Technology

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Terms

Density
Density (specific weight) expresses mass per unit of volume. The unit 1 kg/m^3 is used for gases.

Power
International unit 1 Watt (1 W). The unit is used for all forms of power, e.g. electric power, mechanical power, heating power. Mechanical power in the unit 1 kW will have the same magnitude as the power expressed in the unit 1 hp (1 hp = 0.736 kW). However, heating power in the unit 1 kW will have a completely different magnitude than the previously used 1 kcal/h (1 kW = 860 kcal/h).

Energy
The international unit is 1 Joule (1 J). 1 J = 1 Ws = 1 Nm. In continuation, this unit was used for heat energy, among other things. 1 kcal = 4186 J or 1 kcal = 4.186 kJ. 1 J= 2.38889 $\times 10^{-4}$ kcal. For electric energy, normally the unit 1 kWh (1kWh = 3,600,000 Ws) is used.

Flow
Flow is expressed per time unit of 1 second (1 s). Volume per time unit is m^3/s; to a great degree, this numerical value deviates from the previously used m^3/h. Roughly speaking, 1 m^3/h = 2.8 × 10^{-4} m^3/s.

Mass – weight – force
The international unit of mass is 1 kilogram (1 kg). The kg unit shall only be used for specifying the material content in a body, i.e. mass. This is unchanged howsoever the body is moved on the Earth or in space. The word weight should be avoided as a synonym for mass in cases where there is a risk for misunderstanding.

Weight refers to the effect of gravity on the mass and is thus not a synonym for mass. The weight of a body will vary if it is moved between different places on Earth.

In a satellite where g = 0, no gravitation is exerted on the body (not weightless), but the body still has its mass.

The international unit of force is 1 Newton (1 N). 1 N is the force required to give 1 kg of mass an acceleration of 1 m/s^2. This unit does not match the previous unit of force 1 kp (1 kgf). In most cases, 10 N can be set to ≈ 1 kp.

Temperature
The unit for absolute temperature is 1 Kelvin (1 K). The temperature above the melting point of ice is set to 1 degree Centigrade (1°C).

Temperature difference is specified in the unit of 1 degree (internationally, however, as 1 deg). The unit 1 deg specifies the temperature difference of 1°C or 1 K. Degree shall always be in the singular.

Pressure
Pressure is force per unit of area. The unit for pressure is Pascal, Pa. 1 Pa = 1 Newton per square metre (1 N/m^2). I some cases, this unit provides impractically large numerical values. Then the unit of 1 bar = 100 kPa can be used to advantage. The pressure increase in fans, like the pressure fall in ducts, valves, etc., has previously been specified in the unit 1 mm vp = 1 kp/m^2. The numerical value for pressure in the new unit is almost 10 times as large: 1 mm vp = 9.81 Pa. In many cases, an error level of 2% can be allowed and then it can be appropriate to use 10 Pa = 1 mmwc. The unit 1 millibar (1 mbar) is used for barometric pressure. This measure is used within meteorology.

Rotational speed
The unit in the SI system for rotational speed is 1 radian per second (1 rad/s).

This unit provides completely divergent concepts with respect to the unit 1 revolution/minute. 1 rpm = 2π/60 rad/s. The transition to the unit 1 rad/s will be made when the motor manufacturers introduce this unit.

Terms and conversion factors

Conversion factors

The table consists of a selection of the most common magnitudes within fan and air treatment technology. Where appropriate, the conversion factors are shortened to three decimals.

For practical use, applicable approximations with an error level of no more than 2% are provided within parentheses.

Magnitude	Term	SI unit	Previous	Conversion factor unit	
Force	**F**	N	kp	1 N = 0.102 kp (1 N ≈ 0.1 kp)	1 kp = 9.807 N (1 kp ≈ 10 N)
Pressure	**p**	Pa	mm wc	1Pa = 0.102 mm wc (1 Pa ≈ 0.1 mm wc)	1mm wc = 9.807 Pa (1 mm wc 10 Pa)
		bar	kp/cm^2	1 bar = 1.020 kp/cm^2 (1 bar ≈ 1 kp/cm2)	1 kp/cm^2 = 0.981 bar (1 kp/cm^2 ≈ 1 Bar)
		mbar	dry[1)]	1 mbar ≈ 0.750 dry (1,000 mbar ≈ 760 mm Hg)	1 dry ≈ 1.333 mbar
Flow	**q**	m^3/s	m^3/h	1 m^3/s = 3,600 m^3/h	1 m^3/h = 0.278 × 10-3m^3/s (1,000 m^3/h ≈ 0.28 m^3/s)
Power	**P**	kW kW	hk kcal/h	1 kW = 1.360 hp 1 kW = 860 kcal/h	1 hp = 0.736 kW 1 kcal/h = 1.163 × 10^{-3}kW
Energy	**W**	kJ	kcal	1 kJ = 0,239 kcal	1 kcal = 4,187 kJ
Enthalpy	**i**	kJ/kg	kcal/kg	1 kJ/kg = 0.239kcal/kg	1 kcal/kg = 4.187 kJ/kg
Specific heat	**cp**	kJ/kg deg	kcal/kg°C	1 kJ/kg deg= 0.239 kcal/kg°C	1 kcal/kg°C = 4.187 kJ/kg deg
Thermal conductivity coefficient	**λ**	W/m deg	kcal/m°C	1 W/m deg= 0.860 kcal/m°C h	1 kcal/m°C h = 1.163 W/m deg
Thermal conductance coefficient	**k**	W/m^2 deg	kcal/m^2°C h	1 W/m^2 deg= 0.860 1 kcal/m^2°C h	1 kcal/m^2°C h = 1.163 W/m^2 deg

1) 1 dry= 1 mm Hg at 0°C and g = 9.80665 m/s^2.

Formula collection

Air flow, q m³/s

$q = A \cdot v$

A = cross-sectional area, m^2

v = air speed, m/s

Dynamic pressure, p_d Pa

$$pd = \frac{\rho \cdot v^2}{2}$$

ρ= air density, kg/m^3

v = air speed, m/s

Hydraulic diameter, d_h m

$$d_h = \frac{4 \cdot A}{O}$$

A = cross-sectional area, m^2

O = duct circumference, m

d_h for rectangular duct

$$d_h = \frac{2 \cdot a \cdot b}{a + b}$$

a and b are the sides of the duct

d_h for circular duct

$d_h = d$ = duct diameter

Total pressure drop – supply air, p_t Pa

$p_t = ps + pd$

ps = static pressure drop, Pa

pd = dynamic pressure drop, Pa

Total pressure drop – exhaust air, pt Pa

$p_t = (- p_s) + p_d$

p_s = negative static pressure drop, Pa

p_d = dynamic pressure drop, Pa

Cross-sectional area circular duct, A m²

$$A = \frac{\pi \cdot d^2}{4}$$

d = duct diameter, m

Circumference of circular duct, O m

$O = \pi \cdot d$

d = duct diameter, m

Air density, kg/m³

$$\rho_t = 1.293 \cdot \frac{B}{1,013} \cdot \frac{273}{273 + t}$$

B = barometric pressure, mbar

t = air temperature,°C

Cooling/heating effect, P kW

$P = q \cdot \rho \cdot c_p \cdot \Delta t$

q = air flow, m^3/s

ρ = air density, kg/m3

c_p = the air's specific heatcapacity, kJ/kg,K (≈1.0)

Δt = temperature difference,°C, between exhaust and supply air

Fans – general

Audio technology glossary

Absorption
Reduction of sound energy (conversion to thermal energy in absorbent material).

A-weighted sound level
Sound pressure level determined with sound level meter with connected A filter. Written as dB(A).

Decibel
The unit for a logarithmic function of a certain magnitude. (Often used for the logarithmic function of sound pressure and acoustic power, but also in completely different contexts.)

Equivalent sound absorption area
The equivalent sound absorption area of a room is a measurement of the area of the confining surfaces multiplied by their average absorption capacity.

Frequency
In an acoustic context, frequency is the number of pressure fluctuations per second. Frequency is expressed in the unit Hertz (Hz).

Acoustic power, acoustic power level
The acoustic power, measured in W, is the power that is added to the air and causes pressure fluctuations (sound). The logarithmic function is called the acoustic power level and most often has the unit dB. The unit B is also used sometimes (1 B = 10 dB).

Sound pressure, sound pressure level
Sound pressure, measured in Pa, is a measurement of the magnitude of the pressure fluctuations in the air. The logarithmic function is called the sound pressure level and has the unit dB.

Octave band
A standardised division in frequency ranges. The octave bands are named after their centre frequencies.

Total acoustic power level, $L_{w,tot}$
The logarithmic total of the acoustic power level in octave band 125–8,000 Hz. Used as an initial value for calculating acoustic power in an octave band when reporting sound production.

A fan is designed to achieve a transport of the flow of air or other gas.

In order to bring about a flow, e.g. in a duct system, a pressure increase in the gas is required in an appropriate location in the system. The requisite pressure increase can be achieved with a fan or - in those cases a particularly large pressure increase is required - with a compressor.

Terms

q	=	gas flow at fan inlet	m^3/s (m^3/h)
Δp_t	=	total pressure increase between the fan's connections	Pa (mm wc)
p_d	=	dynamic pressure in fan outlet	Pa (mm wc)
p_a	=	absolute pressure	Pa (mm wc)
T	=	absolute temperature	K
n	=	fan rotational speed	rpmin
P_u	=	theoretic effect	kW
P_r	=	impeller power requirement	kW
P_e	=	active electric power needs from the grid	kW
L	=	work line or number of such	
v	=	gas speed in fan outlet	m/s
η_r	=	degree of efficiency of impeller	%
η_e	=	total degree of efficiency for the fan	%
δ	=	density of gas	kg/m^3

Mode of operation

In a fan, energy is added to a flowing mass of gas via one or more impellers equipped with blades. When passing through the impeller(s), normally both the dynamic and static pressures of the gas increase.

The impeller outlet speed is mostly converted to static pressure during passage from the impeller outlet to the fan outlet.

In radial fans, this conversion from velocity energy to static pressure takes place in the spiral-shaped cowl. Generally, fans that are connected to duct systems have the same connection area on the inlets and outlets. When, in such cases, the gas velocity and consequently the dynamic pressure are the same in the fan's connections, the total pressure increase in the fan will be perceived as an increase in the static pressure between the fan's connection flanges.

A free-standing suction fan sucks air from premises where both the static pressure and velocity are 0, and supplies air in the fan outlet at a specific speed and increased static pressure. Thus, in this case, the fan's total pressure increase is perceived as an increase in both static and dynamic pressure.

Definition of degrees of efficiency for fans

Degree of efficiency of impeller:

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

Total degree of efficiency of the fan:

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

where P_u is a theoretical effect according to

$$P_u = \frac{q \times \Delta p_t}{1{,}000} \text{ kW}$$

where q is stated in m^3/s and Δp_t in Pa.

The effect of the rotational speed on fan capacity

For unchanged load ratios (unchanged choking), the following change:

1. The quantity of air in direct proportion to rotational speed

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

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

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

3. Power requirement in direct proportion to the cube of the rotational speed.

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

These formulae apply if the pressure drop is proportional to the square of the air flow.

Audio report

Fans

For fans in this catalogue, the generated acoustic power level is reported.

The report is made in eight octave bands for different audio paths. The value in each octave band is obtained by reading the total acoustic power level, $L_{w,tot}$, in the fan diagram and by correcting with the relevant correction factor, K_{ok}, according to the table in the fan diagram.

Measurements are carried out in accordance with ISO 3741 or ISO 5136.

ISO 3741 is used when measuring the acoustic power level to the surroundings of fans or units and ISO 5136 when measuring the acoustic power level to a duct.

If the measurement is carried out with a free-standing fan, this will result in a lower sound level. The ASHRAE trade organisation in the USA specifies in Application of Manufacturers Sound Data:

"When measuring sound, a free-standing fan will receive a 5-10 dB lower sound level in octave bands from 250 Hz and lower than a fan in the unit casing".

Measurement inaccuracy

In connection with developing their measurement method for the acoustic power to a duct, ISO has also studied the inaccuracy in different octave bands (90% certainty).

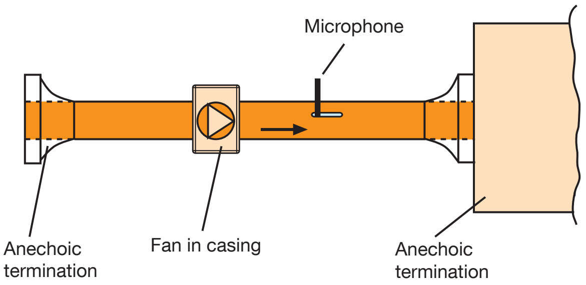
Octave band (Hz)	63	125	250	500
Inaccuracy (dB)	±5.0	±3.4	±2.6	±2.6
Octave band (Hz)	1,000	2,000	4,000	8,000
Inaccuracy (dB)	±2.6	±2.9	±3.6	±5.0

Sound damping products

For sound dampers and other sound damping products, input attenuation Δ is reported.

The input attenuation is measured in accordance with ISO 5136.

ISO method



Measurement is made inside a duct with a specified layout and reflection-free connection. Measurements and calculations are made in a 1/3 octave band.

Aids for sound calculations

Room absorption

The volume of the room, the nature of the surfaces and fittings and fixtures affect the resulting sound level to a large extent. In order to calculate a room's equivalent absorption area, a table with approximations for the absorption factor α and diagram can be used.

In general, the room constant (R) is calculated as follows:

$$R = \frac{S \times \alpha_m}{1 - \alpha_m} \ (m^2)$$

where:

$S \times \alpha_m = S_1 \cdot \alpha_1 + S_2 \cdot \alpha_2 + \ldots\ldots.. + S_n \cdot \alpha_n$

S = the total confining area of the room (m^2)

$S_1 \ldots S_n$ = area of the partial areas (m^2)

$\alpha_1 \ldots \alpha_n$ = absorption factors of the partial areas

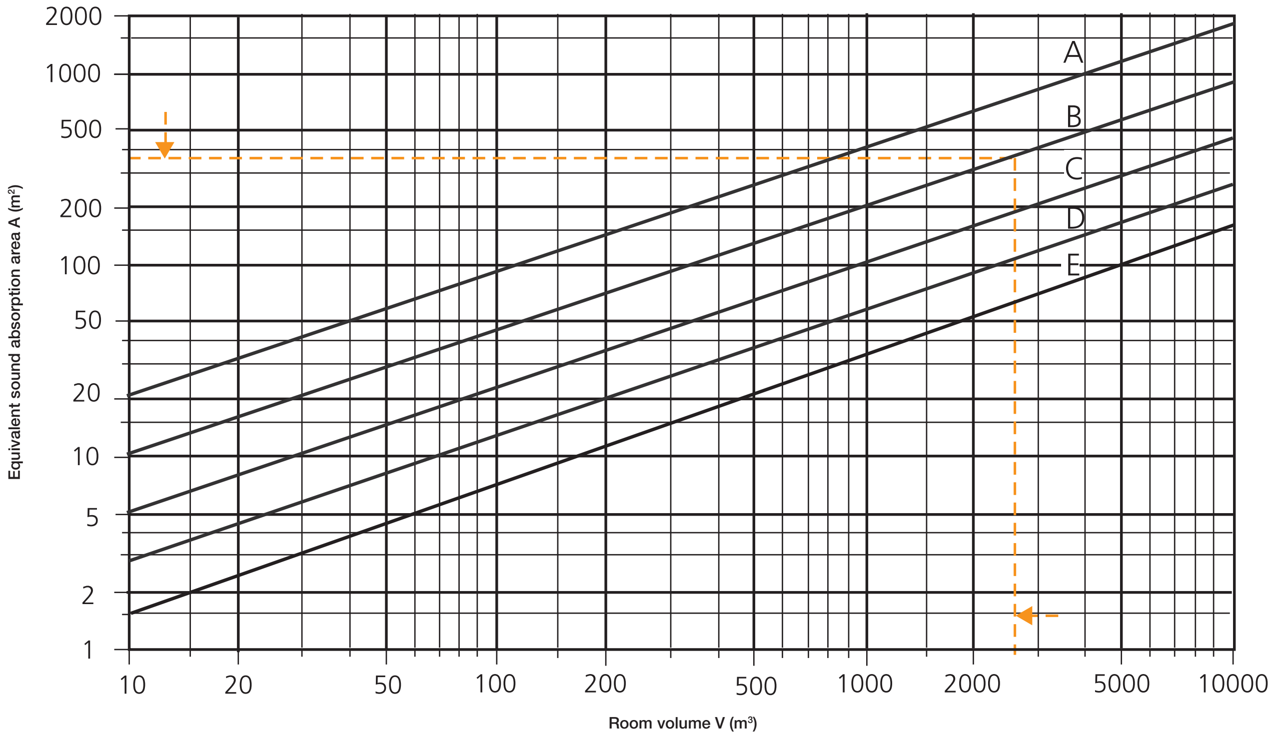
α_m = mean absorption factor for the total confining surface

The average absorption factors of different premises

Type of room	Average absorption factor α_m
Radio studio, music room	0.30–0.45
TV studio, department store, reading room	0.15–0.25
Houses, offices, hotel rooms, Conference premises, theatres	0.10–0.15
School rooms, nursing homes, small churches	0.05–0.10
Factory halls, swimming pools, large churches	0.03–0.05

Example (orange dashed line in diagram):

Shop premises for clothes with the dimensions 20 × 30 × 4.5 m (i.e. 2.700 m^3) having an average absorption fact α_m = 0.25. The equivalent room absorption of the premises is 350 m^2 .



A Strongly attenuated room α_m = 0.40

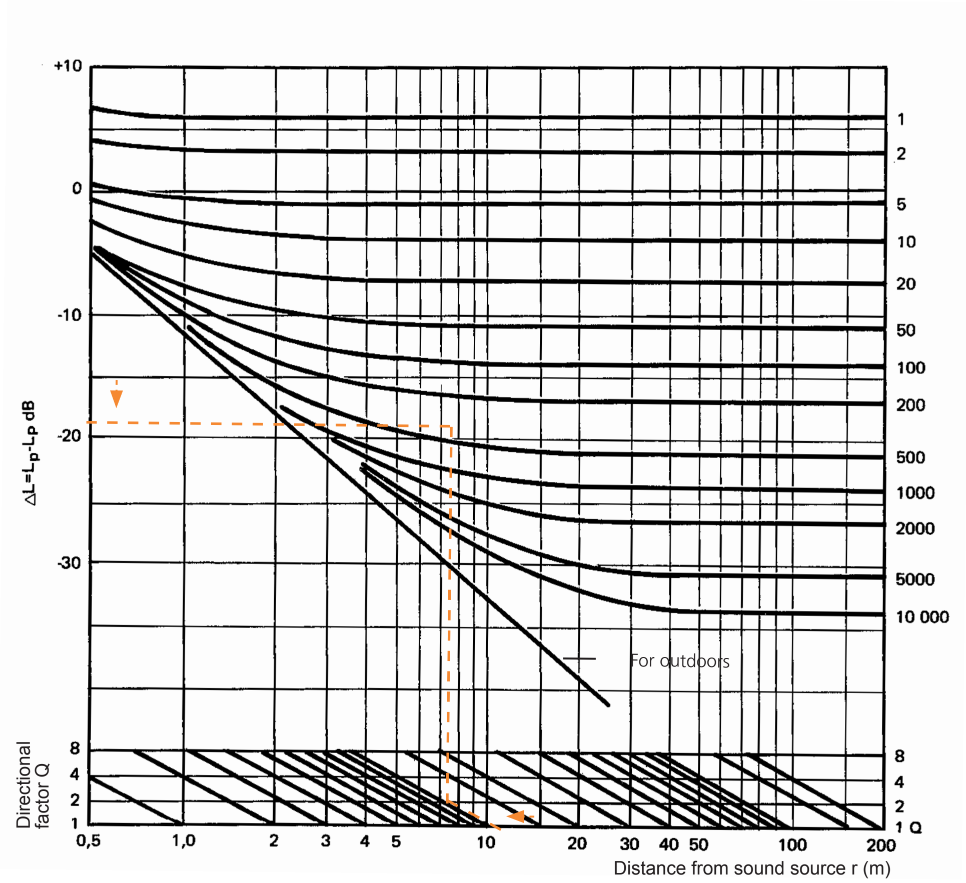
B Attenuated room α_m = 0.25

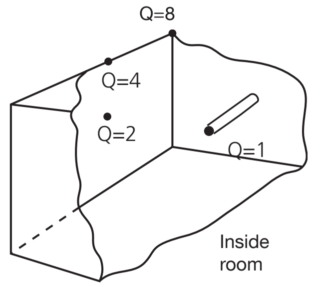
C Normal room α_m = 0.15

D Hard room α_m = 0.10

E Very hard room α_m = 0.05

Difference between sound pressure level and acoustic power level





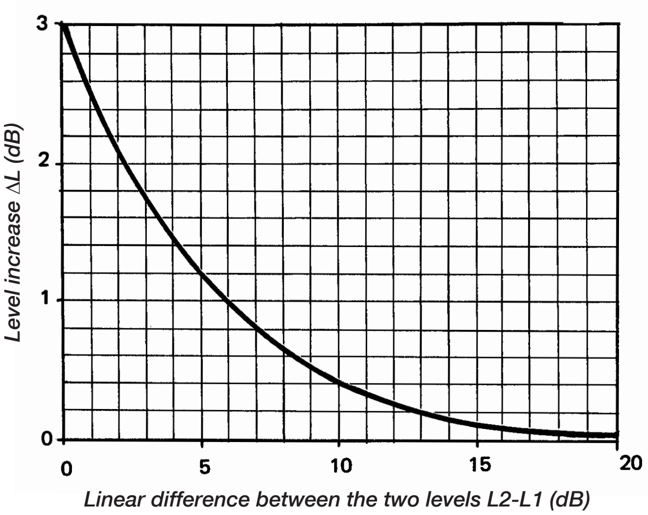
Direction factor, Q

Factors for the distribution pattern of a sound source. The factor is dependent on the location of the sound source in relation to reflecting surfaces.

The difference between the sound pressure level and the acoustic power level at the distance r from a sound source with the direction factor Q. The room's equivalent sound absorption area inserted as a parameter.

Example (orange dashed line): A distance of 10 metres from the sound source. Direction factor Q=2 (at wall). The equivalent room absorption of the premises 350 m^2 (according to example from previous page). The difference is -18 dB at a distance of 10 metres from the sound source.

Addition of two different levels



Weighting filter

Weighting filter, level values with tolerance for precision sound level meters. The values refer to the entire instrument in a free sound field.

Centre frequency octave bande	Curve A (dB)	Curve B (dB)	Curve C (dB)	IEC tolerance limit (±dB)
31.5	-39.4	-17.1	-3.0	1.5
63	-26.2	-9.3	-0.8	1.5
125	-16.1	-4.2	-0.2	1.0
250	-8.6	-1.3	0	1.0
500	-3.2	-0.3	0	1.0
1,000	0	0	0	1.0
2,000	+1.2	-0.1	-0.2	1.0
4,000	+1.0	-0.7	-0.8	1.0
8,000	-1.1	-2.9	-3.0	+1.5/-3.0
16,000	-6.6	-8.4	-8.5	+3.0

Sound level in room

Reverberation field

When measuring the sound level from installations, these are made in the reverberation field.

Sound measurement in the direct field is normally not performed. Among other things, this is because of the difficulty in stating what a direct field is. Is this 0.5 m from the sound source? 0.8 m? 1.5 m?

Consequently, the requirements for the sound level from installations are usually specified in the reverberation field. The reason for this is that the reverberation field is the only well-defined area for sound measurement.

The reverberation field begins where the room attenuation fully affects the sound level, i.e. when there is no longer any decay.

However, in principle one can count on a part of the transition zone also constituting a part of the reverberation field. Otherwise it would be difficult to perform measurement of the sound level in rooms where the reverberation field is very small.

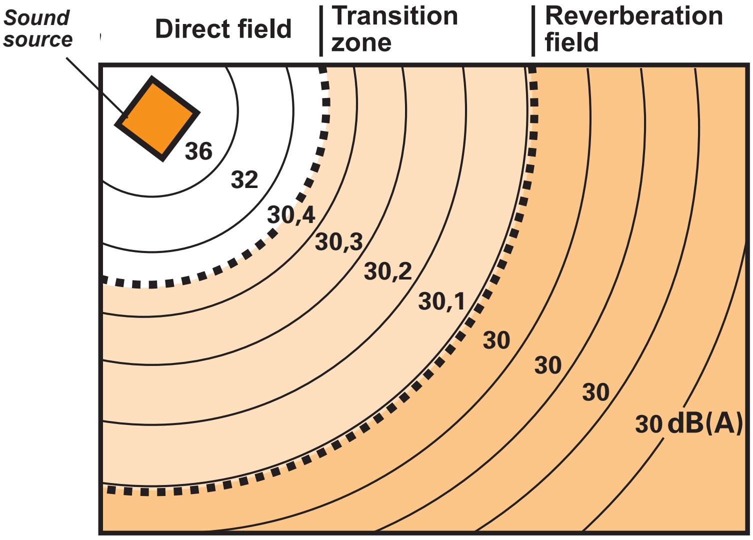
Sound level requirements

The overriding requirements in The Swedish Board of Building, Planning and Housing Regulations, BBR 99, are very stringent, but with limited indication of numerical values.

Largely speaking, one should specify that for nursing homes, youth leisure activities, day care centres, classrooms, working areas intended for offices and the like, the premises shall be designed so that disruptive sounds "are dampened" to the extent required by the activity and do not affect those working or are present in the premises".

Equivalent document for housing is specified.

The requirements are formulated differently compared to the previous ones and there may be opportunities for different interpretations. For this reason, no requirement values are reported in numbers.



Examples of how the sound pressure level can decay from the sound source in a room.

Fan data at deviating density

The diagrams and data for fans reported in this catalogue are for a density of 1.2 kg/m^3 at the fan inlet.

The density is 1.2 kg/m^3 for air with the temperature 20°C at a relative humidity of 50% and at sea level (1,013 mbar). The following relationship applies for converting fan data to another density.

1. The air flow in m^3/s does not vary with the density.
2. Static, dynamic and total pressure is obtained from:

   $p = p_{1.2} \times K_1 \times K_2$
3. The power requirement is obtained from:

   $P = P_{1.2} \times K_1 \times K_2$
4. The density is obtained from:

   $\rho = \rho_{1.2} \times K_1 \times K_2$

   where K_1 and K_2 are obtained from the adjoining diagram.

In many contexts, normal cubic metres are used, nm^3, or normal cubic metres per second, nm^3/s.

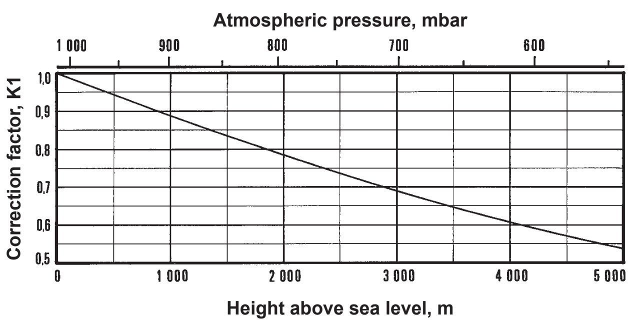
Normal cubic metre, nm^3, implies the amount of gas that has a volume 1 m^3 at a pressure of 1 bar and a temperature of 0°C.

Thus, the air flow expressed in nm^3/s is constant depending on whether the air is cooled or heated. Conversion from air flow expressed in nm^3/s to actual air flow in m^3/s is performed as follows:

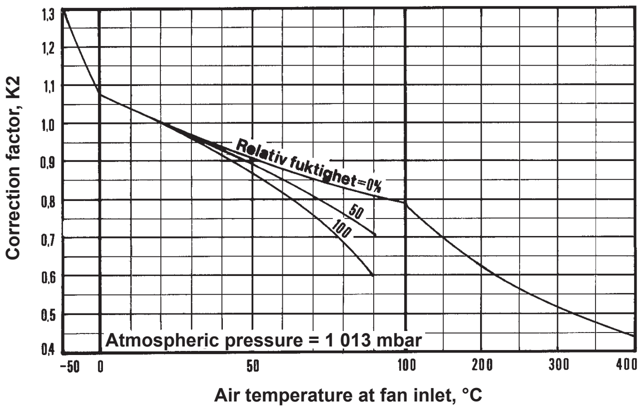
$$q = q_n \times \frac{1.06}{K_1 \times K_2}$$

where q_n is the air flow in nm^3/s.

Correction factor K_1



Correction factor K_2



Environmental classes

Environmental classes according to The Swedish Board of Building, Planning and Housing's Handbook on Steel Structures, BSK 99, based on SS-EN-ISO 12944-2:

Environmental class	Air aggressive-ness	Environmental
C1	Very low	Indoors in dry air, e.g. in heated premises.
C2	Low	Indoor air with fluctuating temperature and humidity and insignificant content of air impurities, e.g. in non-heated premises. Outdoors in areas with low amounts of of air impurities.
C3	Moderate	Indoors with moder-ate effect and moderate amounts of air impurities. Outdoors in areas with a certain amountsalt or moderate amounts of air impurities.
C4	High	Outdoors in air with moderate amount of salt or substantial amounts of air impurities. Indoors in areas with high humidity and a great amount of air impurities, e.g. swimming pools, industrial premises.
C5-I	Very high (Industrial)	Indoors with almost permanent and a large amount of air impurities. Outdoors industrial areas with high air humidity and aggressive atmosphere.
C5-M	Very high (Marine)	Indoors, see above. Out-doors by the coast and in offshore areas with large amounts of salt.

Earlier environmental classes

Translation from BSK 94 to BSK 99:

M0 corresponds to C1

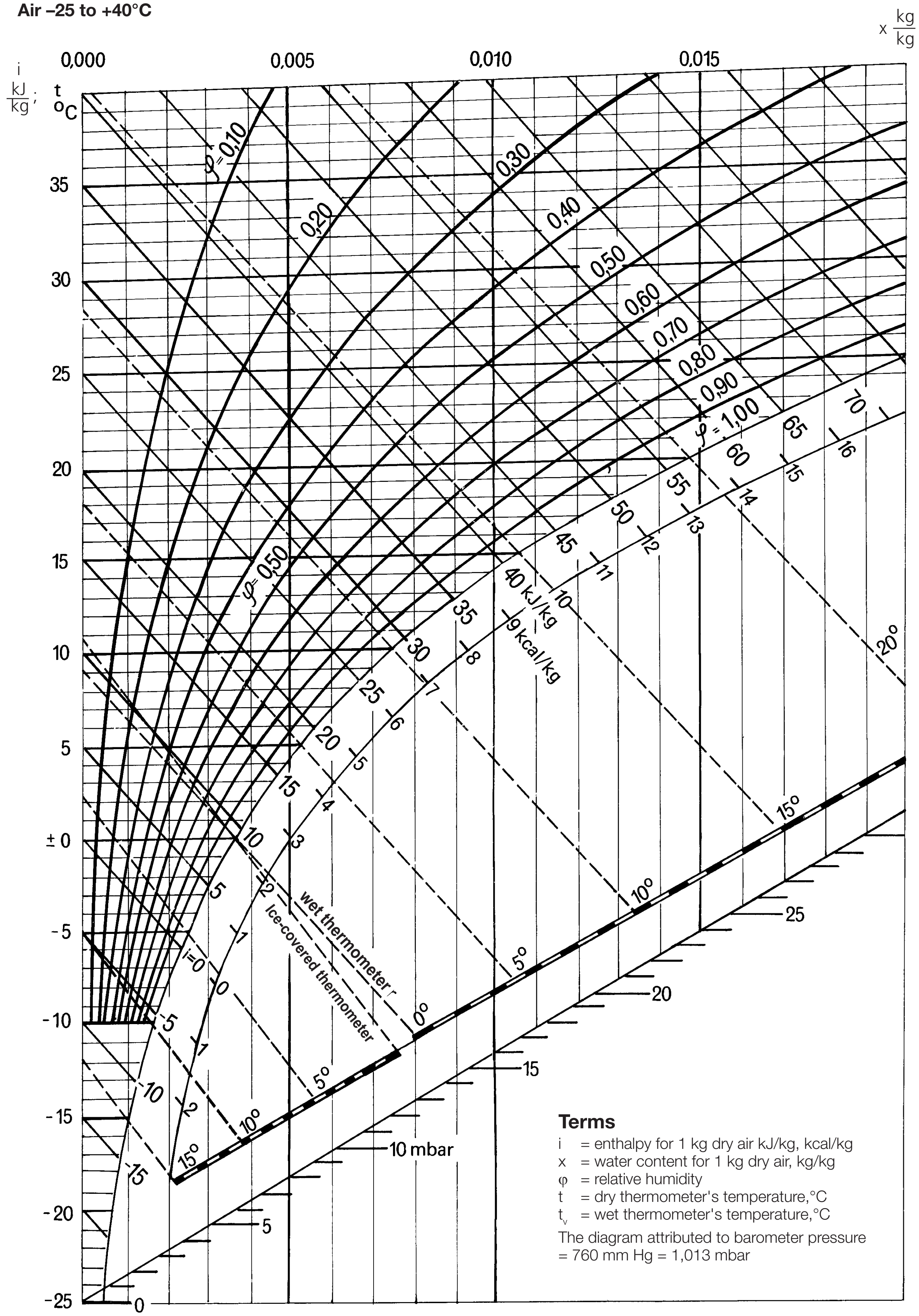
M1, M2 correspond to C2

M3 corresponds to C3, C4

M4 corresponds to C5

Mollier diagram for moist air

Air –25 to +40°C



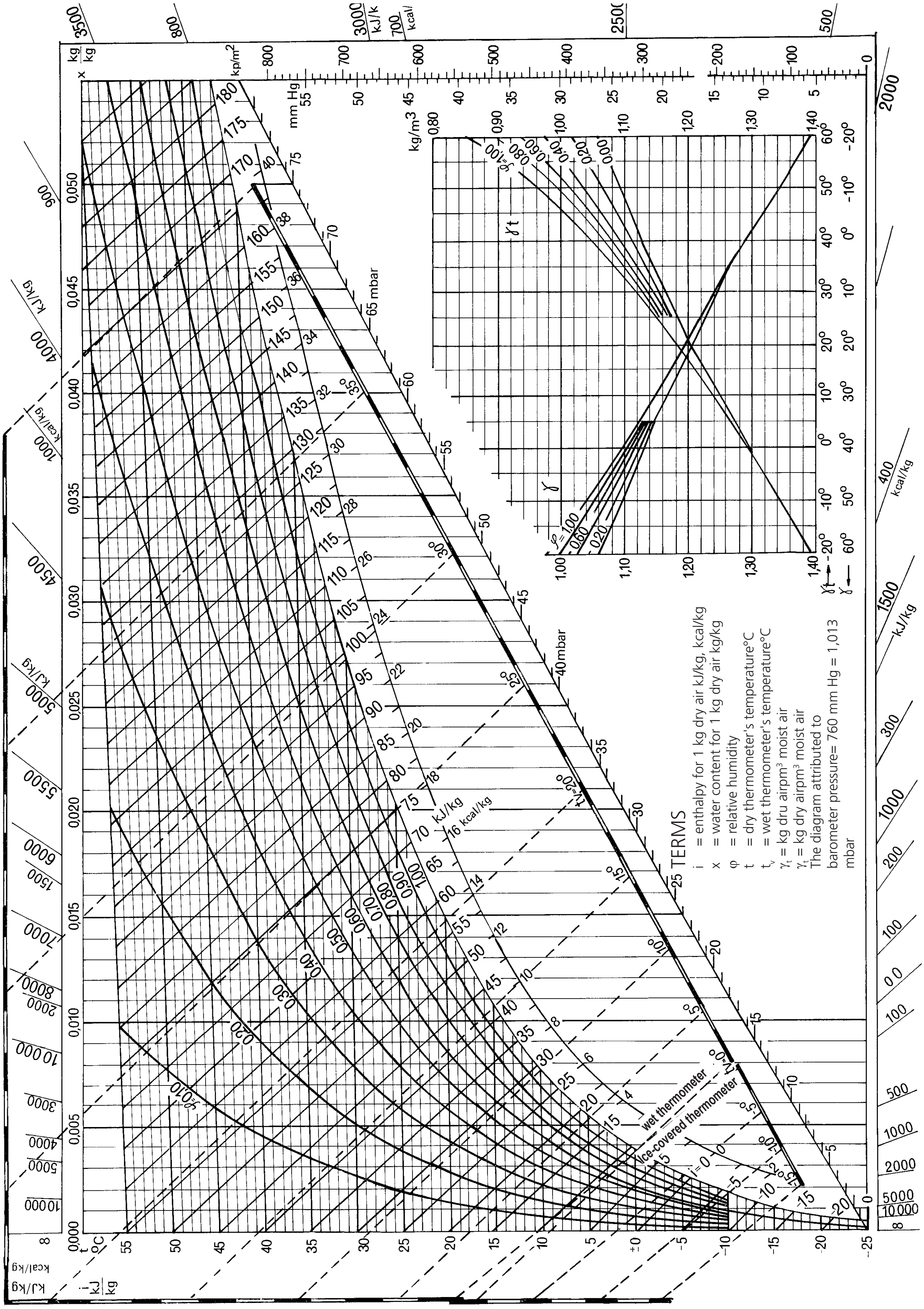
Terms

i = enthalpy for 1 kg dry air kJ/kg, kcal/kg
x = water content for 1 kg dry air, kg/kg
φ = relative humidity
t = dry thermometer's temperature,°C
t_v = wet thermometer's temperature,°C

The diagram attributed to barometer pressure = 760 mm Hg = 1,013 mbar

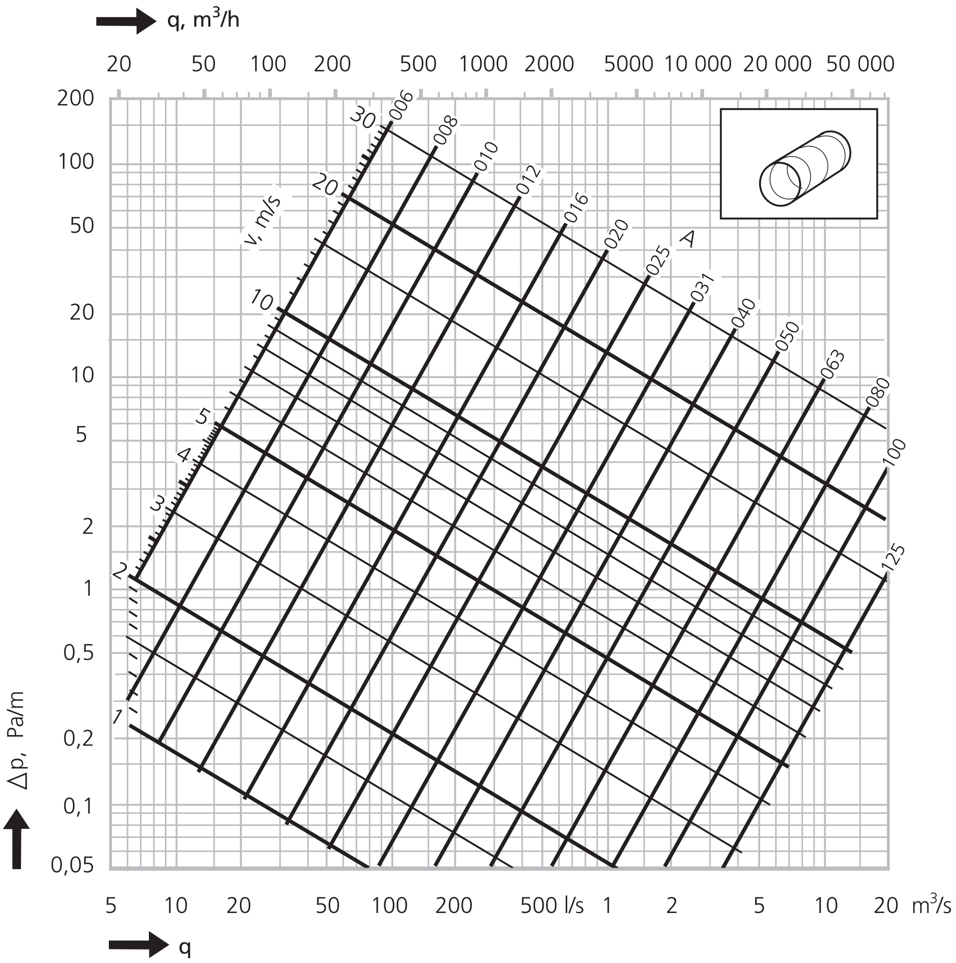
Mollier diagram for moist air

Air –25 to +55°C

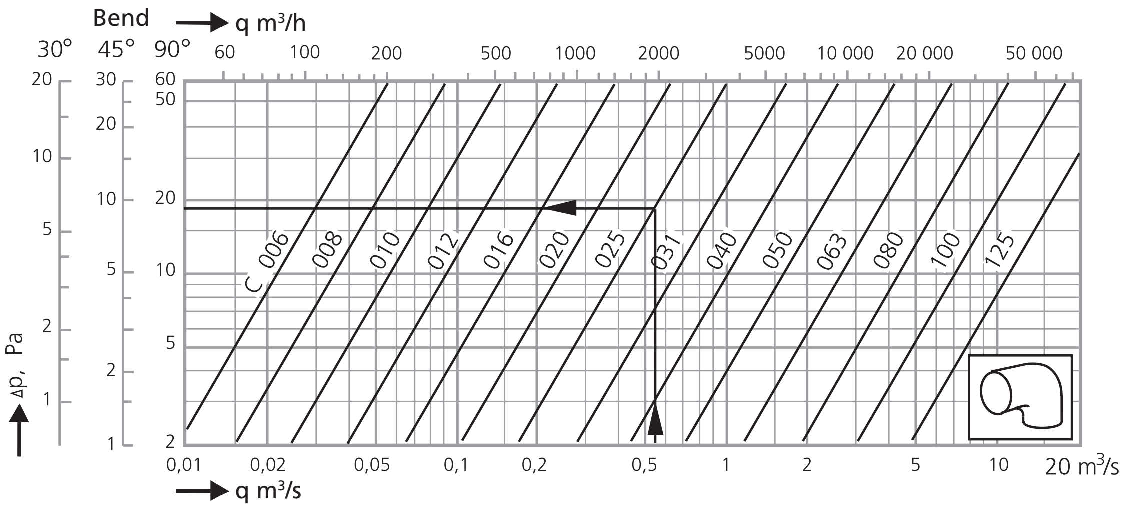


Pressure drop diagram ducts

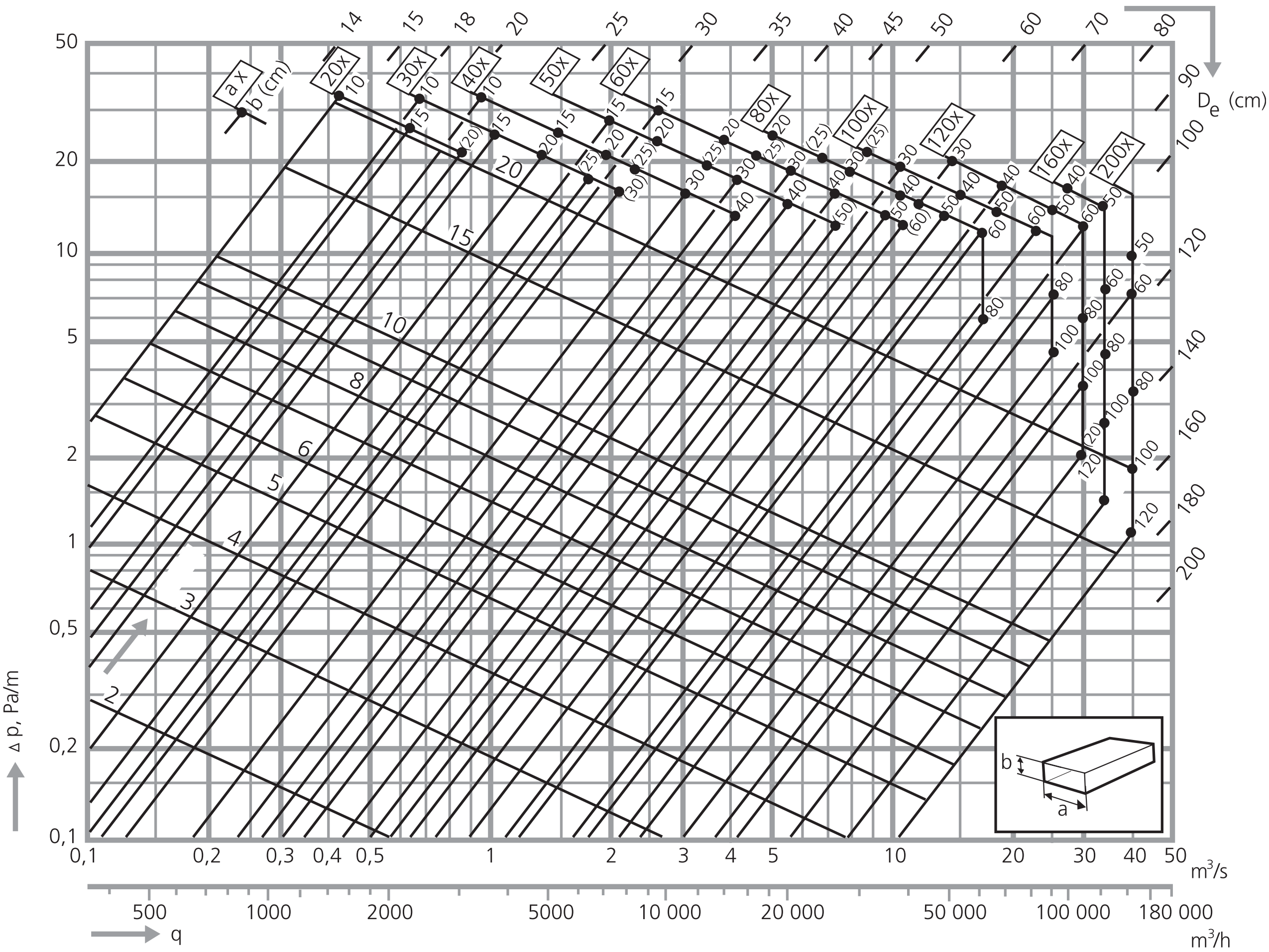
Circular ducts



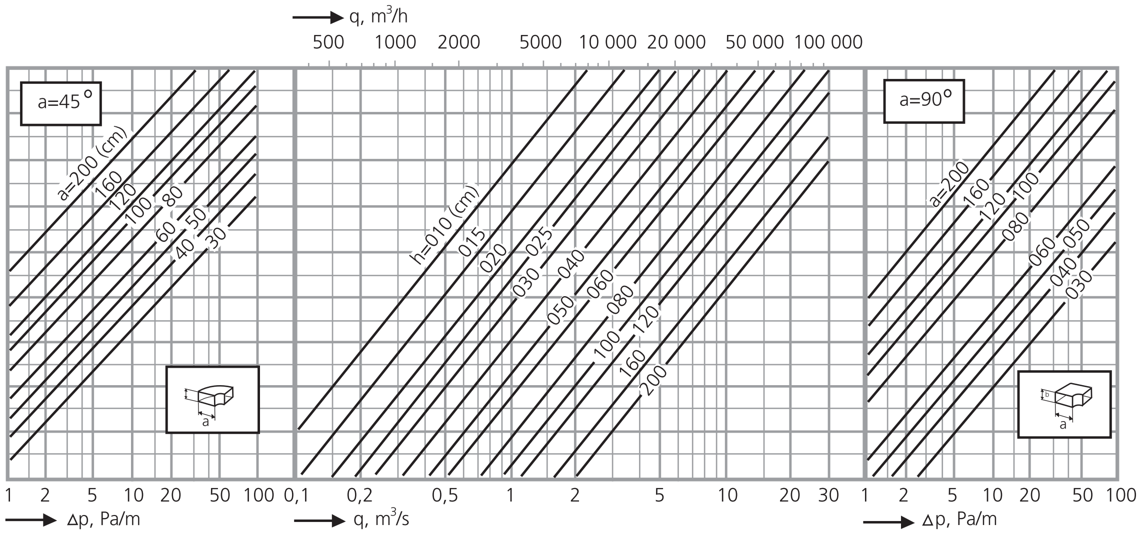
Circular bends



Rectangular ducts



Rectangular bends



Calculation examples

Radial fan FML, FKL, FAM, FAH

General

The diagrams apply for air with a density of 1.2 kg/m^3. In the pressure-flow diagram, the fan's recommended work area, in which VVA-AMA's requirements for the degree of efficiency are maintained, is marked with an orange field. In those cases the fan's performance including pressure drop before the fan inlet or after the fan's outlet are reported, degrees of efficiency will be specified in the diagram for a corresponding fan without these pressure drips. Throttle lines for even values for the degree of efficiency are plotted; they represent different installation characteristics and the pressure drop for these are proportional to the square of the air flow.

The total acoustic power level, $L_{W, tot}$ to a connected outlet duct is marked with orange curves and orange numbers. Corrections for different acoustic paths and octave bands are reported in table form.

Fan diagram, single-suction slow-pressure fans

The total pressure curves apply for a fan that is duct-connected on both inlet and outlet.

In the "System losses" diagram, the following are reported:

p_1 = The influx loss for a fan with a free-standing suction inlet and the duct connection outlet.

p_2 = The shock loss at the outlet (in addition to the dynamic pressure) for a fan with a duct-connected inlet and the outlet free-blowing or connected to a pressure chamber.

p_3 = The sum of the influx loss and shock loss at the outlet (in addition to the dynamic pressure) for a fan with a free-standing suction inlet and the outlet free-blowing or connected to a pressure chamber.

p_d = The dynamic pressure in the fan outlet.

The power curves show the net power requirement of the fan, excluding losses in the belt transmission and bearings.

Connection cases:

1. Fan with duct-connected inlets and outlets

The pressure-flow diagram applies for this connection case. The difference in dynamic pressure between the fan's outlet and inlet is added to the installation's static pressure drop P_{stat} before the fan's operating point is determined in the pressure flow diagram.

$p_{tot} = p_{stat} + (p_d - p_d$, inlet)

2. Fan with free-standing suction inlet and duct-connected outlet

Since the fan has free-standing suction, there is an influx loss p_1 that is shown in the "System losses" diagram. The influx loss p1 and the dynamic pressure in the fan outlet p_d are added to the installation's static resistance p_{sta} before the fan's operating position is determined in the pressure-flow diagram.

$p_{tot} = p_{stat} + p_1 + p_d$.

3. Fan with duct-connected inlet and the outlet is free-blowing or connected to a pressure chamber

Due to the uneven speed distribution in the fan outlet, in addition to the loss of the dynamic pressure p_d, there is a shock loss p_2 that is shown in the "System losses" diagram.

The shock loss p_2 and the difference in dynamic pressure between the fan's outlet and inlet is added to the installation's static pressure drop p_{stat} before the fan's operating point is determined in the pressure-flow diagram.

$p_{tot} = p_{stat} + p_2 + (p_d - p_d$, inlet).

For size FML 71-80, FKL 90-140, pd is = p_d inlet and

$p_{tot} = p_{stat} + p_2$

For FAM and FAH, $p_2 = 0$

4. Fan with a free-suction inlet and the outlet is free-blowing or connected to a pressure chamber

Due to the uneven speed distribution in the fan outlet, in addition to the loss of the dynamic pressure pd, there is also a shock loss. The sum of this shock loss and the influx loss at the inlet is symbolised by p3 and is shown in the "System losses" diagram.

The System loss p3 and the dynamic pressure in the fan outlet pd are added to the installation's static pressure drop before the fan's operating point is determined in the pressure-flow diagram.

$p_{tot} = p_{stat} + p_3 + p_d$

For FAM and FAH, $p_3 = p_1$ when $p_2 = 0$

The fan's data

Radial fan FML, FKL, FAM, FAH

Example

Radial fan FKLB-3-090, with free-standing suction inlet and the outlet connected to a duct.

Air flow = 5.2 m³/s.

p_{stat} = the sum of all pressure drops in the duct system = 1,110 Pa.

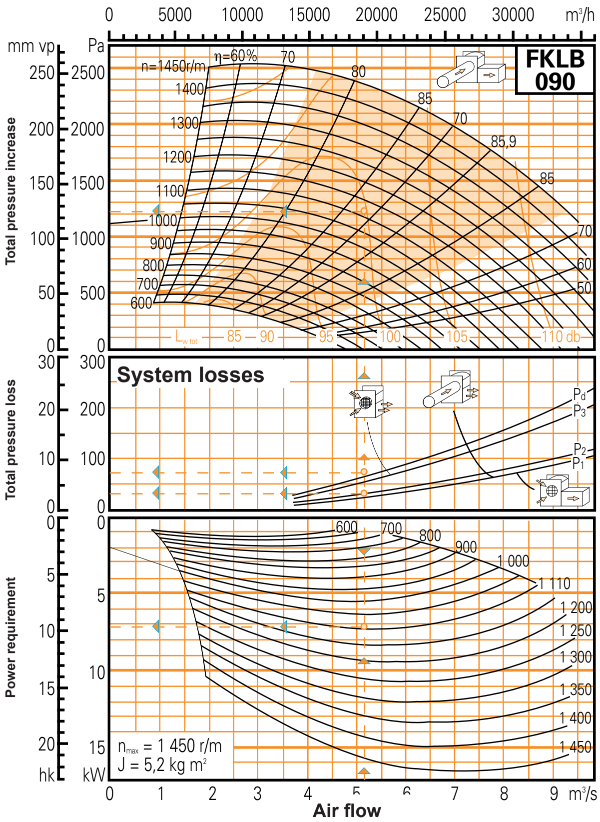
Enter the air flow at 5.2 m³/s and read the system loss P_1 off in the diagram for system losses to 30 Pa and the dynamic pressure P_d to 70 Pa.

Requisite total pressure: $p_{tot} = p_{stat} + p_1 + p_d = 1110 + 30 + 70 = 1{,}210$ Pa.

Continue in the upper diagram with the air flow 5.2 m³/s and the total pressure 2.10 Pa. Read off the rotational speed to 1,100 rpm at the point of intersection.

Draw an imaginary line in the power requirement diagram at an air pressure of 5.2 m³/s and a rotational speed of 1,100 rpm. From the point of intersection, go to the left and read the net power 7.2 kW.

In the upper diagram, read off the total acoustic power level to the connected outlet duct $L_{W,tot}$ to 100 dB; see further example on next page.



Report of audio data

The sound is shown as acoustic power levels in dB per octave band. This provides an image of how much acoustic power emanates from the fan and which frequency distribution the sound has. With a knowledge of the attenuation in parts of the apparatus in the audio path, and the ventilated room's sound-absorbing capacity, the sound pressure level and sound level dB(A) can subsequently be calculated for different places in the room.

The total acoustic power level, $L_{w,tot}$ in dB to a connected outlet duct is marked with blue curves and blue numbers in the fan section's pressure-flow curve.

Using a correction factor K_{ok} that is dependent on audio path, the rotational speed and the sound's frequency, the acoustic power level per octave, $L_{w,ok}$ are calculated for different audio paths.

K_{ok} is shown in table form under the fan curves.

$L_{w,ok} = L_{w,tot} + K_{ok}$

$L_{W,ok}$ = Acoustic power level in octave band, dB (relative to 10–12W) for the audio path.

$L_{w,tot}$ = total acoustic power level to connected outlet duct, dB (relative to 10–12 W), in octave bands 125–8,000 Hz.

K_{ok} = correction factor, depending on audio path, rotational speed and octave band.

Example:

Radial fan FKLB-3-090 with free-standing suction inlet.

Given:

Air flow 5.2 m^3/s.

Total pressure 1,210 Pa.

Determine the acoustic power level in the octave bands for the following audio paths:

A. To connected outlet duct or fan.

B. To inlet duct.

C. To the fan's surroundings.

Solution:

From the fan's pressure-flow diagram, the following are read:
Rotational speed N = 1,100 rpm

Total acoustic power level to connected outlet duct,
$L_{w,tot}$ = 100 dB.

Table A

Audio path: To outlet duct,

Given: Rotational speed range 200–1,300 rpm.

From the table for K_{ok}, corrections as per below are obtained:

Octave band No.	1	2	3	4	5	6	7	8
Mean frequency (Hz)	63	125	250	500	1,000	2,000	4,000	8,000
$L_{W,tot}$ (dB)	100	100	100	100	100	100	100	100
K_{ok} (dB)	–6	–3	–4	–10	–18	–29	–36	–45
$L_{W,ok}$ (dB)	94	97	96	90	82	71	64	55

Table B

Audio path: To inlet duct.

Given: Rotational speed range 200–1,300 rpm.
To the left of the line for highest efficiency, see previous page.

From the table for K_{ok}, corrections as per below are obtained:

Octave band No.	1	2	3	4	5	6	7	8
Mean frequency (Hz)	63	125	250	500	1,000	2,000	4,000	8,000
$L_{W,tot}$ (dB)	100	100	100	100	100	100	100	100
K_{ok} (dB)	–2	–5	–10	–16	–22	–28	–35	–43
$L_{W,ok}$ (dB)	98	95	90	84	78	72	65	57

Table C

Audio path: To the surroundings:

Given: Rotational speed range 200–1,300 rpm.
Free-standing suction fan.

From the table for K_{ok}, corrections as per below are obtained:

Octave band No.	1	2	3	4	5	6	7	8
Mean frequency (Hz)	63	125	250	500	1,000	2,000	4,000	8,000
$L_{W,tot}$ (dB)	100	100	100	100	100	100	100	100
K_{ok}(dB)	–22	–10	–10	–13	–17	–22	–29	–36
$L_{W,ok}$ (dB)	78	90	90	87	83	78	71	64