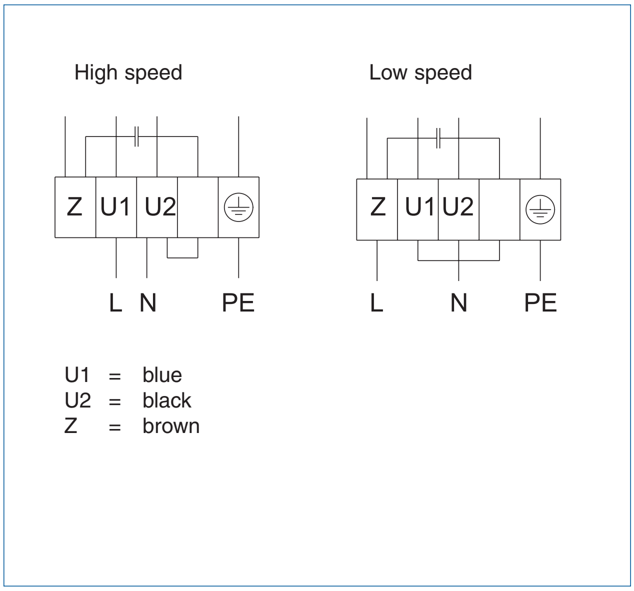

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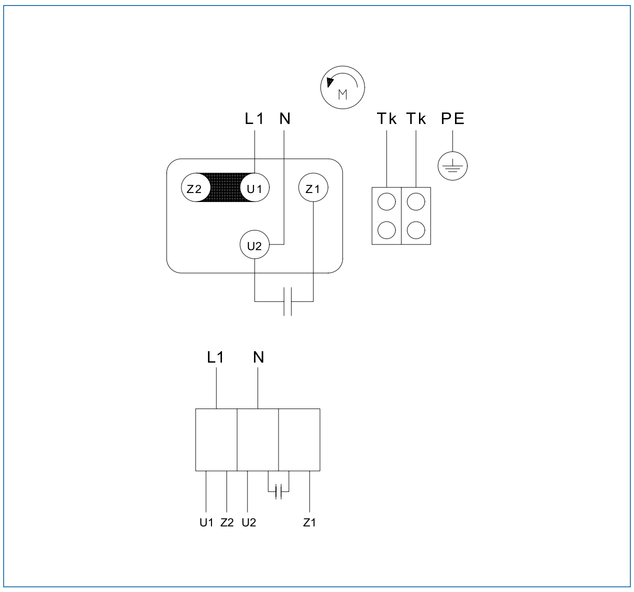

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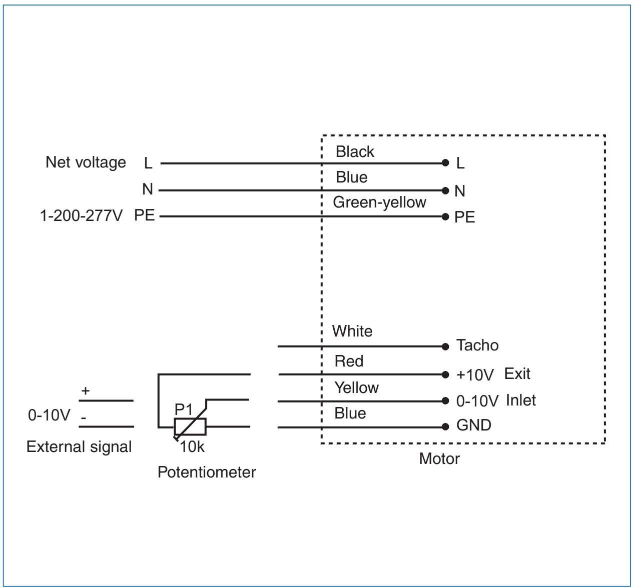
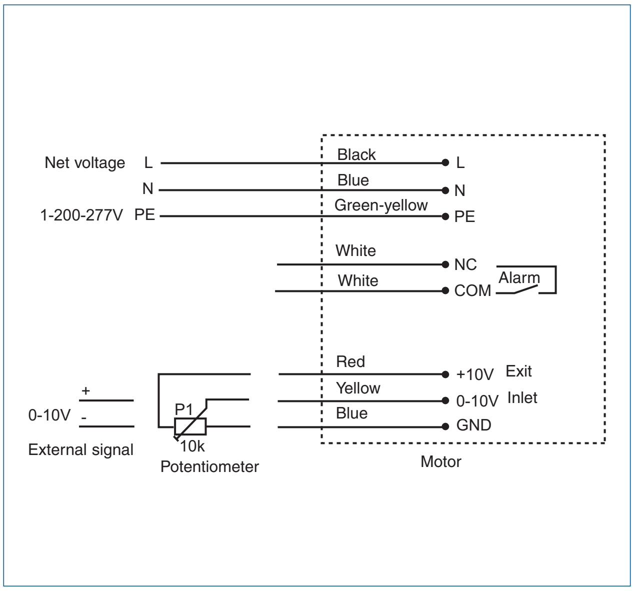
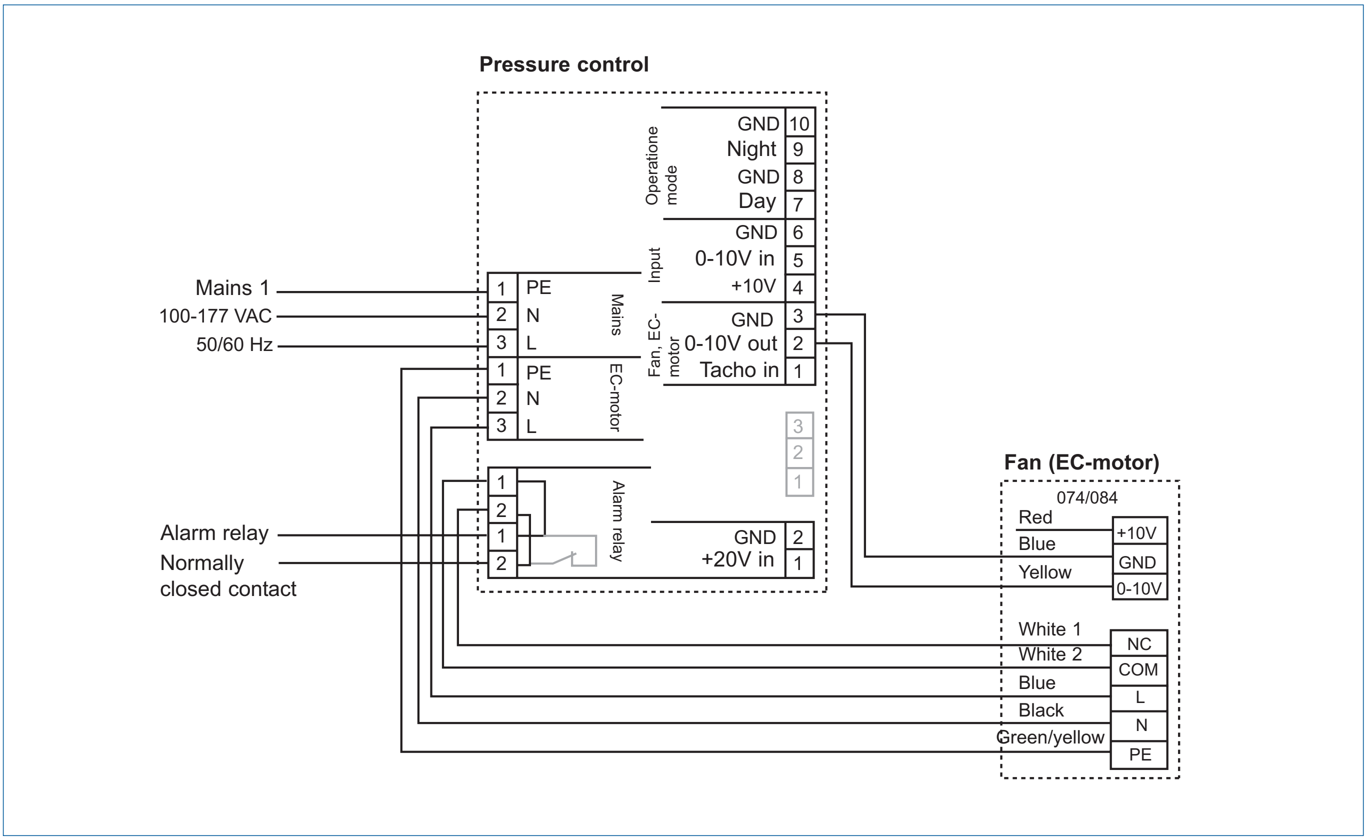

Net voltage L
N
1-200-277V PE
Black
Blue
Green-yellow
L
N
PE
White
Red
Yellow
Blue
Tacho
+10V Exit
0-10V Inlet
GND
0-10V
+
-
External signal
P1
10k
Potentiometer
Motor

23


Net voltage L
N
1-200-277V PE
Black
Blue
Green-yellow
L
N
PE
White
White
NC
COM
Alarm
Red
Yellow
Blue
+10V Exit
0-10V Inlet
GND
0-10V
+
-
External signal
P1
10k
Potentiometer
Motor

24


Pressure control
Operatione mode
GND 10
Night 9
GND 8
Day 7
Input
GND 6
0-10V in 5
+10V 4
Fan, EC-motor
GND 3
0-10V out 2
Tacho in 1
Mains 1
100-177 VAC
50/60 Hz
1 PE
2 N
3 L
Mains
1 PE
2 N
3 L
EC-motor
3
2
1
Alarm relay
Normally
closed contact
1
2
1
2
Alarm relay
GND 2
+20V in 1
Fan (EC-motor)
074/084
Red
Blue
Yellow
+10V
GND
0-10V
White 1
White 2
Blue
Black
Green/yellow
NC
COM
L
N
PE

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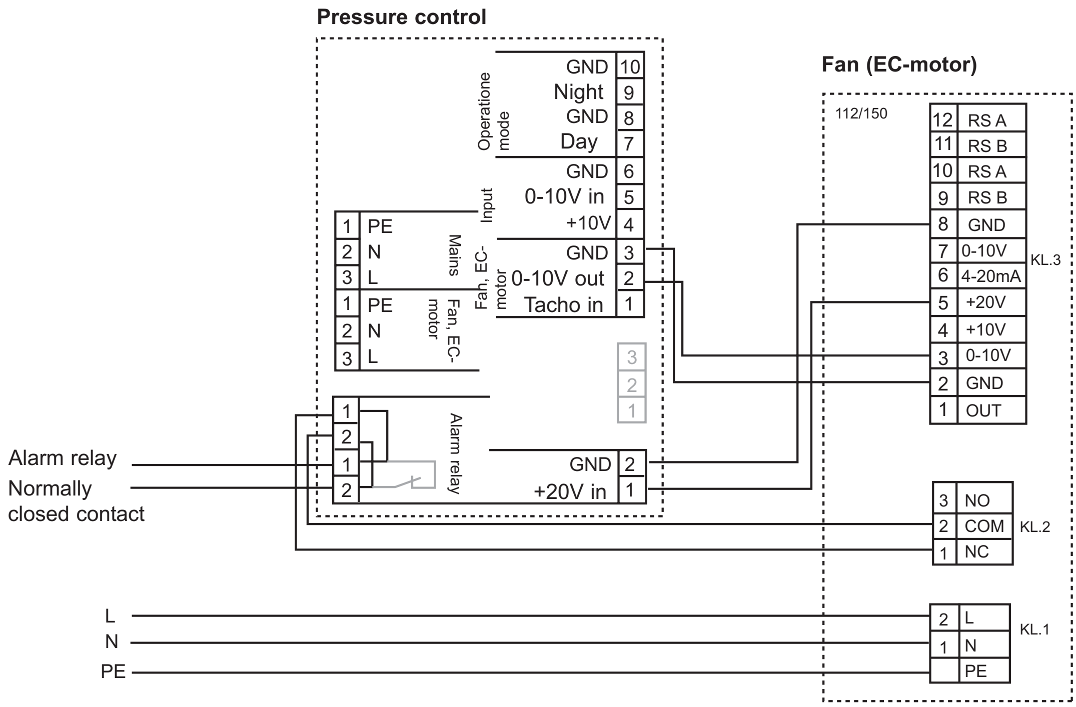

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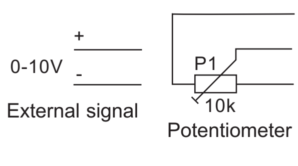

Kl. 3 / class 3	12	RS A	Interface RS485 for ebmBUS
	11	RS B	Interface RS485 for ebmBUS
	10	RS A	Interface RS485 for ebmBUS
	9	RS B	Interface RS485 for ebmBUS
	8	GND	GND
	7	0-10V	Control- / Actual value inlet
	6	4-20mA	Control- / Actual value inlet
	5	+20V	Supply ext. sensor 50mA
	4	+10V	Supply ext. potentiometer 10mA
	3	0-10V	Control- / Actual value inlet
	2	GND	GND
	1	OUT	Master exit 0-10V max. 3mA
Kl. 2 / class 2	3	NO	Error signal relay, closer in the case of error
	2	COM	Error signal relay, COMMON
	1	NC	Error signal relay, opener in the case of error
Kl. 1 / class 1	3	L1	Net, L1
	2	L2	Net, L2
	1	L3	Net, L3
		PE	Protective conductor

Art.-Nr.: 305332

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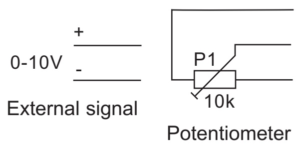
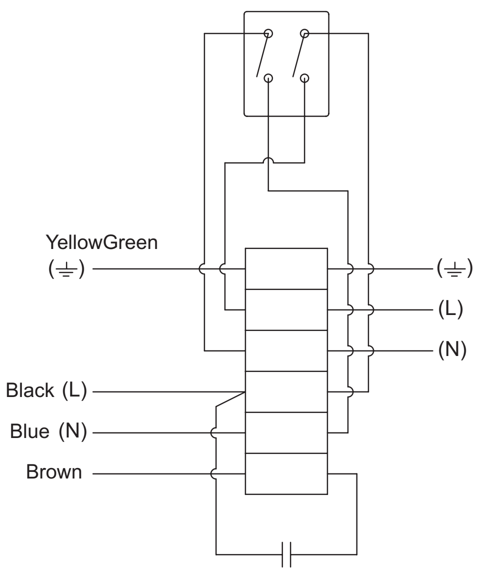


Kl. 3 / class 3	12	RS A	Interface RS485 for ebmBUS
	11	RS B	Interface RS485 for ebmBUS
	10	RS A	Interface RS485 for ebmBUS
	9	RS B	Interface RS485 for ebmBUS
	8	GND	GND
	7	0-10V	Control- / Actual value inlet
	6	4-20mA	Control- / Actual value inlet
	5	+20V	Supply ext. sensor 50mA
	4	+10V	Supply ext. potentiometer 10mA
	3	0-10V	Control- / Actual value inlet
	2	GND	GND
	1	OUT	Master exit 0-10V max. 3mA
Kl. 2 / class 2	3	NO	Error signal relay, closer in the case of error
	2	COM	Error signal relay, COMMON
	1	NC	Error signal relay, opener in the case of error
Kl. 1 / class 1	2	L	Net, L
	1	N	Net, N
		PE	Protective conductor

Art.-Nr.: 305386

28



THEORY SECTION

The intention of this Theory Section is to explain the basic principles of acoustics and ventilation.

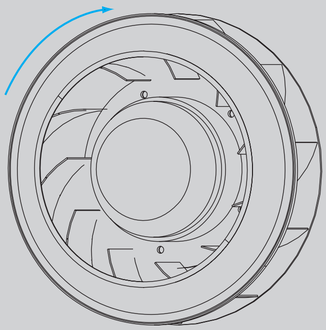
The theory section concludes with a description of the parts which are integral to a ventilation unit or an air-handling unit, i.e. fans, heaters, heat exchangers and filters.

Explanatory texts and further information are provided in the margin. Some diagrams and formula also feature in the margins, together with examples of their application.

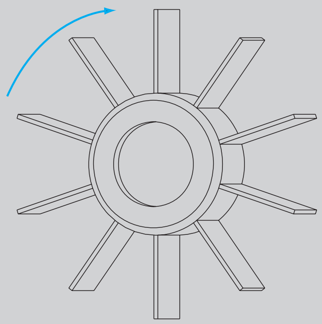
Contents

Fans . . . . .page 500
Heat recovery units, heaters and filters . . . . .page 506
Acoustics . . . . .page 509
Air terminal devices . . . . .page 515

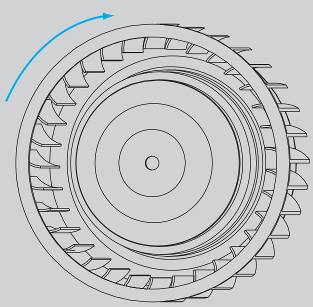
Blade profiles for radial fans
The arrow indicates the impeller's direction of rotation.



Backward curved



Straight radial



Forward curved

Fans

Fans are used in ventilating units to transport the air from various air intakes through the duct system to the room which is to be ventilated. Every fan must overcome the resistance created by having to force the air through ducts, bends and other ventilation equipment. This resistance causes a fall in pressure, and the size of this fall is a decisive factor when choosing the dimensions of each individual fan.

Fans can be divided into a number of main groups determined by the impeller's shape and its operating principle: radial fans, axial fans, semi-axial fans and cross-flow fans.

Radial fan

Radial fans are used when a high total pressure is required. The particular characteristics of a radial fan are essentially determined by the shape of the impeller and blades.

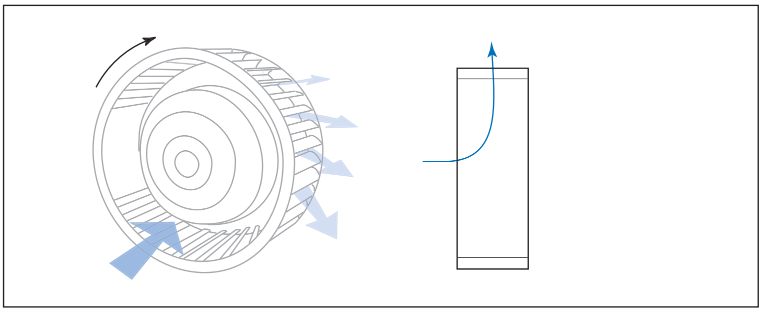


Figure 24: The air stream through a radial fan with forward-curved blades

Backward-curved blades (B impeller): The air volume which can be delivered by backward-curved blades varies considerably according to the pressure conditions. The blade form makes it less suitable for contaminated air. This type of fan is most efficient in a narrow range to the far left of the fan diagram. Up to 80% efficiency is achievable while keeping the fan's sound levels low.

Backward-angled straight blades (P impeller): Fans with this blade shape are well suited for contaminated air. Up to 70% efficiency can be achieved.

Straight radial blades (R impeller): The blade shape prevents contaminants from sticking to the impeller even more effectively than with the P impeller. No more than 55% efficiency can be achieved with this type of fan.

Forward-curved blades (F impeller): The air volume delivered by radial fans with forward-curved blades is affected very little by changes in air pressure. The impeller is smaller than the B impeller, for example, and the fan unit consequently requires less space. Compared with the B impeller, this type of fan's optimal efficiency is further to the right on the diagram. This means that one can select a fan with smaller dimensions by choosing a radial fan with an F impeller rather than a B impeller. An efficiency of approximately 60% can be achieved.

Axial fan

The simplest type of axial fan is a propeller fan. A freely-rotating axial fan of this type has a very poor efficiency rating, so most axial fans are built into a cylindrical housing. Efficiency can also be increased by fitting directional vanes immediately behind the impeller to direct the air more accurately. The efficiency rating in a cylindrical housing can be 75% without directional vanes and up to 85% with them.

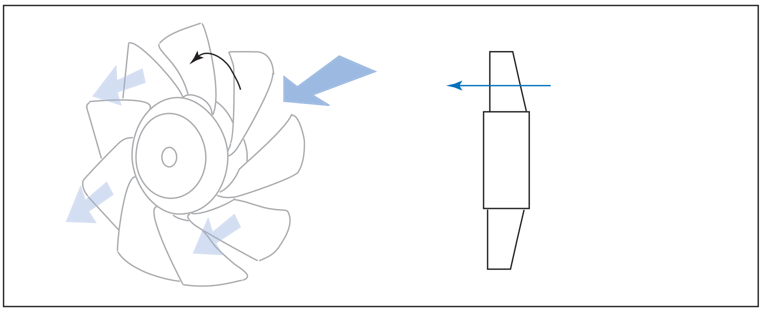


Figure 25. The air flow through an axial fan

Mixed flow fan
Radial impellers produce a static pressure increase because of the centrifugal force acting in a radial direction. There is no equivalent pressure increase with axial impellers because the air flow is normally axial. The mixed flow fan is a mixture between radial and axial fans. The air flows in an axial direction but then is deflected 45° in the impeller. The radial velocity factor which is gained by this deflection causes a certain increase in pressure by means of the centrifugal force. Efficiency of up to 80% can be achieved.

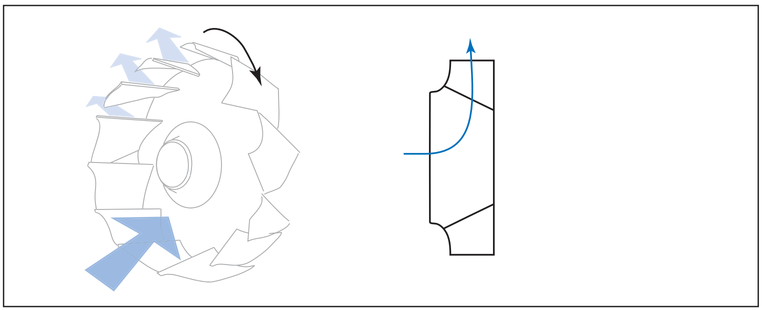


Figure 26. The air flow through a mixed flow fan

Cross-flow fan
In a cross-flow fan the air flows straight across the impeller, and both the in and out flow are in the periphery of the impeller. In spite of its small diameter, the impeller can supply large volumes of air and is therefore suitable for building into small ventilation units, such as air curtains for example. Efficiency of up to 65% can be achieved.

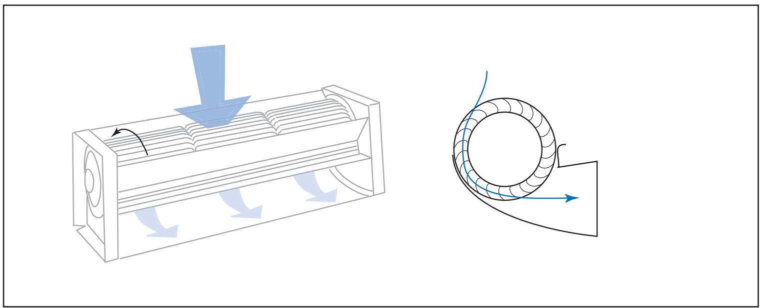


Figure 27. The air flow through a cross-flow fan

Theoretical calculation of the system line

$$\Delta P = k \cdot q_v^2$$

where
ΔP = the fan's total pressure (Pa)
qv = air flow (m^3/h or l/s)
k = constant

Example
A certain fan produces an air flow of 5000 m^3/h at a pressure of 250 Pa.

A. How does one produce a system line in the diagram?

a) Mark the point on the fan curve (1) where the pressure is 250 Pa and the air flow is 5000 m^3/h.

Enter the same value in the formula above to obtain a value for the constant k.
$k = \Delta P/q_V^2 = 250/5000^2 = 0.00001$

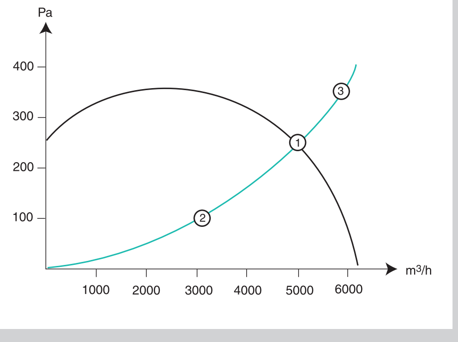
b) Select an arbitrary pressure reduction, for example 100 Pa, calculate the air flow and mark point (2) in the diagram.

$$q = \sqrt{100/(0,00001)} = 3162\ m^3/h$$

c) Do the same thing for 350 Pa and mark point (3) in the diagram.

$$q = \sqrt{350/(0,00001)} = 5916\ m^3/h$$

d) Now draw a curve that indicates the system line.



Fan curves
The fan diagram indicates the fan's capacity at different pressures. Each pressure corresponds to a certain air flow, which is illustrated by a fan curve.

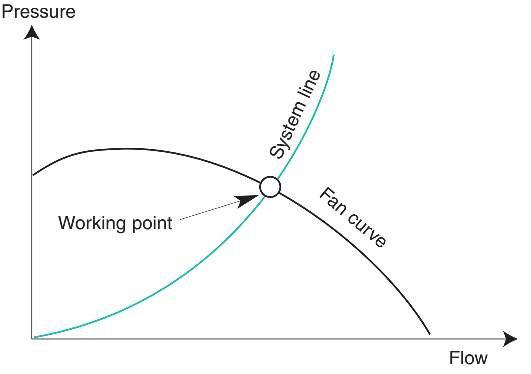


Figure 28. Curves in a typical fan diagram

System lines
The duct system's pressure requirement for various air flows is represented by the system line. The fan's working point is indicated by the intersection between the system line and the fan curve. This shows the air flow which the duct system will produce.

Each change of pressure in the ventilation system gives rise to a new system line. If the pressure increases, the system line will be the same as line B. If the pressure reduces, the system line will be the same as line C instead. (This only applies if the rotational speed of the impeller, i.e. the revolution count, remains constant).

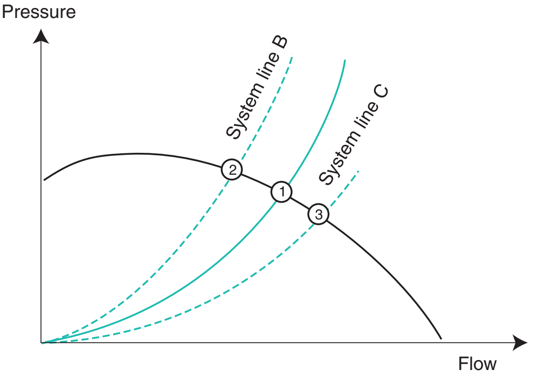


Figure 29. Changes in pressure give rise to new system lines

If the ventilation system's actual pressure requirement is the same as system line B, the working point will move from 1 to 2. This will also entail a weaker air flow. In the same way, the air flow will increase if the system's pressure requirement corresponds instead to line C.

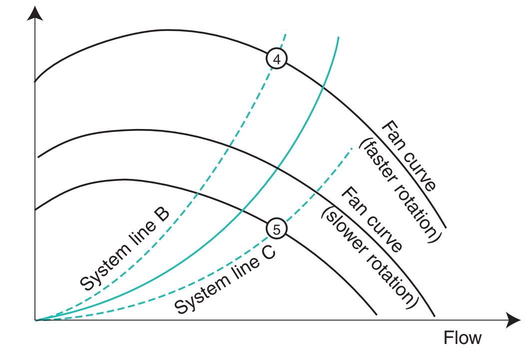


Figure 30. Increase or reduction of the fan speed

To obtain the same air flow as calculated, one can in the first case (where the system line corresponds to B) quite simply increase the fan speed. The working point (4) will then be at the intersection of system line B and the fan curve for a higher rotational speed. In the same way, the fan speed can be reduced if the actual system line corresponds to line C.

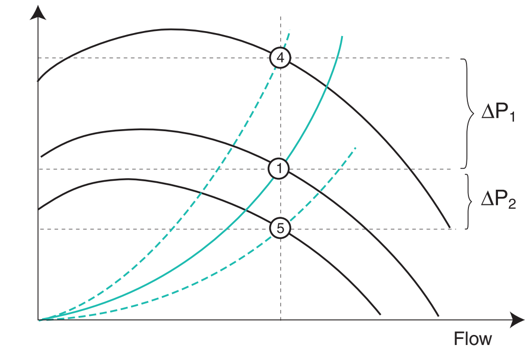


Figure 31. Pressure differences at different rotational speeds

In both cases, there will be a certain difference in pressure from that of the system for which the dimensioning has been calculated, and this is shown as DP1 and DP2 respectively in the figure. This means that if the working point for the calculated system has been chosen so as to give the maximum degree of efficiency, any such increase or decrease of the fan's rotational speed will reduce the fan's efficiency.

B. What will happen if the pressure in the system increases by 100 Pa, (for example because of a clogged filter)?

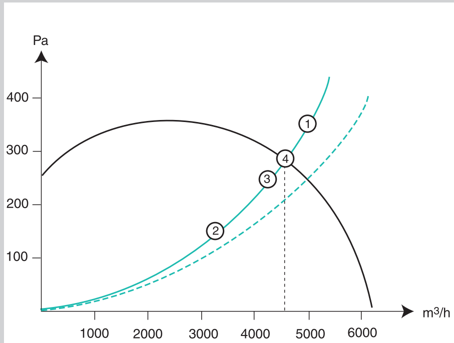
a) Calculate the constant for the new system line: $k = 350/5000^2 = 0{,}000014$

b) Select two other pressure reductions, for example 150 and 250 Pa, and calculate the air flow for them.

$$q = \sqrt{150/0{,}000014} = 4225 \text{ m}^3\text{/h}$$

$$q = \sqrt{250/0{,}000014} = 3273 \text{ m}^3\text{/h}$$

c) Plot in the two new points (2 and 3) and draw in the new system line.



The new working point (4) is located at the intersection between the fan curve and the new system line.

This diagram also indicates that the pressure increase causes a reduction of the air flow to approximately 4500 m³/h.

Definition of the system line

$$L = 10 \cdot \sqrt{\frac{\Delta P_d}{\Delta P_t}}$$

where
L = the fan's system line
Δp_d = dynamic pressure (Pa)
Δp_t = total pressure (Pa)

Efficiency and system lines

To facilitate the selection of a fan, one can plot in a number of considered system lines in a fan diagram and then see between which lines a particular type of fan should operate. If the lines are numbered 0 to 10, the fan will be completely free-blowing (maximum air flow) at line 10 and will be completely choked (no air flow at all) at line 0. This then means that the fan at system line 4 produces 40% of its free-blowing air flow.

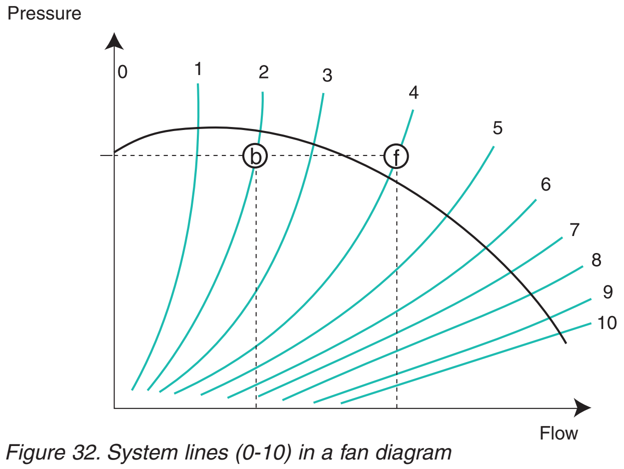


Figure 32. System lines (0-10) in a fan diagram

Each fan's efficiency remains constant along one and the same system line. Fans with backward-curved blades frequently have a greater efficiency than fans with forward-curved blades. But these higher levels of efficiency are only achievable within a limited area where the system line represents a weaker air flow at a given pressure than is the case with fans with forward-curved blades.

To achieve the same air flow as for a fan with forward-curved blades, while at the same time maintaining a high level of efficiency, a fan with backward-curving blades in a larger size would have to be selected.

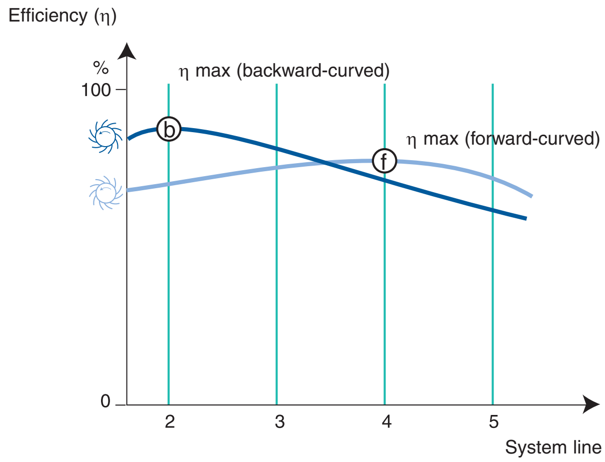


Figure 33. Efficiency values for the same size of radial fan with backward-curved and forward-curved blades respectively

Fan application
It is assumed in the fan diagram that the fan's connections to the inlet and outlet are designed in a specific way. There must be at least 1 x the duct diameter on the suction side (inlet) and 3 x the duct diameter on the pressure side (outlet).

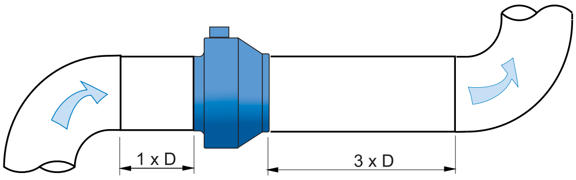


Figure 34. Correctly installed duct fan

If the connections are different from this, there could be a greater pressure reduction. This extra pressure drop is called the system effect or system dissipation, and can cause the fan to produce a smaller volume of air than indicated in the fan diagram. The following factors must be considered in order to avoid system dissipation:

At the inlet

- The distance to the nearest wall must be more than 0.75 x the inlet's diameter
- The inlet duct's cross-section must not be greater than 112% or less than 92% of the fan inlet
- The inlet duct's length must be at least 1 x the duct diameter
- The inlet duct must not have any obstacles to the air flow (dampers, branching or similar)

At the outlet

- The angle at the reduction of the duct cross-section must be less than 15°
- The angle at the enlargement of the duct cross-section must be less than 7°
- A straight length of at least 3 x the duct diameter is required after a duct fan
- Avoid 90° bends (use 45°)
- Bends must be shaped so that they follow the air stream after the fan

Specific Fan Power
There are now stringent requirements to ensure that power consumption in a building is as efficient as possible so as to minimise energy costs. The Svenska Inneklimatinstitutet [Swedish Inner Climate Institute] has introduced a special concept known as the Specific Fan Power (SFP_E) as a measurement of a ventilation system's energy efficiency.

The Specific Fan Power for an entire building can be defined as the total energy efficiency of all the fans in the ventilation system divided by the total air flow through the building. The lower the value, the more efficient the system is at transferring the air.

The recommendations for public sector purchasing and similar are that the maximum SFP_E should be 2.0 when maintaining and repairing ventilating units, and 1.5 for new installations.

Efficiency of a fan

$$\eta = \frac{\Delta P_t \cdot q}{P}$$

where
ΔP_t = total pressure change (Pa)
q = air flow (m³/s)
P = power (W)

Specific Fan Power
The Specific fan power for an entire building

$$SFP_E = \frac{P_{tf} + P_{ff}}{q_f} \ (kW/m^3/s)$$

where
P_{tf} = total power for air supply fans (kW)
P_{ff} = total power for air exhaust fans (kW)
q_f = dimensioned air flow (m³/s)

Theoretical calculation of a fan's power consumption

$$P = \frac{p_t \cdot q}{\eta_{fan} \cdot \eta_{belt} \cdot \eta_{motor}}$$

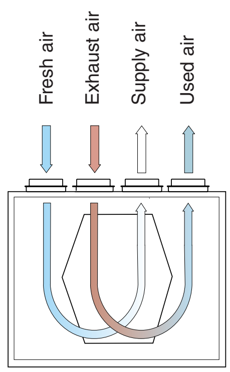
where
P = the fan's consumption of electric power from the network (kW)
p_t = the fan's total pressure (Pa)
q = air flow (m³/s)
η_{fan} = the fan's efficiency
η_{belt} = efficiency of the transmission
η_{motor} = efficiency of the fan motor

Thermal efficiency

$$\eta = \frac{t_i - t_u}{t_f - t_u}$$

where
t_u = outside air temperature
t_f = exhaust air temp. (no heat recovery)
t_i = supply air temp. (after heat recovery)

Counterflow plate heat recovery units
The air streams (exhaust and supply air) pass in opposite directions through the entire heat recovery unit, which results in an efficient recovery of heat.



Heat recovery units

In a ventilating unit, it is often economical to attempt to recover the heat which is contained in the exhaust air and use it to warm the supply air. There are several methods for achieving this type of heat recovery.

Plate heat recovery units

The exhaust air and supply air pass on each side of a number of plates or lamellae. The exhaust- and supply air are not in contact with each other which results in low leakage. There may be some condensation in a plate heat recovery unit, so they need to be fitted with condensation drains. The drains should have a water seal to prevent the fans from sending the water back into the unit. Because of this condensation there is also a serious risk of ice formation, so some type of defrosting system is also needed. Heat recovery can be regulated by means of a bypass valve which controls the intake of exhaust air. Plate heat recovery units have no moving parts. High efficiency (50-90%).

Rotary heat recovery units

Heat is transferred by a rotating wheel between exhaust and supply air. This system is open and there is a risk that impurities and odours will be transferred from the exhaust to the supply air. This can be avoided to some extent by correct designed ventilation system with the right pressure conditions or by positioning the fans in a preventing way. The degree of heat recovery can be regulated by increasing or decreasing the rotational speed. There is little risk of freezing in the heat recovery unit. Rotary heat exchange units contain moving parts. High efficiency (75-85%).

Battery heat recovery units

Water, or water mixed with glycol, circulates between a water battery in the exhaust air duct and a water battery in the supply air duct. The liquid in the exhaust air duct is heated so that it can transfer the heat to the air in the supply air duct. The liquid circulates in a closed system and there is no risk of transferring impurities from exhaust air to supply air. Heat recovery can be regulated by increasing or decreasing the water flow. Battery heat recovery units have no moving parts. Low efficiency (45-60%).

Chamber heat exchangers

A chamber is divided into two parts by a damper valve. The exhaust air first heats one part of the chamber, then the damper valve changes the air stream so that the supply air is heated by the warmed-up part of the chamber. Impurities and odours can be transferred from exhaust air to supply air. The only moving part in a chamber heat exchanger is the damper valve. High efficiency (80-90%).

Heat pipe

This heat recovery unit consists of a closed system of pipes filled with a liquid that vaporises when heated by the exhaust air. When the supply air passes the pipes, the vapour condenses back into liquid again. There can be no transfer of impurities, and the heat recovery unit has no moving parts. Low efficiency (50-70%).

Heating batteries

In most cases the outside air is colder than the required temperature for the supply air, so it is often necessary to warm the air before it enters the building. The air can be warmed in a heating battery, by using either a hot water, or an electric heating battery.

Electric-heating battery
An electric-heating battery consists of a number of enclosed metal filaments or wire spirals. They create an electrical resistance which converts the energy to heat. The advantages of the electric battery are: it has a small pressure drop, it is easy to calculate the power and it is inexpensive to install. The disadvantage is that the metal filaments have a considerable heat inertia so the electric battery has to be fitted with overheating protection.

Water-heating battery
Crossflow water-heating batteries are the most common type of water-heating batteries in ventilation units. The water flows at right angles and in the opposite direction to the air stream. The water is conducted from below and flows upwards through the battery, and this allows any air bubbles to collect at the highest point where they can be easily drawn off via a ventilating pipe.

Water-heating batteries have to be protected against ice formation to ensure they do not crack as the result of freezing. The greatest risk of this happening is actually when the air temperature is immediately below 0°C. Most water batteries therefore have a frost guard which stops the intake of fresh air when there is a risk of freezing. Because still water freezes faster than flowing water, it is also usual to fit an internal pump which keeps the water flowing through the battery.

The air velocity through the battery, calculated for the entire front area, should be dimensioned to 2-5 m/s. The water velocity should not be below 0.2 m/s, as this could cause difficulties with venting. Nor should the water velocity be higher than 1.5 m/s in copper pipes or 3 m/s in steel pipes, as this could lead to erosion of the metal pipes.

Filters
There are two reasons for using filters in an air-handling unit: to prevent impurities in the outside air from entering the building and to protect the unit's components from contamination.

An analysis of the impurities in the air indicates that among other things the air contains soot particles, smoke, metallic dust, pollen, viruses and bacteria. The particles vary in size from less than 1 µm to whole fibres, leaves and insects. It is thought that these pollutants are a significant contributing factor in the cause of many asthmatic and allergic conditions, and it is therefore important for people to protect themselves against them.

Since as much as 99.99% of all particles in the air are smaller than 1 µm, it is necessary to use filters in a ventilation system that are adequately fine-meshed. The filter's capacity to trap particles is called its Dust Holding Capacity and filters are often divided into three classes depending on this capacity: coarse filters, fine filters and absolute filters.

Filter classes

Coarse filter	EU1 to EU4
Fine filter	EU5 to EU9
Absolute filter	EU10 to EU14

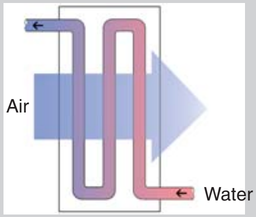
The coarse filter essentially only traps particles larger than 5 µm, and has virtually no effect at all on particles smaller than 2 µm. This means, therefore, that it does not trap soot particles, which are the most prevalent impurities in the outside air. Fine filters should be fitted in a ventilation unit instead. The best fine filters work effectively with particles larger than 0.1 µm, and therefore trap the most important impurities in the outside air.

Water-heating battery
The power input (kW) to a water-heating battery in a ventilating unit is:

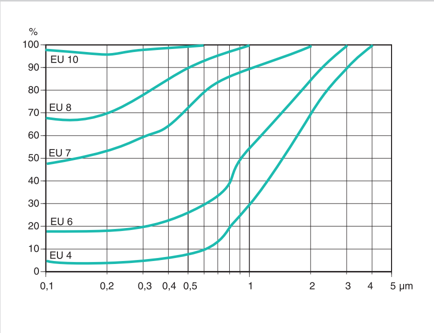
$$Q = \frac{L \cdot 1{,}2}{3600} \cdot (t_i - t_u) \cdot (1 - \eta)$$

where
L = the air flow (m³/h)
t_i = required supply air temperature (°C)
t_u = dimensioned outside temperature (°C)
η = the efficiency of the heat recovery unit

Water battery
The hot water should be conducted in the opposite direction to the air, otherwise it will cool too quickly and the water battery's warming of the air will not be as efficient.



Dust Holding Capacity for different filter classes



Pressure drop

The pressure drop caused by a completely clean filter is called the start pressure drop, and this is somewhere between 80 and 120 Pa for fine filters. After impurities have been trapped by the filter, the pressure drop will increase and the air flow will be reduced. Eventually there will be a pressure drop which makes the filter no longer usable. For fine filters this will be between 200 and 250 Pa. It is usual for filters in a unit to be fitted with some kind of filter monitor which constantly measures the pressure drop caused by the filter. This can give a signal when a pre-set pressure drop has been reached and it is time to replace the filter. In any event it is advisable to replace the filter twice a year, irrespective of whether or not the final pressure drop has been reached, so as to prevent the dirt in the filter becoming a breeding ground for bacteria.

Suppliers of filters have been debating for a long time as to whether glass fibre or synthetic fibre provides the best filter material. Some research has been carried out, but without any clear results. It appears, however, that glass fibre filters maintain a better Dust Holding Capacity throughout their working life.

Just as important as the selection of the filter material is the need to ensure that there is a good seal around the filter to prevent dirt and dust passing around the edge. The filter housing should be designed so that repeated filter replacements can be made without any space developing between the filter and the housing. It is also important to protect the filter from moisture as this can alter the characteristics of the filter fibres and impair its Dust Holding Capacity. Glass fibre filters are more susceptible to the effects of moisture than synthetic filters.

ACOUSTICS

Basic principles of sound

Before we discuss the connection between the sound power level and the sound pressure level, we must define certain basic concepts such as sound pressure, sound power and frequency.

Sound pressure

Sound pressure is the pressure waves with which the sound moves in a medium, for instance air. The ear interprets these pressure waves as sound. They are measured in Pascal (Pa).

The weakest sound pressure that the ear can interpret is 0.00002 Pa, which is the threshold of hearing. The strongest sound pressure which the ear can tolerate without damage is 20 Pa, referred to as the upper threshold of hearing. The large difference in pressure, as measured in Pa, between the threshold of hearing and the upper threshold of hearing, makes the figures difficult to handle. So a logarithmic scale is used instead, which is based on the difference between the actual sound pressure level and the sound pressure at the threshold of hearing. This scale uses the decibel (dB) unit of measurement, where the threshold of hearing is equal to 0 dB and the upper threshold of hearing is 120 dB.

The sound pressure reduces as the distance from the sound source increases, and is affected by the room's characteristics and the location of the sound source.

Sound power

Sound power is the energy per time unit (Watt) which the sound source emits. The sound power is not measured, but it is calculated from the sound pressure. There is a logarithmic scale for sound power similar to the scale for sound pressure.

The sound power is not dependent on the position of the sound source or the room's sound properties, and it is therefore easier to compare between different objects.

Frequency

Frequency is a measurement of the sound source's periodic oscillations. Frequency is measured as the number of oscillations per second, where one oscillation per second equals 1 Hertz (Hz). More oscillations per second, i.e. a higher frequency, produces a higher tone.

Frequencies are often divided into 8 groups, known as octave bands: 63 Hz, 125 Hz, 250 Hz, 500 Hz, 1000 Hz, 2000 Hz, 4000 Hz and 8000 Hz.

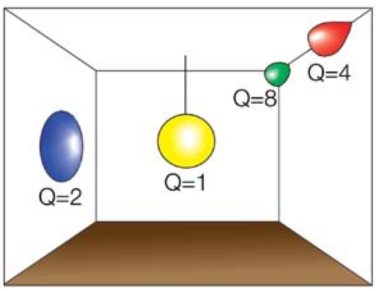
Sound power level and sound pressure level

There is a link between a sound source's sound power level and the sound pressure level. If a sound source emits a certain sound power level, the following factors will affect the sound pressure level:

The position of the sound source in the room, including the direction factor (1), the distance from the sound source (2) and the room's sound-absorbing properties, referred to as the room's equivalent absorption area (3).

1) Direction factor, Q

The direction factor indicates the sound's distribution around the sound source. A distribution in all directions, spherical, is measured as Q = 1. Distribution from a diffuser positioned in the middle of a wall is hespherical, measured as Q = 2.



Q = 1 In centre of room
Q = 2 On wall or ceiling
Q = 4 Between wall or ceiling
Q = 8 In a corner

Figure 1. The distribution of sound around the sound source

Calculation of equivalent absorption area A_{eqv}

$$A_{eqv} = \alpha_1 \cdot S_1 + \alpha_2 \cdot S_2 + ... + \alpha_n + S_n$$

where

S = Size of surface (m^2)
a = Absorption factor, depending on the material
n = Number of surfaces

Calculation of sound pressure level

Estimate based on figures 1, 2 and 3 together with table 1.

A normally damped room in a nursing home, measuring 30 m³, is to be ventilated. According to the information in the catalogue, the directional supply-air terminal device fitted in the ceiling has a sound pressure level (L_{pA}) of 33 dB(A). This applies to a room with a space damping equivalent to 10 m² Sabine, or 4 dB(A).

A) What will the sound pressure level be in this room, 1 m from the diffuser?

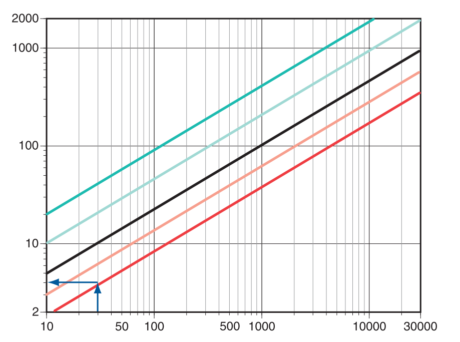
The sound pressure level depends on the room's acoustic properties, so first of all it is necessary to convert the value in the catalogue to a sound power level (L_{WA}).

Fig.3 shows that ΔL (space damping) = L_{pA} - L_{WA}

$L_{WA} = L_{pA} + \Delta L$

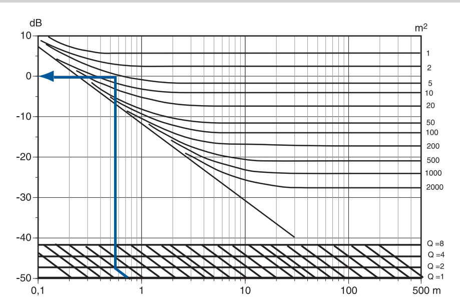
$L_{WA} = 33 + 4 = 37$ dB(A)

With the following values
r = 1
Q = 2 (fig.1)
and information about the room's dimensions, you can calculate the equivalent absorption area with the help of figure 2.



The equivalent absorption area is therefore 4 m².

It is now possible to use figure 3 to establish the difference between the sound pressure and the sound power.



$L_{pA} - L_{WA} = 0$
$L_{pA} = 0 + L_{WA}$

Enter the L_{WA} value which has already been calculated.
$L_{pA} = 0 + 37 = 37$ dB(A)

A) The sound pressure level (L_{pA}) one metre from the diffuser in this particular nursing home room is therefore 37 dB(A).

This calculation has to be made for all rooms not corresponding to the information in the catalogue which assumes a standard 10 m² Sabine.

The less damped (harder) the room is, the higher the actual sound pressure level will be in comparison with the value indicated in the catalogue.

2) Distance from sound source, r
Where r indicates the distance from the sound source in metres.

3) The room's equivalent absorption area, Aeqv
A material's ability to absorb sound is indicated as absorption factor a. The absorption factor can have a value between '0' and '1', where the value '1' corresponds to a fully absorbent surface and the value '0' to a fully reflective surface. The absorption factor depends on the qualities of the material, and tables are available which indicate the value for different materials.

A room's equivalent absorption area is measured in m² and is obtained by adding together all the different surfaces of the room multiplied by their respective absorption factors.

In many instances it can be simpler to use the mean value for sound absorption in different types of rooms, together with an estimate of the equivalent absorption area (see figure 2).

3) Equivalent absorption area based on estimates
If values are not available for the absorption factors of all the surfaces, and a more approximate value of the room's total absorption factor is quite adequate, an estimate can be calculated in accordance with the diagram below. The diagram is valid for rooms with normal proportions, for example 1:1 or 5:2.

Use the diagram as follows to estimate the equivalent absorption area: calculate the room's volume and read off the equivalent absorption area with the correct mean absorption factor, determined by the type of room, see also table 1.

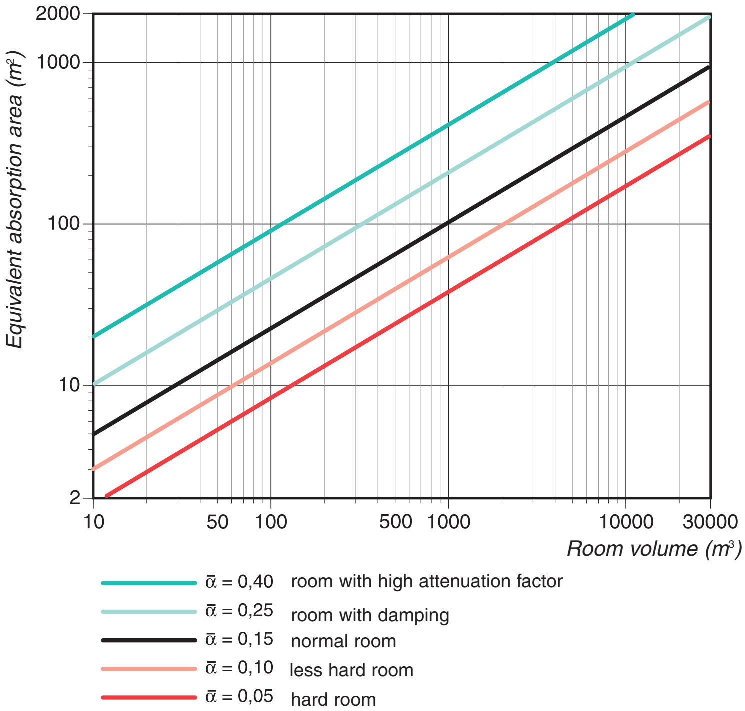


Figure 2. Estimate of equivalent absorption area.

Type of room	Mean absorption factor
Radio studios, music rooms	0.30 - 0.45
TV studios, department stores, reading rooms	0.15 - 0.25
Domestic housing, offices, hotel rooms, conference rooms, theatres	0.10 - 0.15
School halls, nursing homes, small churches	0.05 - 0.10
Industrial premises, swimming pools, large churches	0.03 - 0.05

Table 1. Mean absorption factors for different types of rooms

Calculation of sound pressure level
With the help of the factors previously described, it is now possible to calculate the sound pressure level if the sound power level is known. The sound pressure level can be calculated by means of a formula incorporating these factors, but this equation can also be reproduced in the form of a diagram.

When the diagram is used for calculating the sound pressure level, you must start with the distance in metres from the sound source (r), apply the appropriate directional factor (Q), and then read off the difference between the sound power level and the sound pressure level next to the relevant equivalent absorption area (A_{eqv}). This result is then added to the previously calculated sound power (see also the example on page 509).

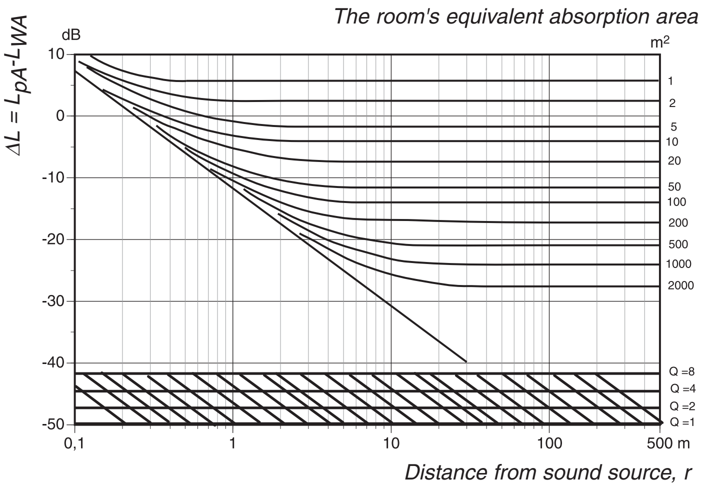


Figure 3. Diagram for estimating the sound pressure level

Calculation of sound pressure level

$$L_{pA} = L_{wA} + 10 \cdot \log\left[\frac{Q}{4\pi r^2} + \frac{4}{A_{eqv}}\right]$$

where
L_{pA} = sound pressure level (dB)
L_{wA}= sound power level (dB)
Q = direction factor
r = distance from sound source (m)
A_{eqv} = equivalent absorption area (m^2 Sabine)

Near field and reverberation field

Near field is the term used for the area where the sound from the sound source dominates the sound level. The reverberation field is the area where the reflected sound is dominant, and it is no longer possible to determine where the original sound comes from.

The direct sound diminishes as the distance from the sound source increases, while the reflected sound has approximately the same value in all parts of the room.

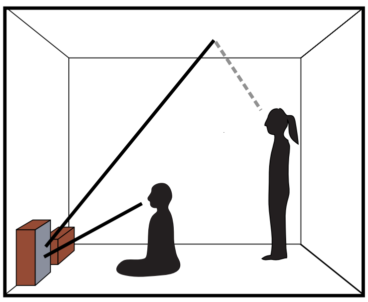


Figure 4. Direct and reflected sound

The reverberation time indicates the time it takes for the sound level to reduce by 60 dB from the initial value. This is the echo effect one hears in a quiet room when a powerful sound source is switched off. If the reverberation time is measured precisely enough, the equivalent absorption area can be calculated.

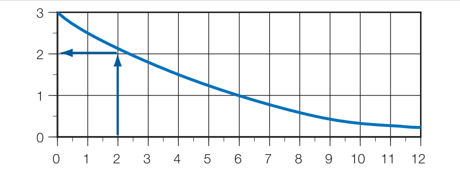
Calculation of reverberation time
If a room is not too effectively damped (i.e. with a mean absorption factor of less than 0.25), the room's reverberation time can be calculated with the help of Sabine's formula:

$$T = 0{,}163 \cdot \frac{V}{A_{eqv}}$$

where
T= Reverberation time (s). Time for a 60 dB reduction of the sound pressure value
V= Room volume (m^3)
A_{eqv}=The room's equivalent absorption area, m^2

Example of addition
There are two sound sources, 40 dB and 38 dB respectively.

1) What is the value of the total sound level?

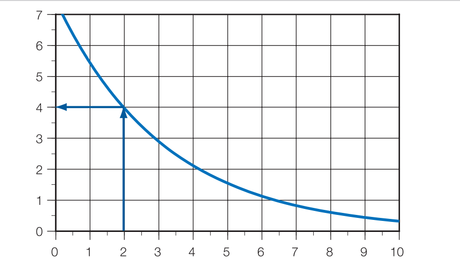


The difference between the sound levels is 2 dB and, according to the diagram, 2 dB must be added to the highest level.

1) The total sound level is therefore 42 dB.

Example of subtraction
The total sound level is 34 dB in a room fitted with both supply and exhaust ventilation systems. It is known that the supply system produces 32 dB, but the value for the exhaust system is not known.

2) What is the sound level produced by the exhaust system?



The difference between the total sound level and the sound level of the supply system is 2 dB. The diagram indicates that 4 dB must be deducted from the total level.

2) Therefore the exhaust system produces 30 dB.

Several sound sources

To establish the total sound level in a room, all the sound sources must be added together logarithmically. It is, however, often more practicable to use a diagram to calculate the addition or subtraction of two dB values.

Addition
The input value for the diagram is the difference in dB between the two sound levels which are to be added. The dB value to be added to the highest sound level can then be read off they scale.

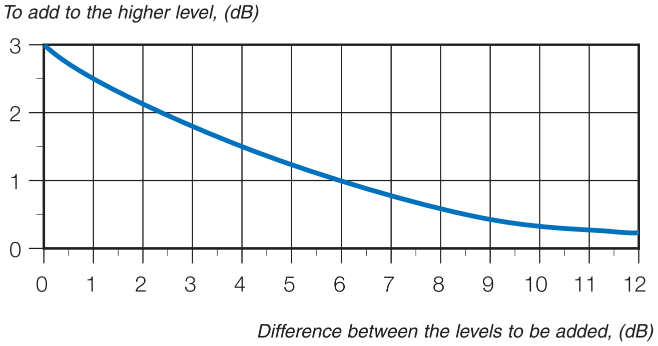


Figure 5. Logarithmic addition

Subtraction
The input value for the diagram is the difference in dB between the total sound level and the known sound source. The y scale then shows the number of dB that have to be deducted from the total sound level to obtain the value for the unknown sound source.

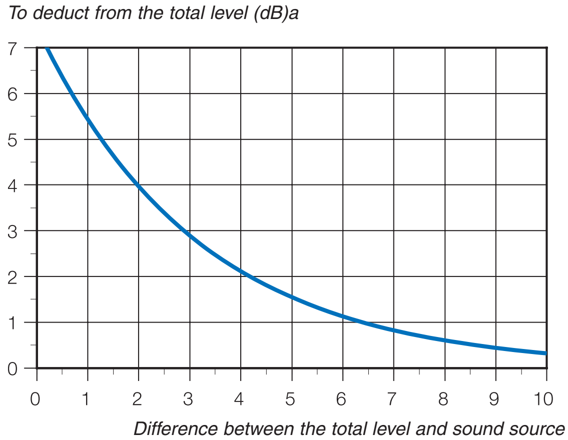


Figure 6. Logarithmic subtraction

Adjustment to the ear

Because of the ear's varying sensitivity at different frequencies, the same sound level in both low and high frequencies can be perceived as two different sound levels. As a rule, we perceive sounds at higher frequencies more easily than at lower frequencies.

A filter
The sensitivity of the ear also varies in response to the sound's strength. A number of so called weighting filters have been introduced to compensate for the ear's variable sensitivity across the octave band. A weighting filter A is used for sound pressure levels below 55 dB. Filter B is used for levels between 55 and 85 dB, and filter C is used for levels above 85 dB.

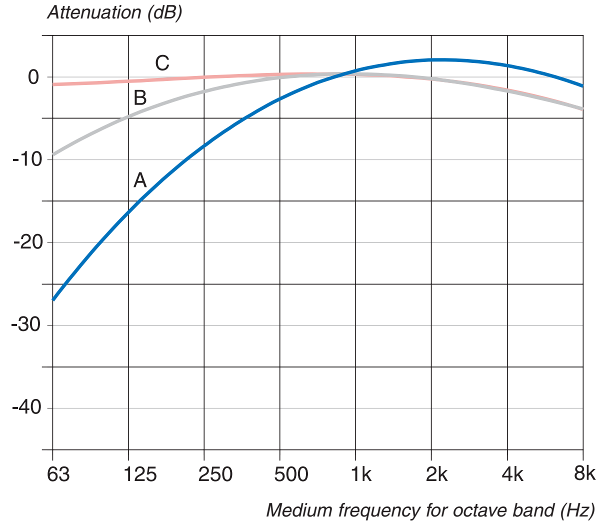


Figure 7. Damping with different filters

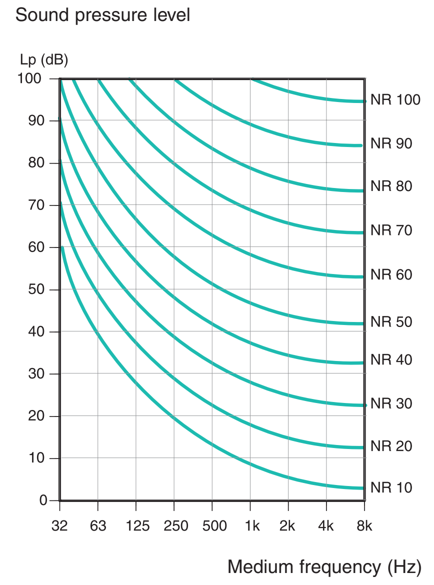
The A filter, which is commonly used in connection with ventilation systems, has a damping effect on each octave band as shown in table 2. The resultant value is measured in dB(A) units.

Hz	63	125	250	500	1k	2k	4k	8k
dB	-26,2	-16.1	-8.6	-3.2	0	+1.2	+1.2	-1.1

Table 2. Damping with the A filter

There are also other ways of compensating for the ear's sensitivity to different sound levels, apart from these filters. A diagram with NR curves (Noise Rating) shows sound pressure and frequency (per octave band). Points on the same NR curve are perceived as having the same sound levels, meaning that 43 dB at 4000 Hz is perceived as being as loud as 65 dB at 125 Hz.

NR curves



Sound attenuation

Sound attenuation is principally achieved in two ways: either by absorption or by reflection of the sound.

Attenuation by absorption is achieved by internal insulation in ducts, by special silencers or by means of the room's own sound absorption. Attenuation by reflection is achieved by forking or bending, or when the sound bounces back from a supply-air device into the duct, which is referred to as end reflection.

The degree of sound attenuation can be calculated by using tables and diagrams presented in the relevant suppliers technical documentation.

Air terminal devices

There are essentially two ways of ventilating a building: ventilation by displacement and ventilation by diffusion.

Ventilation by diffusion is the preferable method for supplying air in situations requiring what is known as comfort ventilation. This is based on the principle of supplying air outside the occupied zone which then circulates the air in the entire room. The ventilation system must be dimensioned so that the air which circulates in the occupied zone is comfortable enough, in other words the velocity must not be too high and the temperature must be more or less the same throughout the zone.

Ventilation by displacement is chiefly used to ventilate large industrial premises, as it can remove large volumes of impurities and heat if properly dimensioned. The air is supplied at low velocity directly into the occupied zone. This method provides excellent air quality, but is less suitable for offices and other smaller premises because the directional supply-air terminal device takes up a considerable space and it is often difficult to avoid some amount of draught in occupied areas.

The theory section which follows will discuss what happens to the air in rooms ventilated by diffusion, how to calculate air velocity and displacement in the room, and also how to select and position a directional supply-air terminal device correctly in the premises.

Ventilation by diffusion
An air stream which is injected into a room will attract, and mix together with, large volumes of ambient air. As a result, the air stream's volume increases while at the same time the air velocity is reduced the further into the room it travels. The mixing of the surrounding air into the air stream is termed 'induction'.

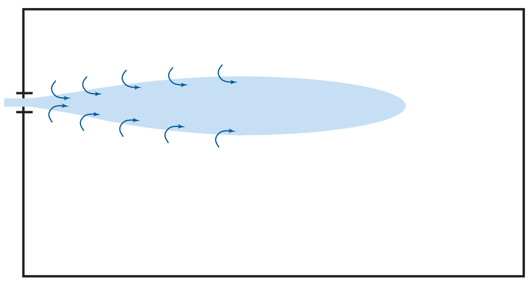
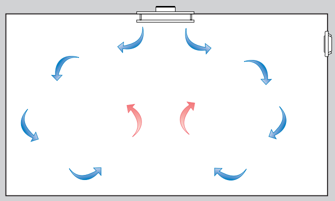


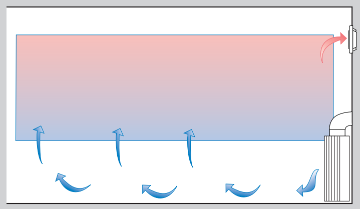
Figure 8. Induction of the surrounding air into the air stream.

The air movements caused by the air stream very soon mix all the air in the room thoroughly. Impurities in the air are not only attenuated but also evenly distributed. The temperatures in the different parts of the room are also evened out.

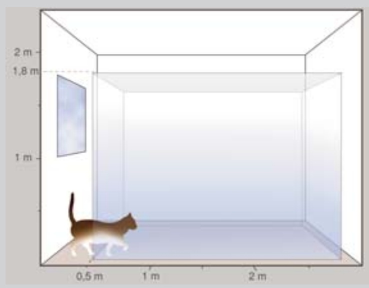
When dimensioning for ventilation by diffusion, the most important consideration is to ensure that the air velocity in the occupied zone will not be too high, as this will be experienced as a draught.



Ventilation by diffusion
The air is blown in from one or more air streams outside the occupied zone.



Ventilation by displacement
Air which is somewhat cooler than the ambient air flows at low velocity into the occupied zone.



Occupied zone
The occupied zone is that part of the room normally occupied by people. This is usually defined as being a space 50 cm from an outer wall with windows, 20 cm from other walls, and up to 180 cm above the floor.

Air stream theory

The figure below shows an air stream that is formed when air is forced into a room through an opening in the wall. The result is a free air stream. If it also has the same temperature as the rest of the room, it is referred to as a free isotherm stream. To begin with, this section will only deal with streams of this type.

Distribution and shape
The air stream actually consists of several zones with different flow conditions and air velocities. The area which is of most practical interest is the main section. The centre velocity, the velocity around the centre axis, is in inverse proportion to the distance from the diffuser or valve, i.e. the further away from the diffuser the slower the air velocity.

The air stream is fully developed in the main section, and the prevailing conditions here are the ones that will principally affect the flow conditions in the room as a whole.

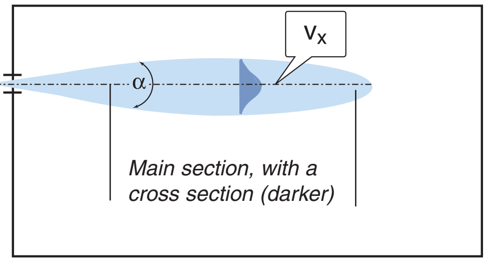


Figure 9. The main section of the air stream, the centre velocity vx and discharge angle.

The shape of the diffuser or valve opening determines the shape of the air stream. Circular or rectangular openings produce a conic (axial) stream, and this also applies to very long and narrow openings.

To produce a completely flat air stream, the opening must be more than ten times as wide as it is high, or nearly as wide as the room so that the walls prevent the stream widening out laterally.

Radial air streams are produced by completely circular openings where the air can spread in all directions, as is the case with a supply-air diffuser.

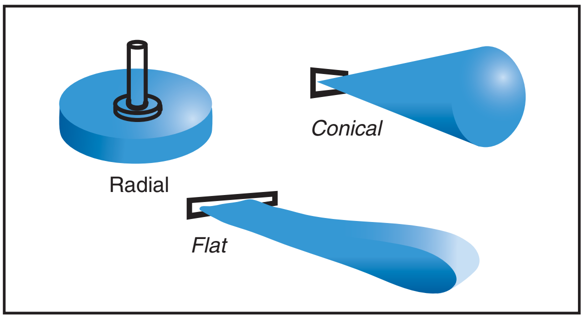


Figure 10. Different kinds of air stream

α = the discharge angle

The discharge angle
According to ASHRAE's Handbook (AHRAE [The American Society of Heating, Refrigerating and Air-Conditioning Engineers], 1996) the distribution of an air stream has a constant angle of 20-24° (22° on average).

The shape of the vent, the geometry of the room and also the number of vents all have an effect on the discharge angle. Diffusers and valves with plates or other details which spread the air can produce a wider discharge angle, but even after a relatively short distance from the valve opening, these air streams have a distribution of between 20 and 24°.

Calculation of air velocity
For a conical or radial air stream:

$$\frac{v_x}{v_0} = K \cdot \frac{\sqrt{A_{eff}}}{x} \qquad A_{eff} = \frac{q}{v_0}$$

x = distance from the diffuser or valve (m)
v_x = centre velocity at distance x (m/s)
v_0 =velocity at the diffuser/valve outlet (m/s)
K = the diffuser coefficient
A_{eff} = the diffuser/valve's effective outlet area (m^2)
q = air volume through the vent (m^3/s)

For a flat air stream

$$\frac{v_x}{v_0} = \sqrt{K \cdot \frac{h}{x}}$$

x = distance from the diffuser/valve (m)
v_x = velocity at distance x (m/s)
v_0 = velocity at the diffuser/valve outlet (m/s)
K = the diffuser coefficient
h = the height of the slot (m)

The velocity at the cross section of the air stream will be:

$$\frac{v}{v_x} = \left[1 - \left(\frac{y}{0,3 \cdot x}\right)^{1,5}\right]^2$$

y = vertical distance from the central axis (m)
x = distance from the diffuser/valve (m)
v = velocity at distance y (m)
vx = centre velocity at distance x (m/s)

Velocity profile
It is possible to calculate mathematically the air velocity in each part of the stream. To calculate the velocity at a particular distance from the diffuser or valve, it is necessary to know the air velocity at the diffuser/valve outlet, the shape of the diffuser/valve and the type of air stream produced by it. In the same way, it is also possible to see how the velocities vary in every cross section of the stream.

Using these calculations as the starting point, velocity curves for the entire stream can be drawn up. This enables one to determine the areas which have the same velocity. These areas are called isovels. By checking that the isovel corresponding to 0.2 m/s is outside the occupied zone, one can ensure that the air velocity will not exceed this level in the normally occupied areas.

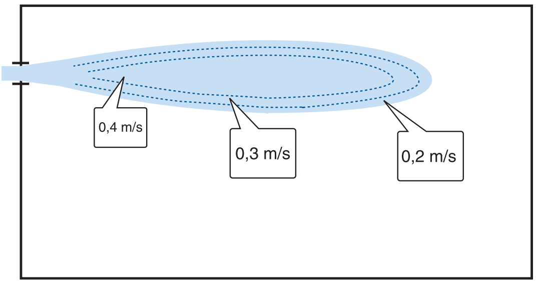


Figure 11. The different isovels of an air stream

The diffuser coefficient
The diffuser coefficient is a constant which depends on how the diffuser or valve is shaped. It can be calculated theoretically by using the following factors: the impulse dissipation and contraction of the air stream at the point where it is blown into the room, together with the degree of turbulence created by the diffuser or valve.

In practice, the constant is simply determined by taking measurements on each type of diffuser or valve. The air velocity is measured at a minimum of eight different distances from the diffuser/valve, with at least 30 cm between each measuring point. These values are then plotted into a logarithmic diagram, which indicates the measurement value for the main section of the air stream, and this in turn provides a value for the constant.

The diffuser coefficient enables one to calculate air velocities and to predict an air stream's distribution and path. It must not be confused with the K-factor which is used for such tasks as entering the correct air volume from a directional supply-air terminal device or iris damper.

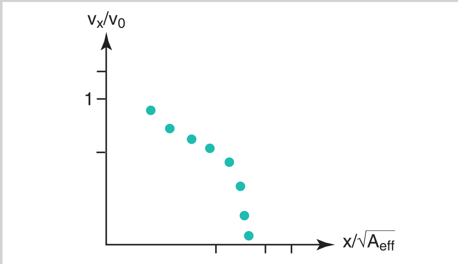
The K factor is described on page 389.

Theoretical calculation of the diffuser coefficient

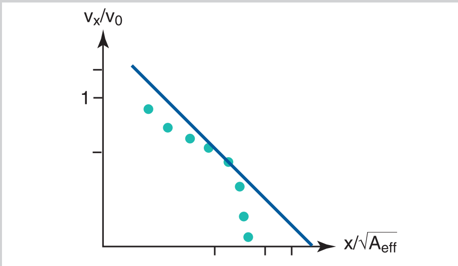
$$K = \sqrt{\frac{i}{\varepsilon}} \cdot \frac{1{,}5}{C_b}$$

i = impulse factor indicating impulse dissipation at point where air is blown in (i<1)
e = contraction factor
C_b = turbulence constant (0.2-0.3 depending on type of diffuser or valve)

Practical calculation of the diffuser coefficient
The measurement values (v_x/v_0) and (x/Ö A_{eff}) are plotted into the diagram.



Using the values obtained from the main section of the air stream, a tangent (angle coefficient) is drawn at an angle of -1 (45°).



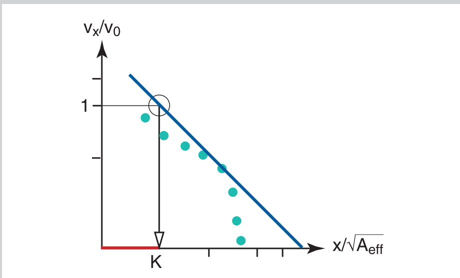
The formula for the velocity profile

$$\frac{v_x}{v_0} = K \cdot \frac{\sqrt{A_{eff}}}{x}$$

shows that

$$K = \frac{x}{\sqrt{A_{eff}}} \quad \text{when} \quad \frac{v_x}{v_0} = 1$$

A line should now be drawn from the intersection of the angle coefficient and 1 on the y scale to produce a value for the diffuser coefficient K.



The diffuser coefficient when the Coanda effect is influencing the air stream:

$$K_{corrected} = \sqrt{2} \cdot K_{free\ flow}$$

The horizontal discharge angle also increases to 30° when the stream is sucked towards the ceiling, while the vertical angle remains unchanged (20-24°).

Deflection
The deflection from the ceiling to the central axis of the air stream (Y) can be calculated using

$$Y = \sqrt{A_{eff}} \cdot 0{,}0014 \cdot \frac{\Delta t_0 \cdot \sqrt{A_{eff}}}{K \cdot v_0^2} \cdot \left[\frac{x}{\sqrt{A_{eff}}}\right]^3$$

where
Δt_0 = the temperature difference between the air stream and the ambient air
x = distance from the diffuser/valve (m)
v_x = centre velocity at distance x (m/s)
v_0 = velocity at the diffuser/valve outlet (m/s)
K = the diffuser coefficient
A_{eff} = the diffuser or valve's effective outlet area (m^2)

Point of separation
The point where a conical air stream leaves the ceiling (xm) will be:

$$x_m = \frac{1{,}6 \cdot K \cdot v_0 \cdot A_{eff}}{(A_{eff})^{0{,}75} \cdot \sqrt{\Delta t_0}}$$

and for a radial air stream will be

$$x_m = \frac{3{,}5 \cdot K^{1{,}5} \cdot v_0 \cdot A_{eff}}{(A_{eff})^{0{,}75} \cdot \sqrt{\Delta t_0}}$$

where
Δt_0 = the temperature difference between the air stream and the ambient air
v_0 = velocity at the diffuser/valve outlet (m/s)
K = the diffuser coefficient
A_{eff} = the diffuser or valve's effective outlet area (m^2)

After the stream has left the ceiling, a new path can be calculated with the aid of the formula for deflection (above). The distance x is then calculated as the distance from the point of separation.

Coanda effect
If a directional supply-air terminal device is fitted close enough to a flat surface, usually the ceiling, the air stream will cling to the surface. This is due to the fact that the ambient air will be drawn into the stream, but close to the flat surface, where no new air can flow from above, an underpressure forms instead, and this causes the stream to be sucked to the surface. This is known as the Coanda effect.

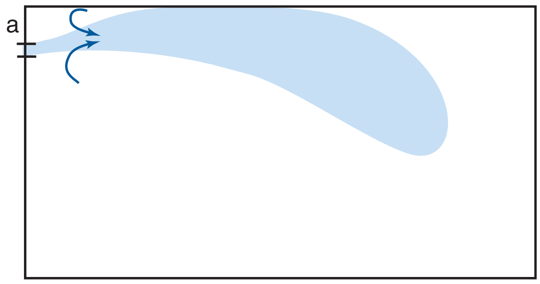


Figure 12. The Coanda effect

Practical experiments have shown that the distance between the diffuser or valve's upper edge and the ceiling ('a' in figure12) must not be greater than 30 cm if there is to be any suction effect.
The Coanda effect can be used to make a cold air stream stick to the ceiling and travel further into the room before it reaches the occupied zone.
The diffuser coefficient will be somewhat greater in conjunction with the suction effect than for a free air stream. It is also important to know how the diffuser or valve is mounted when using the diffuser coefficient for different calculations.

Non-isothermal air
The flow picture becomes more complex when the air that is blown in is non-isothermal air, in other words warmer or colder than the ambient air. A thermal energy, caused by differences in the air's density at different temperatures, will force a cooler air stream downwards and a warmer air stream upwards.
This means that two different forces affect a cooler stream that is sticking close to the ceiling: both the Coanda effect which attempts to adhere it to the ceiling and the thermal energy which attempts to force it towards the floor. At a given distance from the diffuser or valve's outlet, the thermal energy will dominate and the air stream will eventually be dragged down from the ceiling.
The stream's deflection and point of separation can be calculated using formulae which are based on the temperature differentials, the type of diffuser or valve and the size of its outlet, together with air velocities etc.

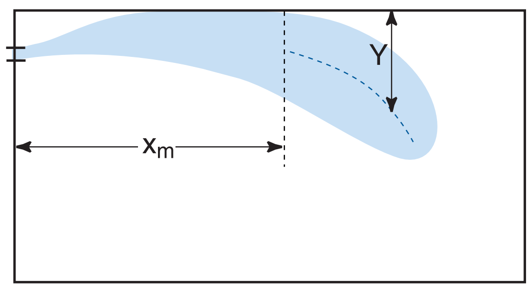


Figure 13. The air stream's point of separation (Xm) and deflection (Y)

Important considerations when dimensioning air supply

It is important to select and position the directional supply-air terminal device correctly. It is also important that the air temperature and velocity are as required for producing acceptable conditions in the occupied zone.

Correct air velocity in the occupied zone

A specification called 'throw' is indicated for most supply-air equipment in the manufacturer's product catalogue. 'Throw' is defined as the distance from the diffuser or valve opening to the point in the air stream where the centre velocity has been reduced to a particular value, generally 0.2 m/s. A throw of this type is designated by $l_{0.2}$ and is measured in metres.

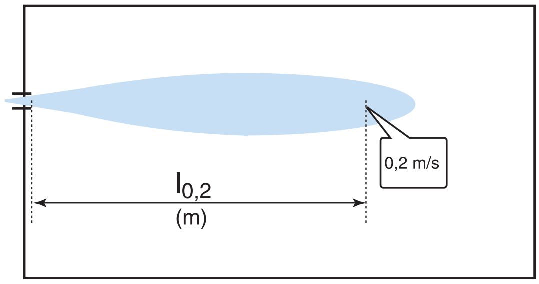


Figure 14. The 'throw' concept

One of the first considerations when dimensioning an air supply system is usually to avoid velocities in the occupied zone that are too high, but as a rule it is not the air stream itself that reaches us there.

In the occupied zone we are more likely to be exposed to high velocities in the return air stream: see the figure below.

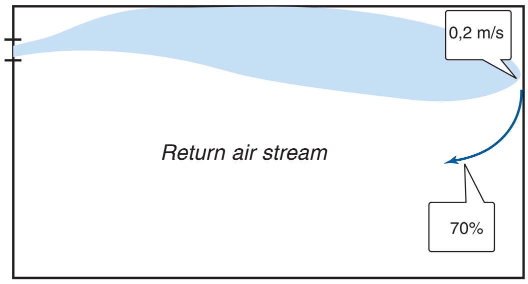


Figure 15. Return air stream with a wall-mounted diffuser

It has been shown that the velocity of the return air stream is approximately 70% of the velocity it had when it reached the wall. This means that a diffuser or valve fitted on the rear wall, with an end velocity of 0.2 m/s, will cause an air velocity of 0.14 m/s in the return air stream. This is within the limits for comfort ventilation, which is understood to mean that the velocity should not exceed 0.15 m/s in the occupied zone.

The throw for the diffuser or valve described above is the same as the length of the room, and in this instance is an excellent choice. A suitable throw for wall-mounted ventilation is somewhere between 70% and 100% of the room's length.

Effective penetration

The most common method for selecting the correct directional supply-air terminal device is to consider the throw $l_{0,2}$. But since the desired end velocity in the air stream depends on both the room's geometry and the required air velocity in the occupied zone, this can sometimes be rather misleading. Therefore the concept of the air stream's effective penetration has been introduced instead.

The effective penetration is the distance to the point where an end velocity is to be calculated. This can be the distance along the centre of the air stream from the diffuser itself to the furthest point in the room where the supply air is required. For wall-mounted diffusers, this means that the effective penetration is the same as the room's depth, while for ceiling diffusers the penetration is half the room's depth.

The velocity of the return air stream is approximately 30% slower than the air stream's velocity when it meets the wall. If the maximum air velocity in the occupied zone is to be 0.18 m/s, this means that the air stream must have a maximum velocity of 0.26 m/s when it meets the wall.

Effective penetration – calculation

The velocity at the effective penetration depth of a diffuser can be calculated theoretically by using the formula for calculating air velocity.

$$v_x = v_0 \cdot K \cdot \frac{\sqrt{A_{eff}}}{x_v}$$

where

v_x = velocity at the effective penetration (m/s)

v_0 = velocity at the diffuser outlet (m/s)

K = the diffuser coefficient

A_{eff} = the vent's effective outlet area (m²)

x_v = the effective penetration (m)

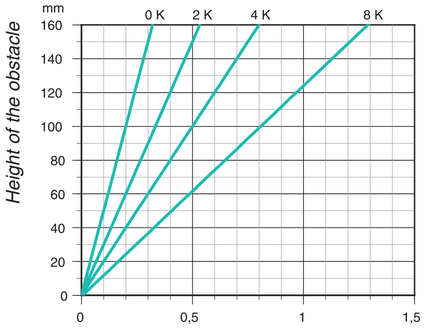
This method enables one to dimension the ventilation system more precisely than is possible when only using the throw data, and is therefore frequently used in different diffuser selection programmes.

Throw data for isothermal air

Rear-wall diffuser and wall-mounted diffuser:
0.7 to 1.0 x room depth.

Ceiling diffuser (supply air blown horizontally):
0.5 x room depth
(with rectangular rooms, the distance is calculated to the nearest wall).

Distance to an obstacle (estimate)
The diagram shows the minimum distance to the obstacle as a function of the obstacle's height (h in figure 17) and the air stream's temperature at the lowest point.



The penetration of the air stream
The shape of the room can affect the flow picture. If the cross section of the air stream is more than 40% of the cross section of the room, all induction of air in the room will stop. As a result, the air stream will deflect and start to suck in the induction air itself. In such a situation it does not help to increase the velocity of the supply air, as the penetration will remain the same while the velocity of both the air stream and the ambient air will increase.

Other air streams, secondary vortices, will start to appear further into the room where the main air stream does not reach. However, if the room is less than three times as long as it is high, it can be assumed that the air stream will reach all the way in.

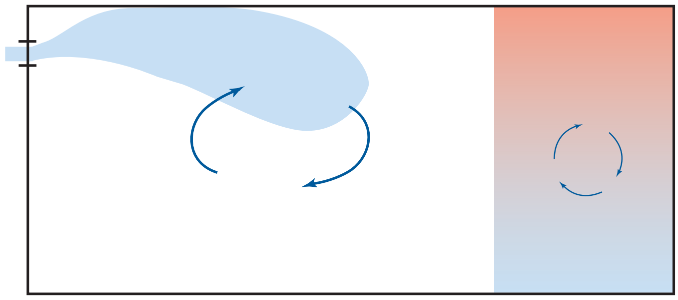


Figure 16. Secondary vortices are formed at the furthest point in the room, where the air stream does not reach.

Avoid obstacles
Unfortunately, it is very common for the air stream to be obstructed by light fittings on a ceiling. If these are too close to the diffuser and hang down too far, the air stream will deflect and descend into the occupied zone. It is therefore necessary to know what distance (A in the diagram) is required between an air supply device and an obstacle for the air stream to remain unimpeded.

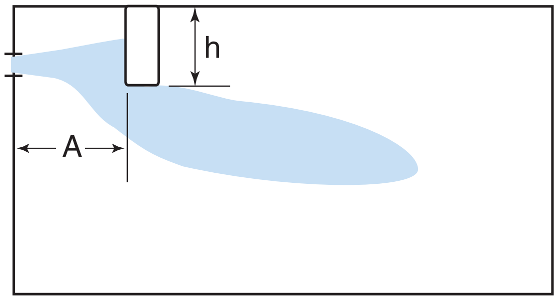


Figure 17. Minimum distance to an obstacle

Installing several directional supply-air terminal devices
If a single ceiling diffuser is intended to service an entire room, it should be positioned as close to the centre of the ceiling as possible, and the total surface should not exceed the dimensions indicated in Figure 18 below.

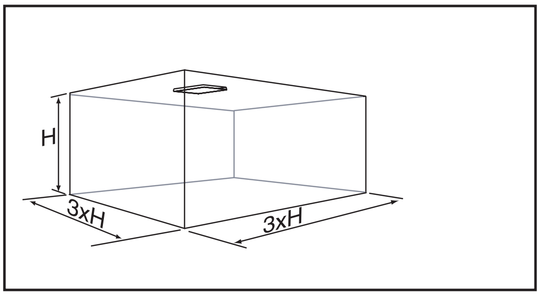


Figure 18. A small room ventilated by a single ceiling diffuser

If the room is larger than this, it usually has to be divided into several zones, with each zone ventilated by its own diffuser.

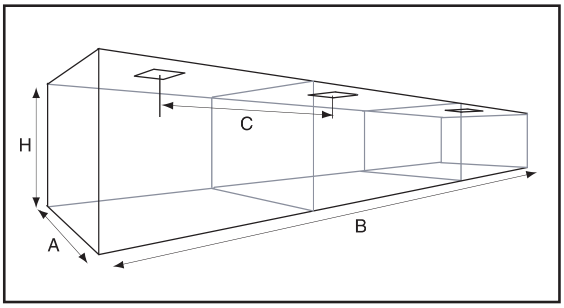


Figure 19. A large room ventilated by several ceiling diffusers

A room which is ventilated by several wall-mounted diffusers must also be divided into several zones. The number of zones is determined by the requirement to ensure sufficient distance between the diffusers to prevent the air streams affecting each other. If two air streams mix together, the result will be one stream with a longer throw.

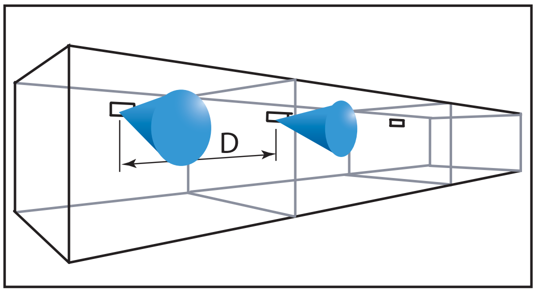


Figure 20. A large room ventilated by several wall diffusers

Dimensioning with several ceiling diffusers
A large room has to be divided into several zones. The maximum dimension for each zone is 1.5 x the room's length (A), as long as this does not exceed 3xH (see figure 18).

The appropriate throw is 0.5 x C, where C = the distance between two diffusers, (see figure 19).

Example
A large room (see figure 19) has the following dimensions:
H = 3 m
A = 4 m
B = 16 m

1) *How many zones should the room be divided into?*
2) *What will be distance be between the diffusers?*

1) The maximum size for each zone is 1.5 x A = 6 m, which means that the room should be divided into three zones, each 5.33 m long.
2) If the diffuser is placed in the centre of each zone, the distance (C) will be 5.33 m.

Dimensioning with several wall diffusers
The smallest distance between two wall-mounted valves or diffusers (D in figure 20) is 0.2 x $l_{0.2}$.

The appropriate throw is between 0.7 and 1.0 x A, where A = the depth of the room.

Example
A room which is 5 m deep is ventilated from the rear wall by means of diffusers with a throw of 4 m.

1) *What distance should there be between two diffusers?*

0.2 x $l_{0.2}$ = 0.2 x 4 = 0.8 m

1) There should be 80 cm between two diffusers.

Blowing in warm air
Blowing supply air horizontally from the ceiling works excellently for most rooms, including those with very high ceilings. If the supply air is above ambient temperature and also used to heat the premises, practical experiments have shown that this works well in rooms with ceiling heights of no more than 3.5 metres. This assumes that the maximum temperature difference is 10-15°C.

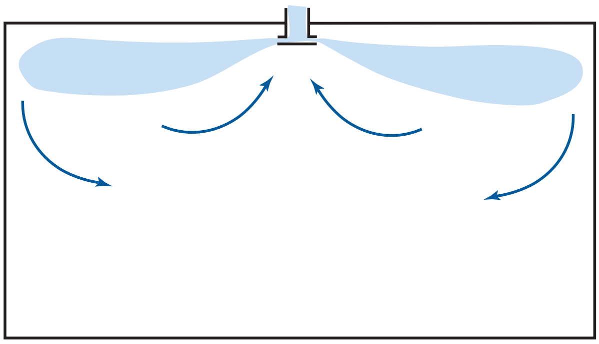


Figure 21. Blowing supply air horizontally from a ceiling diffuser

In very high rooms, however, the supply air has to be jetted vertically if it is also used for heating. If the temperature difference is no more than 10°C, the air stream should flow down to approximately 1 metre above the floor in order to produce a satisfactory evenness of temperature in the occupied zone.

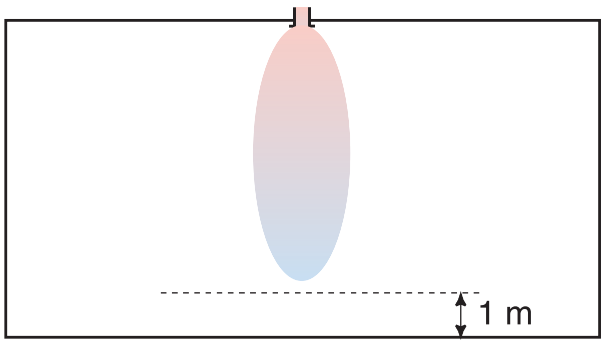


Figure 22. Supply air blowing vertically from a ceiling diffuser

Blowing in cold air
When supplying air that is colder than the ambient air, it is particularly important to make use of the Coanda effect to prevent the air stream from falling down into the occupied zone too early. The ambient air will then be sucked in and mixed more effectively, and the temperature of the air stream will have a better chance to increase before it reaches the occupied zone.
If the sub-ambient-temperature air is directed along the ceiling in this way, it is also important that the air stream velocity is high enough to ensure that there is sufficient adherence to the ceiling. If the velocity is too low there is also a risk that the thermal energy will push the air stream down towards the floor too early.
At a certain distance from the supply-air diffuser, the air stream will in any case separate from the ceiling and deflect downwards. This deflection occurs more rapidly in an air stream that is below the ambient temperature, and therefore in such cases the throw will be shorter.

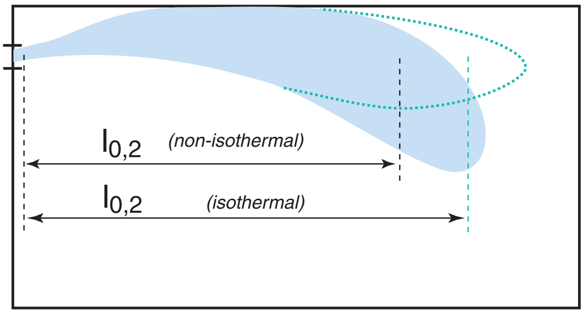
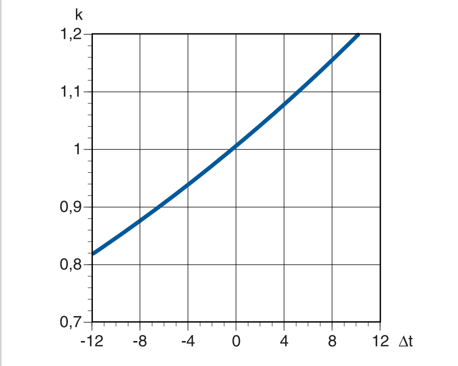


Figure 23. The difference between the throws of isothermal and non-isothermal air streams.

The air stream should have flowed through at least 60% of the room's depth before separating from the ceiling. The maximum velocity of the air in the occupied zone will thus be almost the same as when the air supply is isothermal.
The method for calculating where the air stream will separate from the ceiling is explained in the paragraph headed 'Non-isothermal air' on page 518.
When the supply air is below ambient temperature, the ambient air in the room will be cooled to some extent. The acceptable degree of cooling (known as the maximal cooling effect) depends on the air velocity requirements in the occupied zone, the distance from the diffuser at which the air stream separates from the ceiling, and also on the type of diffuser and its location.
In general a greater degree of cooling is accepted from a ceiling diffuser than a wall-mounted diffuser. This is because the air from a ceiling diffuser spreads in all directions, and therefore takes less time to mix together with the ambient air and to even out the temperature.

Correction of throw (estimate)
This diagram can be used to obtain an approximate value for the throw of non-isothermal air.

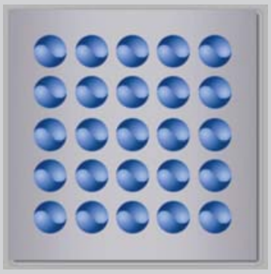


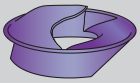
$l_{0.2}$ *(corrected) =* $k \cdot l_{0.2}$ *(isothermal air)*

Maximum acceptable cooling effect
A rule of thumb for the maximum acceptable cooling effect (Q_{max}) is:

Supply air blown from rear wall
Q_{max} = 20-40 W per m² floor surface at Δt 8K

Supply air blown from ceiling
Q_{max} = 60-100 W per m² floor surface at Δt 12K





The nozzles on the new Sinus series have been specially designed to provide the fastest possible mixing of supply air with ambient.

Selecting the correct supply-air terminal device

A supply-air terminal device for ventilation by diffusion can be fitted on either the ceiling or the wall. Diffusers are often equipped with nozzles or perforations which facilitate the admixture of ambient air in the air stream.

Nozzle diffusers are the most flexible devices because they allow individual fitting of each nozzle. They are ideal for supplying air that is well below ambient temperature, particularly if they are fitted in the ceiling. The throw pattern can be altered by turning the nozzles in different directions.

Perforated diffusers have a positive effect where the air stream temperature is significantly below that of the ambient air. They are not as flexible as nozzle diffusers, but by shielding off the air supply in different directions it is still possible to change the distribution pattern.

Wall-mounted grilles have a long throw. They have limited possibilities for altering the distribution pattern, and they are not particularly suitable for the supply of air that is below ambient air temperature.

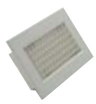
	Ceiling			Wall		
	 Nozzle diffuser	 Perforated diffuser	 Conical air distributor	 Nozzle diffuser	 Perforated diffuser	 Grille
Short throw	x	x	(x)	x	x	
Long throw	x			x		x
Flexible distribution pattern	x	(x)	(x)	x		
Sub ambient temperature air	x	(x)		x	(x)	

Table 3. Comparison of the different types of directional supply-air terminal device.