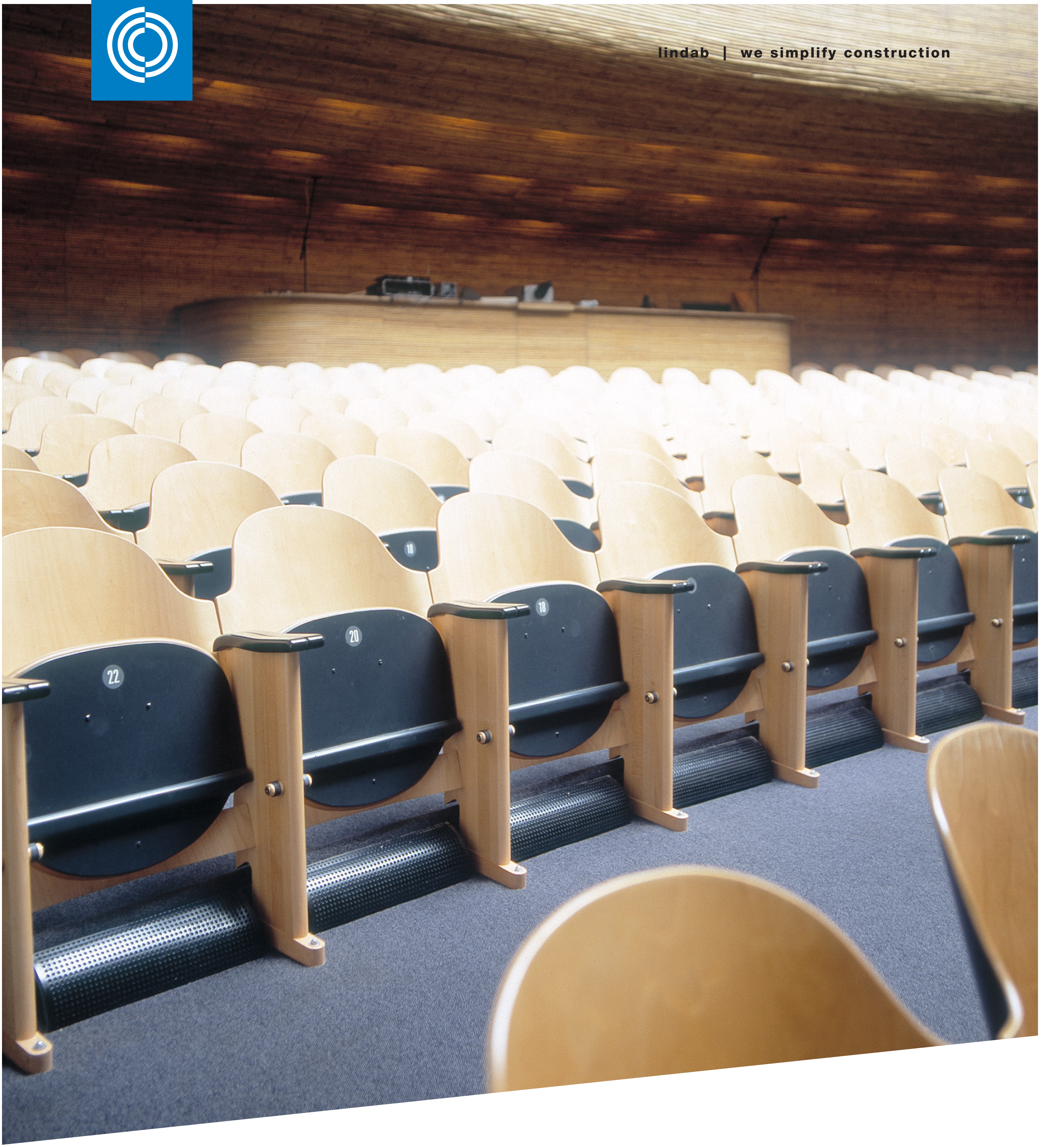

Lindab **Air Theory**

Airborne solutions

Lindab®

Theory

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Air distribution systems Theory

Mixed ventilation

In mixed ventilation the air is supplied with a relatively high velocity outside the occupied zone, usually from the ceiling or the wall. The high velocity of the supplied air means, that a considerable amount of room-air is circulated as well. The velocity of the supplied air should be kept at a level which ensures that the mixing is effective, but at the same time ensures that the air velocity has fallen to the required level by the time it reaches the occupied zone. This makes demands on the efficiency of the units used as regards to velocity and mixing capacity.

An increase in the supplied air velocity will cause an increase in the sound level. Requirements for a low sound level consequently means a limit on the diffusers efficiency. The temperature and the contamination concentration is roughly the same throughout the room, for both isothermal and cold air.
Mixed ventilation is mostly unaffected by outside influences and can be used for both heating and cooling needs.

Supply of heated air

Since heated air is lighter than the room-air, it takes a considerable energy to force the air into the occupied zone. This means that the requirements for the downward supply air velocity rises with the increase of ceiling height and rising temperatures. When the ceiling height is high, it is usually necessary to blow the air vertically down wards.

Supply of cold air

The heavier cold air, supplied from the ceiling, may lead to excessive air velocity in the occupied zone if the thermal loads are large. The air jets from diffusers (normally horizontal) and the convection streams from the heating sources (people, lighting, machines) result in a velocity in the occupied zone, which in addition to the supplied air velocity from the diffuser, depends on the removed effect per square meter (W/m^2), the distribution on the individual diffusers and the diffusers jet pattern.

The supply of both heated and cold air in the same diffuser, from the ceiling cannot normally fulfil requirements for temperature gradient, ventilation efficiency and velocity in the occupied zone at the same time.

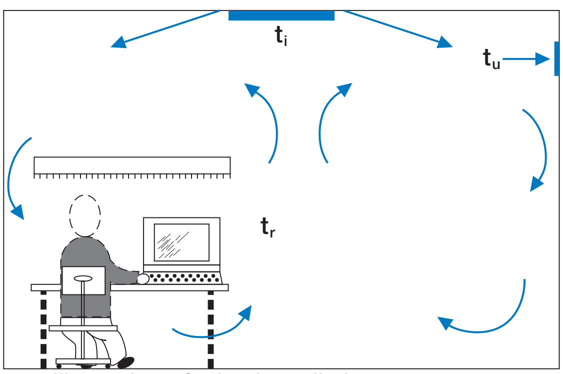


Fig. 1, Illustration of mixed ventilation.

The solution to this problem may be motorized diffusers, which can change the jet patterns. Another option is to dimension the diffusers to suit the cooling situation in question, and then add vertical nozzles for supply of heated air.

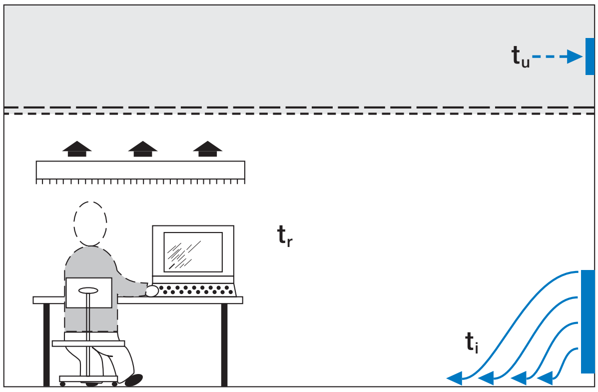


Fig. 2, Illustration of displacement ventilation.

Displacement ventilation

When using displacement ventilation, it is the thermal forces from the heating sources in the room, that control the air distribution. The air is supplied directly into the occupied zone at floor level - at low velocity and a cooling temperature. The air spreads across the floor, and displaces the hot, contaminated air, which is forced to the ceiling by the convection flow from the heating sources. Exhaust units should be placed in the ceiling, where a hot “contaminated” layer is formed.

The ventilation efficiency of displacement ventilation is larger than the mixed ventilation owing to this stratification of the air. The difference is increased with the ceiling height.
The increased temperature efficiency means, that cooling power can be saved, or that the cooling effect of the outside air can be used better, since the exhaust air is warmer and consequently will transport more effect from the room.
In normal circumstances displacement ventilation is not suitable for heating purposes.

The near-zone of the units depends primarily on the amount of supplied air, the cooling temperature and the placement of the unit. Within the recommended air flow area, the units size has no practical influence on the near-zone. The near-zone geometry can however be altered to suit the individual needs just by adjusting the nozzles.

Air distribution systems

Theory

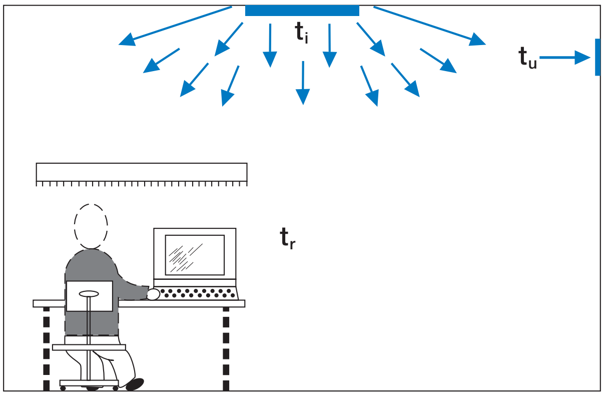


Fig. 3, Illustration of low impulse supply air.

Low-impulse supply air

By using low-impulse supply air, cold air from the ceiling is supplied at a low velocity. The clean air displaces the contaminated air.

The best result is obtained by distributing the supplied air flow in small portions spread out over the entire ceiling.

The system cannot be used for heating.

Choice of air distribution system

The different systems have their advantages and disadvantages. These should be considered carefully before choosing a system solution.
All the system solutions have one thing in common: the more units used, and the better the distribution of units in the room the better the thermal and atmospheric comfort achieved.
The advantages and disadvantages are outlined below.

Mixed ventilation

- + Can be used for cooling and heating.
- + Large induction allows supply air with a larger cool ing temperature.
- + Largely the same temperature and air-quality through out the room, ie. a small temperature gradi ent and a small concentration gradient.
- + Stable flow pattern.
- + Flexibility with regards to placement of the diffusers.
- + No reduction of useful area (near-zone).
- - Risk of short-circuits/low ventilation efficiency (par ticularly for heating).
- - More power required for cooling.
- - Risk of draft when large cooling effect.

Displacement ventilation

- + High ventilation - and temperature efficiency.
- + High air quality in the occupied zone.
- + Low velocity in the occupied zone, although not in the near-zone.
- + Suitable for cooling of rooms with high ceiling height.
- - Less freedom with regard to furniture positioning, and the room space is reduced due to the diffusers near-zone.
- - Low induction.
- - Large vertical temperature gradient.
- - Heating is not possible.

Low impulse

- + No reduction of useful area.
- + Suitable for large air replacement with limited cooling temperature.
- + High local efficiency.
- - Low induction.
- - Heating is not possible.
- - Risk of short-circuit when the exhaust is in the ceil ing.

Planning of sound level

Theory

Choice of air distribution system

		Mixed								Displacement			Low-impulse
		Nozzles	Grilles	wall diffusers	Ceiling diff. w. one slot	Perf. diffusers	Cone diffusers	Swirl diffusers	Slot diffusers	Wall diffusers	Floor diffusers	Under seat supply	
Office	Heat + cold				••	••	••	•••					
	0-30 W/m²			••	•••	•••	•••	•••	•	•••			
	30-60 W/m²				••	•••	••	•••					
	>60 W/m²					••	••	••		••	•••		
Audience	Conference rooms				•	••	••	••		•••	•••		
	Cinema					•	•	•		••	••	•••	
	Auditoria					•	•	•		••	••	•••	
	Restaurants				•	••	••	••		•••	••		
	Educational estab.			••	••	•••	•••	•••		•••			
	Exhibition halls		•	••	•••	•••	•••	•••	•	•••	•••		
Stores	Shops	•	•	••	••	•••	•••	•••	•				
	Supermarkets	•	•		•••	•••	•••	•••					
	Sportshalls	•••	••				••			•			
	Swimming baths	•••	••			•	••	••					
	Industrial kitchens				•	••	••	••		••			•••
	Laboratories				•	••	••	••		••			•••
	"Clean room"				•	••	••	•••					
	Homes		••	••	•••	•••			••				
	Institutions		••	••	•••	•••				••			

• Usable •• Good ••• Best

Choice of air distribution system in industrial environments

Ventilation need	Heating need	Cooling need	Mixed ventilation	Displacement ventilation	Low-impulse
❄	❄	❄	X		
❄	❆	❄	X		
❄	❆	❆	X		
❄	❄	❆		X	X
❆	❄	❄	X		X
❆	❆	❄	X		
❆	❆	❆	X		
❆	❄	❆		X	X

❄ Little need ❆ Large need

Mix ventilation

Theory

Mixed ventilation

An air distribution unit must supply a certain amount of air, in order to provide adequate ventilation. At the same time requirements for sound pressure, air velocity and temp- erature gradient in the occupied zone must be respected. In order to fulfil these requirements, certain planning guidelines are necessary. The most important ones are specified below. When choosing a diffuser, values such as pressure loss, sound level and air throw have to be taken into account. This data is specified for each individual product separately.
The selection and performance data contained in the Lindab product data sheets are the result of measurements conducted in Lindabs laboratory using modern precision instruments. The conditions are rarely as ideal in practice as in a laboratory, since constructional choices, furnishing, placement of air distribution units etc. have a great influence on the distribution of the air in the room. Lindab offers to test the conditions in practice by conducting full-scale testing, which is very useful when large and complicated projects are being planned.

Descriptions

A	Total room absorption	[m²]
b_h	Maximum horizontal spread to final velocity 0.2 m/s	[m]
b_v	Maximum vertical spread to final velocity 0.2 m/s	[m]
F	Free cross-section (q/v_0,where v_0 is measured)	[m²]
K_{ok}	Octave correction value for sound power level	[dB]
$l_{0,2}$	Air throw to terminal velocity 0.2 m/s	[m]
$l_{0,0}$	Turning point at vertical supply air	[m]
l_b	istance from the unit to point of maximum spread	[m]
L_A	A-balanced sound pressure level	[dB(A)]
L_{WA}	A-balanced sound power level	[dB(A)]
L_{Wok}	Sound power level in octave-bands	[dB]
L_p	Sound pressure level	[dB]
L_W	Sound power level	[dB]
ΔL	Sound attenuation	[dB]
D	Room attenuation	[dB]
Δp_t	Total pressure loss	[Pa]
q	Air flow	[m³/h], [l/s]
Δt	Temperature difference between supply air temperature and room air temperature	[K]
v_0	Supply velocity	[m/s]
v_x	Jet velocity at distance ‘x’ from centre of diffuser	[m/s]
v_{term}	Thermal maximum velocity in the occupied zone	m/s]

Pressure loss

The diagram shows the total pressure loss for the diffuser (at ρ = 1.2 kg/m³), meaning the sum of static and dynamic pressure (incl. a possible plenum box) connected to a straight air duct with a length of 1 m and the same dimensions as the diffuser.

Sound level

The diagrams show the A-balanced sound power level L_{WA} for diffuser and possible plenum box connected with a straight air duct with a length of 1 m and the same dimensions as the diffuser.
Sound pressure level is a measurement for the power of the sound, ie. the pressure vibrations we perceive, while the sound power level is a parameter to characterize the source of the sound. Both are normally noted in the unit dB (decibels), which can cause some confusion.

Sound pressure (Lp)
Is a measure of the intensity of the sound, characterized by pressure vibrations, perceived by the ear or measured with a microphone on a noise meter. Sound pressure is measured in Pascal (Pa) and is usually noted as sound pressure level in decibels (dB) or dB(A).

Sound power (Lw)
The power, a sound source (eg. a machine) sends out in the shape of a sound. The sound effect is measured in Watt (W) and is usually noted as sound effect level in decibels (dB) or dB(A).

In the product data sheets, sound properties of the diffusers are specified as sound power level.

Sound power level $L_W = 10 \times \log \frac{N}{N_{re}}$ [dB]

N is the actual sound power [W], which is sent out in the shape of pressure vibrations and $N_{re}=10^{-12}$ W which is the reference sound power.

Sound pressure level: $L_P = 20 \times \log \frac{P}{P_{re}}$ [dB]

P is the actual sound pressure [N/m²] and $p_{re} = 2 \times 10^{-5}$ N/m² is the reference sound pressure.
Room attenuation D [dB] is the difference between sound power level and the sound pressure level

$L_P = L_W - D$

The A-balanced sound power level, L_{WA} is calculated to sound power level in the individual octave-bands by.

$L_{Wok} = L_{WA} + K_{ok}$

K_{ok} is a correctional value. K_{ok} is noted in tabular form for each unit.

Sound attenuation

Is noted for each unit, and refers to the reduction in sound power level between duct and room (incl. end-reflection).

Isothermal supply air

All technical data refer to isothermal conditions.

Air throw

The air throw $l_{0.2}$ is defined as the largest distance between the centre of the unit and the terminal velocity 0.2 m/s.
The values specified for air throw $l_{0.2}$ correspond to diffusers mounted in the ceiling. (*Fig. 4*).

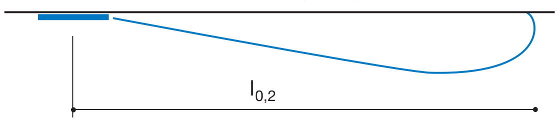


Fig. 4, Air throw $l_{0.2}$ for diffusers mounted in ceiling.

Mix ventilation

Theory

Suspended mounting, ie. diffusers mounted more than 300 mm from the ceiling (*Figure 5*), reduces the air throw by 20 %, so that $l_{0.2}$ suspended = 0.8 × $l_{0.2}$.

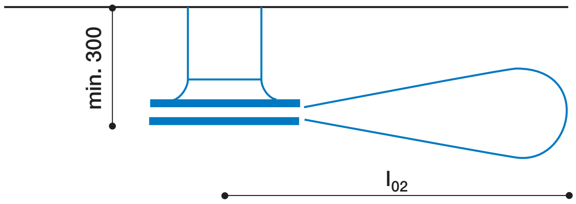


Fig.5, Suspended diffuser.

For grilles $l_{0.2}$ applies for mounting more than 800 mm from the ceiling. (*Figure 6*).

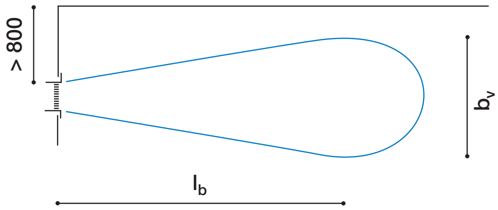


Fig. 6, Air throw for grilles mounted more than 800 mm from ceiling.

Should a grille be mounted less than 300 mm from the ceiling (*Figure 7*) the air throw $l_{0.2}$ is extended by 40%, so that $l_{0.2}$ grilles by ceiling = 1.4 x $l_{0.2}$.

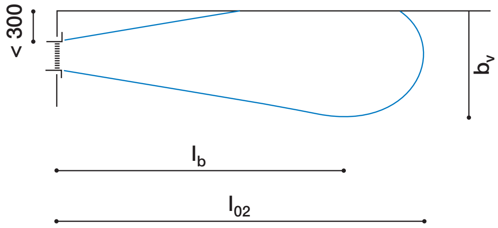


Fig.7, Air throw for grille mounted less than 300 mm from ceiling.

Spread

The maximum vertical spread b_v specifies the largest vertical distance between the ceiling and the terminal velocity 0.2 m/s (*Figure 8*).

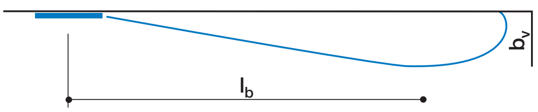


Fig.8, Vertical spread.

The horizontal spread is noted as b_h and specifies the maximum horizontal spread of the air jet for the terminal velocity 0.2 m/s *(Figure 9)*. The distance between the unit and the point of largest jet width is noted as l_b. b_v, b_h and l_b are specified for each unit as a function of the air throw $l_{0.2}$.

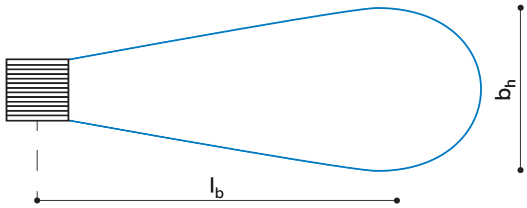


Fig. 9, Horizontal spread.

Coanda effect

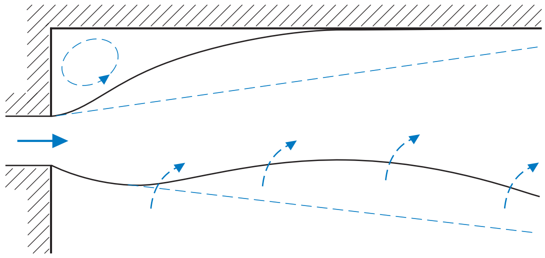


Fig. 10, Air flow with Coanda effect.

When the air is supplied parallel to a surface (eg. a ceiling) negative pressure occurs between the air jet and the ceiling, causing the jet to “stick” to the ceiling (this is known as the Coanda effect). (*Figure 10*). This effect is of great importance, particularly when supplying cooling air. To achieve the greatest possible Coanda effect, the air should be supplied in small quantities to each unit, with the widest possible spread on the ceiling and the greatest possible velocity.
This means that the best method is always to supply the air from the diffuser in a full 360°-pattern without side covers. In particular, linear diffusers (MTL) are divided into active and inactive sections to avoid drop.

Velocity in the jet

The air velocity of the core jet can be calculated within a limited area, using the following formula:

$$v_x = \frac{l_{0,2} \times 0,2}{x} \iff x = \frac{l_{0,2} \times 0,2}{v_x}$$

Where x is the distance in metres between the unit and the point in the core jet where the air velocity is v_x m/s.

<u>Example</u>
A diffuser has an air throw of $l_{0.2}$ = 3 m. The distance to the point where the jet velocity is 0.3 m/s is calculated as follows:

$$X = \frac{3\text{ m} \times 0,2\text{ m/s}}{0,3\text{ m/s}} = 2\text{ m}$$

Mix ventilation

Theory

Thermal supply air

The product data sheets values for air throws are valid in the case of isothermal supply air.
When using cold or heated supply air the thermal forces work by forcing the jet downwards (cooling) or giving the jet a lift (heating). A description of the jet flow would require a determination of the ratio of temperature difference and supplied air velocity (in the jet-theory expressed by Archimedes number). If a more detailed calculation of supply air velocities is needed - where this is factored in - in addition to a visual of the jet flow from the diffusers, we refer you to the software programme DIMcomfort.

The general rule below for horizontal and vertical supply air with cold- or heated air, can however be used for correction of the air throw in a more simple calculation.

Horizontal supply air at the ceiling

1. When air is supplied horizontally with cold air, the air throws are reduced by 1.5% per degree (*Figure 11*), while the vertical spread b_v is increased.

2. When air is supplied horizontally with heated air, the air throws are increased with 2% per degree (*Figure 11*).

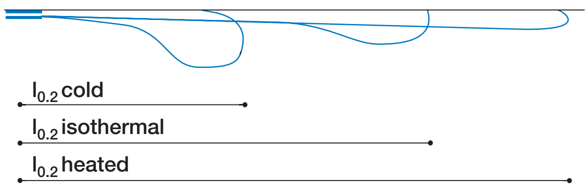


Fig. 11, Air throw $l_{0.2}$ for diffusers mounted in ceiling.

Vertical supply air at the ceiling

The throw lengths for vertical supply air is are valid for isothermal conditions.

1. When air is supplied at a cooling temperature the throw length is increased. The throw length is doubled at Δt = -10˚C.

2. When air is supplied with heated air, the throw length is reduced. The throw length is halved at Δt = 10˚C.

For products, which can be set for vertical supply air, there are also other separate turning point diagrams for heated air (Δt = +5K, +10K and possibly +15K) for turning point $l_{0.0}$ in addition to the other product data.

Dimensioning mixed ventilation

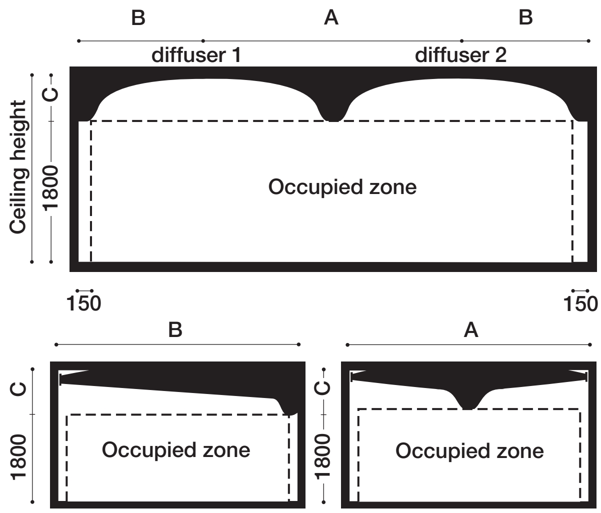


Fig. 12, Planning af mixed ventilation.

In order to avoid velocities more than 0.2 m/s in the occupied zone, the diffusers must be dimensioned so that the air throw $l_{0.2}$ has the right ratio to the distance A, B and C (*Figure 12*). If there are two opposing diffusers the following formula must be observed.

$$0.75 \times \left(\frac{A}{2} + C\right) \le l_{0.2} \le \left(\frac{A}{2}\right) + C$$

In the case of a diffuser blowing towards a wall the following formula must be observed:

$$0.75 \times (B + C) \le l_{0.2} \le B + C$$

If two or more diffusers with a parallel delivery of supply air (1-way or 2-way) are placed with a spacing A between them, which is less than b_h, the air throw increases in accordance with the following formula :

$$l_{0.2}\,(\text{corrected}) = K \times l_{0.2}$$

where K is the correctional factor to be read from *Figure 13*.

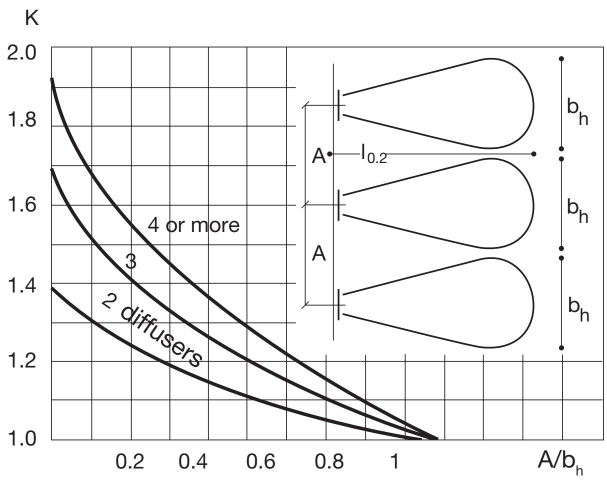


Fig. 13, Planning mixed ventilation.

Mix ventilation

Theory

For nozzles and suspended diffusers with 1-way supply of air, the lift or drop of the jet as a consequence of heating or cooling supplied air can be read in *Figure 14*.

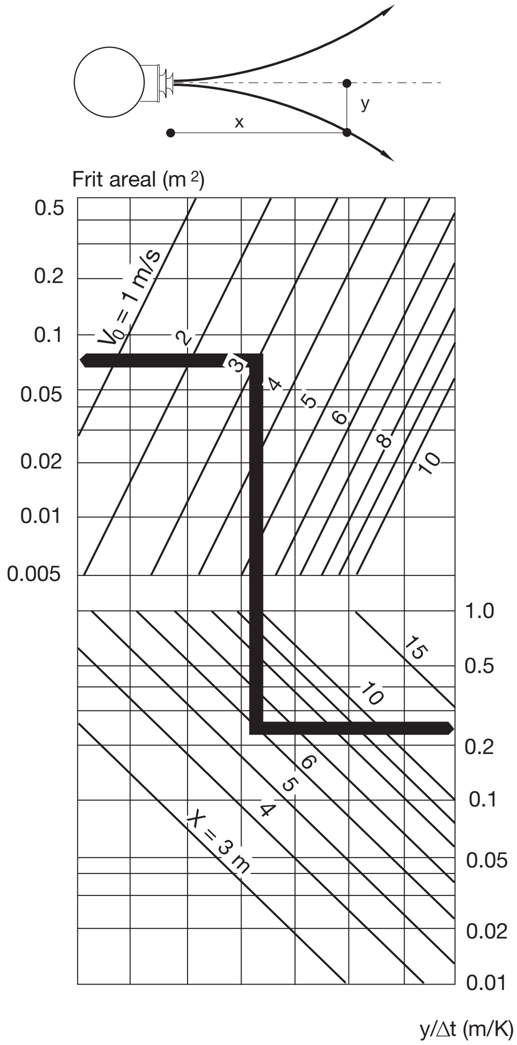


Fig. 14, Planning mixed ventilation.

<u>Example</u>

A nozzle has a free area of 0.075 m².

With an air volume of 756 m³/h a supplied air velocity of v_0 = 3 m/s ($v_0 = q / A_0$) is achieved.

Figure 14 has a thick horizontal line between A_0 = 0.075 m² and v_0 = 3 m/s.

By following the thick line straight down to x = 6 m and then horizontal to the right, the ratio between y (lift/drop) and Δt (temperature difference between supplied air and room air) can be read to be 0.24.

With a temperature difference of 10 K a lift/drop at y = 0.24 m/K × 10 K = 2.4 m at a distance of x = 6 m from the nozzle is achieved.

To avoid the jet being deflected by possible obstacles, the minimum distances in *Figure 15* must be observed.

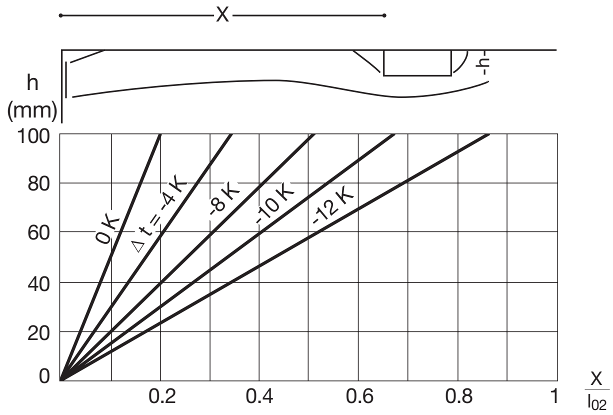


Fig. 15, Air throw $l_{0.2}$ for diffusers.

Mix ventilation

Theory

Heat loads in the room create upward convection flows, and in the same manner downward cold convection flows are created from the supplied air.

The calculated maximum velocity v_{term} in the occupied zone, which occurs due to thermal flows is shown in *Figure 16*. These flows depend on the heat load in the room (W/m²) in addition to the distribution of the supplied air (number of diffusers and jet pattern), but not of the impulse of the supplied air. Furthermore the velocity depends on the ceiling height. The determination of the maximum velocity in the occupied zone is made by the help of an empirical model from the heat load (W/m²), number of diffusers (W/diffuser) and air pattern (1-, 2-, 3-, 4-way) at a ceiling height of 2.5 m. If there is any doubt regarding a project, or special conditions need investigating, Lindab offers to test the conditions by conducting a full-scale test, which will often hold great value in the case of bigger and more complex constructional tasks.

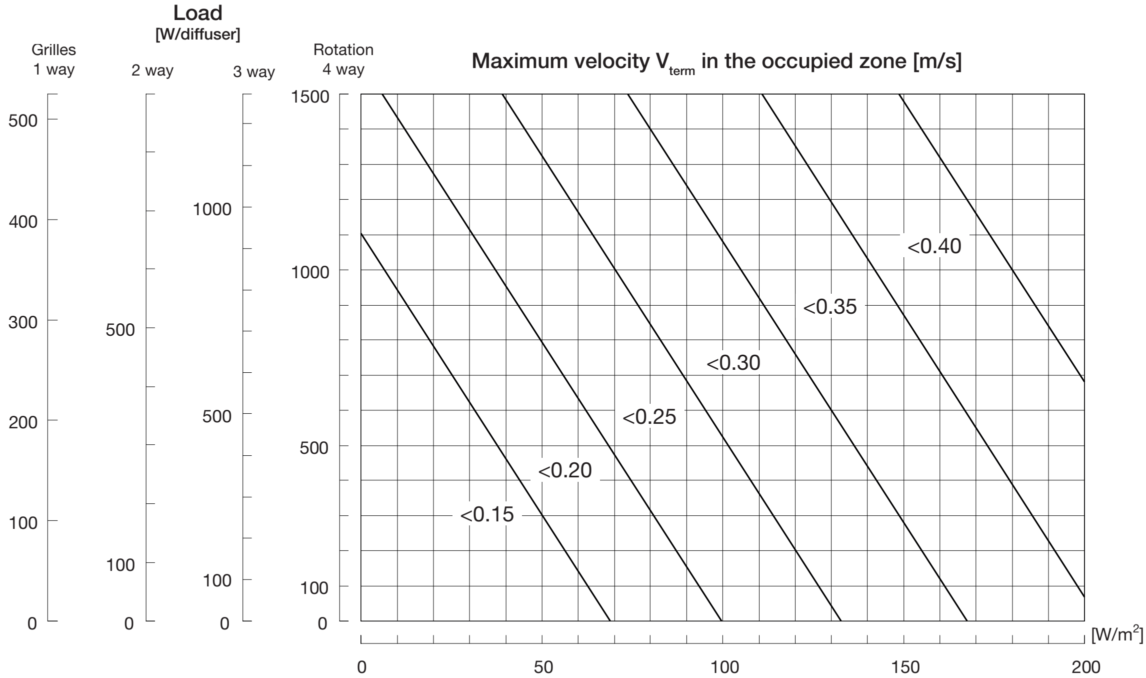


Fig. 16 a, Thermal maximum velocity in the occupied zone. The diagram is advisory and valid for ceiling heights of 2.5 m.

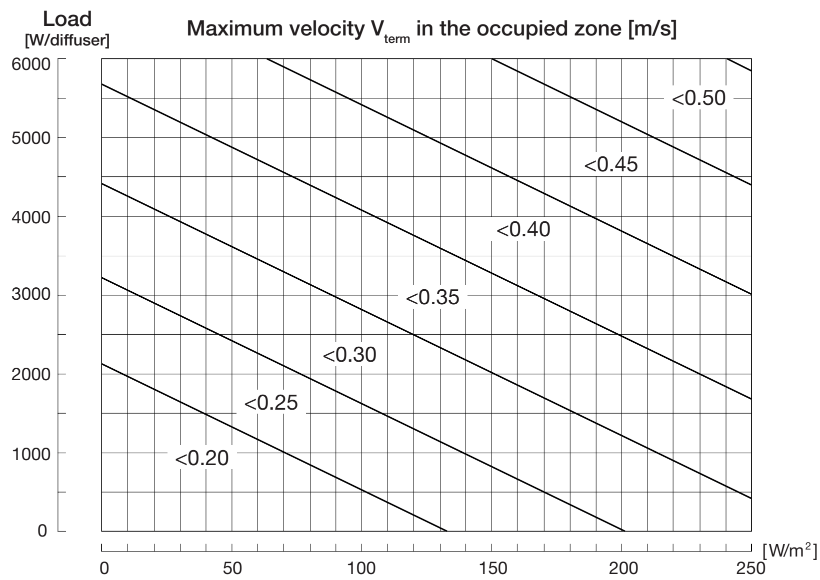


Fig. 16 b, Thermal maximum velocity in the occupied zone. The diagram is advisory and valid for ceiling heights > 4 m.

Mix ventilation

Theory

Calculation example

Room: L × B × H = 10 m × 6 m × 4 m

Thermal load:

10 pers., sitting activity (10 × 130 W)	= 1300 W (22 W/m²)
10 table lamps of 60 W (10 × 60 W)	= 600 W (10 W/m²)
10 machines of 100 W (10 × 100 W)	= 1000 W (17 W/m²)
Total	= 2900 W (48 W/m²)

In order to achieve a satisfactory air quality in the room, the typical calculations determine that the ventilation should have a supply air of 4-10 l/s per person in addition to 0.4 l/s per m² floor area. If 10 l/s is used, the following necessary air volume can be calculated.

q_{min} =

10 persons × 10 l/s per person + 60 m² × 0.4 l/s per. m²

= 124 l/s

If the ventilation at the same time has to remove the collective heat load in the room, it is necessary to have a temperature difference Δt between the supply air and the room/ exhaust air.

Δt can be determined to be :

$$\Delta t = \frac{2900\ \text{W}}{\frac{124\ \text{l/s}}{1000\ \text{l/m}^3} \times 1{,}2\ \text{kg/m}^3 \times 1007\ \text{J/kg/K}} = 19{,}4\ \text{K}$$

Since Δt of almost 20 K is very likely to cause thermal discomfort, eg. due to drop from a ceiling diffuser, it is recommended to increase the air volume and use less Δt between supply and room temperature.

If Δt = 6 K is chosen the air volume can be determined to be :

$$q = \frac{2900\ \text{W}}{6\ \text{K} \times 1{,}2\ \text{kg/m}^3 \times 1007\ \text{J/kg/K}} \times 1000\ \text{l/m}^3 = 400\ \text{l/s}$$

Displacement ventilation Theory

Displacement ventilation

A displacement unit should add a certain amount of air to properly ventilate the room, and at the same time meeting the requirements for sound level, air velocity and temperature gradient in the occupied zone. In order to meet these requirements, planning guidelines are needed, and the most important ones are stated hereafter. When choosing a unit, the demands on pressure loss, sound level and air throw should be made clear. These data can be found for each individual product. The selection- and performance data shown in Lindabs product data sheets is the result of measurements carried out in Lindabs laboratory and are all conducted with modern and accurate measuring devices. In practice the conditions are rarely as ideal as in a laboratory, since the constructional environments, furnishing, placement of the air distribution units etc. has a great influence on the jet pattern spread in the room. Lindab attempts to test the conditions in practice by carrying out full-scale testing, which is often very valuable in the case of bigger and complicated tasks.

Descriptions

$a_{0.2}$	Width of near-zone	[m]
$b_{0.2}$	Length of near-zone	[m]
ε_t	Temperature efficiency	[-]
K_{ok}	Octave Correction value for sound power level	B]
L_A	A-balanced sound pressure level	[dB(A)]
L_{WA}	A-balanced sound power level	[dB(A)]
L_{Wok}	Sound power level in octave bands	[dB]
L_p	Sound pressure level	[dB]
L_w	Sound power level	[dB]
ΔL	Sound attenuation	[dB]
D	Room attenuation	[dB]
Δp_t	Total pressure	[Pa]
q	Air flow	[m³/h], [l/s]
t_i	Supply air temperature	[°C]
t_r	Room temperature (1.1 m over the floor)	[°C]
t_u	Exhaust air temperature	[°C]
Δt	Temperature difference between room air and supply air	[K]
v_x	Velocity at distance x from the centre of the unit	[m/s]

Vertical temperature distribution

Due to the stratified flow, displacement ventilation causes a big difference in temperature throughout the room. In comfort ventilation, where the heating sources are placed in the bottom part of the room, the temperature gradient, meaning the temperature rise per m (K/m) will be bigger in the lower part of the room, and smaller in the upper part.

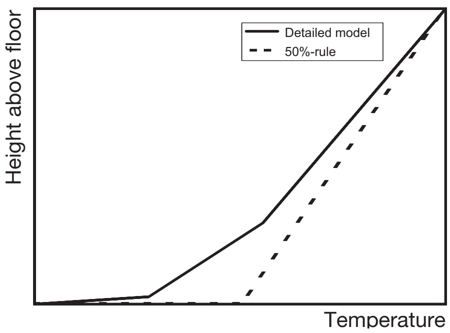


Fig. 17, Comparison of models for description of the vertical temperature distribution.

The simplest models for description of the vertical temperature distribution are the so-called "%-rules".
The most used one is the 50%-rule, in which it is assumed, that half of the temperature rise from supply air to exhaust air occurs at the floor, and the other half occurs up throughout the room (see *Figure 17*). The model is a good one, as a first evaluation of the most typical rooms and units, but because of the simplicity it does not precise determine the temperature gradient in the occupied zone.
Lindab recommends the use of a more detailed model instead. One that describes the variation of the temperature gradient up through the room. A close assumption is that the temperature gradient in the occupied zone is half of the temperature difference between the room air and the supply air. The model is based on a number of full scale tests, and factors in the temperature efficiency and the fact that the temperature gradient is larger in the lower part of the room than in the upper part.

Temperature efficiency

The efficiency in displacement ventilation is due to the stratification. The difference is increased at larger ceiling heights. The effect taken from the room is proportional to the temperature difference between supply air and exhaust air (t_u-t_i).
Since the exhaust temperature (t_u) is higher than the room temperature (t_r) in displacement ventilation, the same effect can be taken from the room at a higher supply air temperature (t_i) than with mixed ventilation, where $t_u \leq t_r$.
This means that cooling effect can be spared, or that it is possible to use the cooling effect of the outer air more efficiently.
Displacement ventilation is furthermore partly self-regulating at varying thermal loads, because a rising load first and foremost will give a higher temperature gradient and consequently a higher temperature at the ceiling.
The temperature efficiency is given at:

$$\varepsilon_t = \frac{t_u - t_i}{t_r - t_i} \times 100\%$$

With displacement ventilation it is the case that $\varepsilon_t > 100\%$ ($t_u \geq t_r$), while $\varepsilon_t \leq 100\%$ at mixed ventilation ($t_u \leq t_r$).
By ideal mixing $\varepsilon_t = 100\%$ ($t_u = t_r$).

Pressure loss

The diagrams show the total pressure loss for the unit (at ρ = 1.2 kg/m³), meaning the sum of static and dynamic pressure, connected to a straight air duct with a length of 1 m and the same dimension as the diffuser.

Sound level

The diagrams show the A-balanced sound power level L_{WA} for a diffuser connected with a straight air duct with a length of 1 m and the same dimensions as the diffuser. Sound pressure level is a measurement of the result of the sound, ie. the pressure vibrations we perceive, while the sound power level is a parameter to characterize the source of the sound. Both are normally noted in the unit dB (decibels), which can cause some confusion.

Displacement ventilation Theory

Sound pressure (Lp)
Is a measure of the intensity of the sound, characterized by pressure vibrations, perceived by the ear or measured with a microphone on a noise meter. Sound pressure is measured in Pascal (Pa) and is usually noted as sound pressure level in decibels (dB) or dB(A).

Sound power (Lw)
The power, a sound source (eg. a machine) sends out in the form of a sound. The sound effect is measured in Watt (W) and is usually noted as sound effect level in decibels (dB) or dB(A).

In Lindabs product data sheets sound properties of the units are named sound power level.

Sound power level: $L_W = 10 \times \log \frac{N}{N_{re}}$ [dB]

where N is the actual sound power [W], which is sent out in the shape of pressure vibrations and $N_{re} = 10^{-12}$ W is the reference sound power.

Sound pressure level: $L_P = 20 \times \log \frac{P}{P_{re}}$ [dB]

where p is the actual sound pressure [N/m²] and $p_{re} = 2 \times 10^{-5}$ N/m² which is the reference sound pressure.

Room attenuation D [dB] is the difference between sound power level and sound pressure level. $L_{wok} = L_W - D$

The A-balanced sound power level, L_{WA} is calculated to sound power level in the individual octave bands by :

$L_P = L_{WA} + K_{ok}$,

K_{ok} being a correctional value. K_{ok} is specified in tabular form for each individual unit.

Sound attenuation
Specified for each individual diffuser, the reduction of sound power level from air duct to room (including end reflection).

Near-zone
The area around the unit, where the air velocity is above 0,2 m/s, is referred to as the near-zone.
The size of the near-zone is specified for each unit at a cooling temperature of $\Delta t = t_r - t_i = 3K$.
The near-zone length (a_0) and – width (b_0) is valid for evenly distributed thermal loads.

Dimensioning displacement ventilation
To plan a ventilation system by displacement principle, which "works" on the basis of thermal powers, and where the supply air is added directly to the occupied zone, makes special demands on dimensioning and placement of the air distribution units. They should, as such, never be placed directly by a powerful heating source, like a radiator. Powerful sunlight can also disturb the system, and in some cases make it function as a mixed ventilation system. Large, cold walls - or window surfaces in the room can also cause a back-flow of contaminated air to the occupied zone.

The system is not suitable for heating purposes, and consequently requires heating and ventilation to be separate. Exhaust should always take place as high up in the room as possible.
If in any doubt about a project, or if there are any points to be analysed, Lindab offers to test the conditions in practice by conducting full-scale tests, which is often of great value, at bigger and complicated tasks.

Convection flow
The supplied air flow should at least be the same as the total convection flow in the room (*Figure 18*). If the supplied air flow is less than this the convection flow will draw contaminated air from above down into the occupied zone (*Figure 19*).
The following factors affect the convection flow:

- The shape and surface of the heat source
- The surface temperature of the heat source
- Convective proportion of the heating output emitted
- Mean temperature of the room
- The level of the contaminated zone in relation to the level of the heat sources in the room

The convection flow from people, lighting, and machinery can be determined from the output and the placement of the heat sources in the room (*see Table 1 and Table 2*).

Table 1, Convection flows for people based on experiences

Activity	met	Heat outputW	Airflow l/s 1.2 above floor	Airflow l/s 1.8 above floor
Setting, relaxing	1.0	100	8-10	-
Sitting activity	1.2	130	10-12	-
Light act. standing	1.6	170	-	25-30
Medium act. standing	2.0	200	-	30-35
High act. standing	3.0	300	-	35-40

Met: metabolism, 1 met = 58 W/m² body surface.

Table 2, Convection flows for various heat sources.

Heating source	Airflow l/s pr.W 1.2 above floor	Airflow l/s pr.W 1.8 above floor
Table lamps	0.10	0.20
Ceiling lights	-	-
machines	0.10	0.20
Sunlight	0.11	0.22

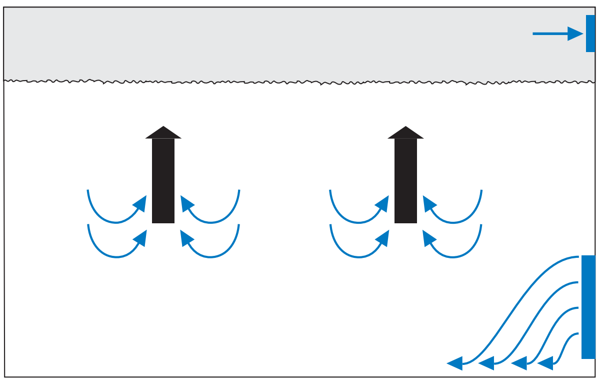


Fig. 18, Displacement ventilation with sufficient air flow.

Displacement ventilation

Theory

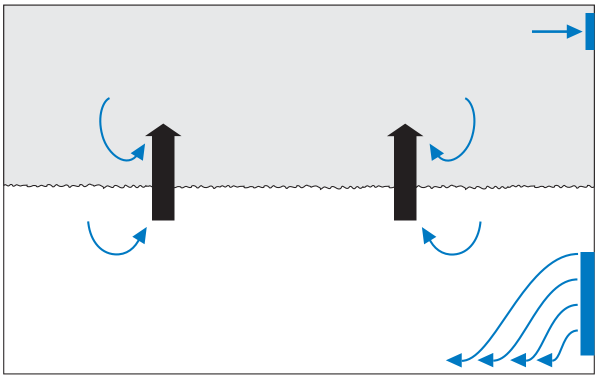


Fig. 19, Displacement ventilation with insufficient air flow.

Temperature gradient

The demands made on thermal comfort in the occupied zone places a limit on the size of the temperature gradient. Table 3 show the maximum gradient recommended by Lindab Comfort at various levels of activity.
Further more the corresponding maximum cooling temperature (t_r-t_i) is mentioned when using Lindabs COMDIF-units. The temperature gradient in the occupied zone (K/m) can with a small margin be set at half of the cooling temperature t_r-t_i (K).

Table 3, Recommended temperature gradients and cooling temperatures.

Activity	Max. temperature gradient (K/m)	Max. undertemperature t_r-t_i (K)
Sitting, relaxing	1.5	3.0
Sitting activity	2.0	4.0
Light act., standing	2.5	5.0
Medium activity	3.0	6.0
High activity	3.5	7.0

Near-zone

The size of the near-zone is specified for each unit in the product data sheets. If several units are placed close to one another, the near-zone will increase (*Figure 20*).

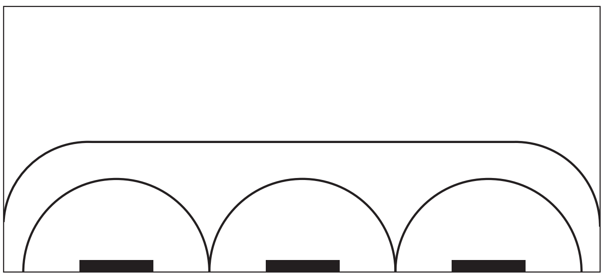


Fig. 20, Diffusers placed too close, limiting the individual diffusers induction.

A big air flow from one unit can result in a too big near-zone (*Figure 21*). If the air is instead distributed on two units, smaller near-zones are the result. (*Figure 22*).
To achieve the smallest possible near-zones, and thus the best possible use of the room, the air flow should be distributed evenly in the room with as many units as possible.

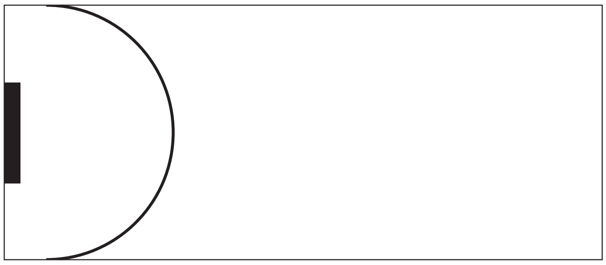


Fig. 21, Too great air flow on one diffuser results in a too big near zone.

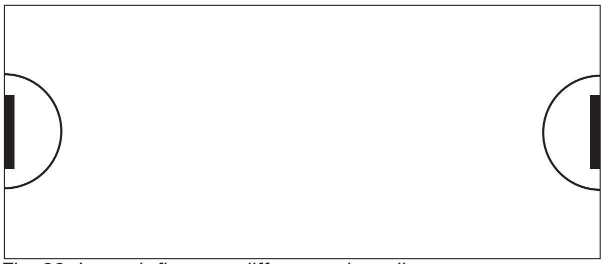


Fig. 22, Less air flow per diffuser and smaller near zones.

More units

When several units are placed too close to one another by the same wall, the near-zone is increased as shown in *Figure 20*, since jet streams can form between the units. In a certain distance from the units however, a continuous jet flow will be formed with a near constant velocity. This end-velocity is dependent on the total airflow per m wall and the cooling temperature. In *Figure 23* this end-velocity can be read. It will often be an advantage to distribute the air on units placed on adjacent walls at a 90 degree angle. In this case, the units should also be placed evenly along the walls, since of course jets also form between too closely placed units around the corner of a wall.

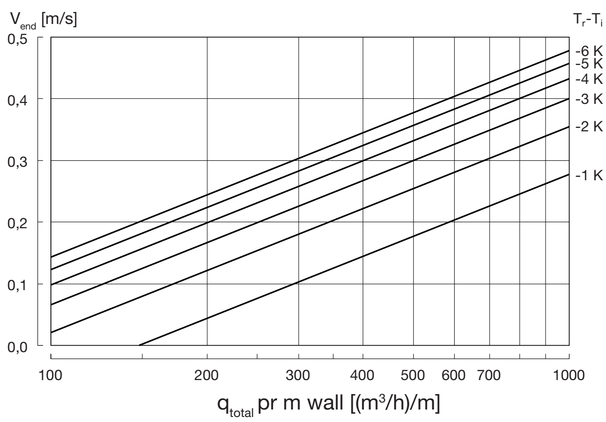


Fig. 23, End velocity at continuous jet flow.

Displacement ventilation Theory

Output

In order to calculate the output which can be removed from the room by a displacement system, the temperature difference t_u-t_i, has to be known (depends on the thermal load, ceiling height and cooling temperature (t_r-t_i)).

By calculating the temperature efficiency and the necessary difference in temperature t_u-t_i the heating sources close to the ceiling (eg. lighting) are accounted for by 50% of the output.

From *Figure 24* the temperature efficiency ε_t can be read at different combinations of ceiling height and heat loads.

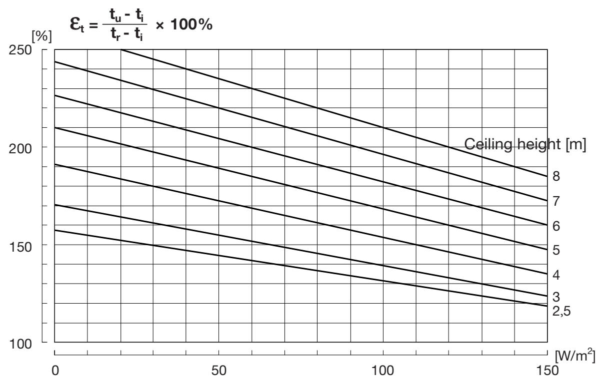


Fig. 24, Temperature efficiency is dependent on ceiling height and heat load.

Calculation example

Room: L × B × H = 10 m × 6 m × 4 m

Thermal load:

10 pers., sitting activity (10 × 130 W)	= 1300 W (22 W/m²)
10 table lamps of 60 W (10 × 60 W)	= 600 W (10 W/m²)
10 machines of 100 W (10 × 100 W)	= 1000 W (17 W/m²)
Total	= 2900 W (48 W/m²)

Minimum air flow (from Table 1 and Table 2):
q_{min} = 10 pers. × 11 l/s/pers. + 10 table lamps × 60 W/table lamps × 0.1 l/s/W + 10 machines × 100 W/machines × 0.1 l/s/W = 270 l/s

Required temperature difference (t_u-t_i):

$$t_u - t_i = \frac{2900\ \text{W}}{\frac{270\ \text{l/s}}{1000\ \text{l/m}^3} \times 1{,}2\ \text{kg/m}^3 \times 1007\ \text{J/kg/K}} = 8{,}9\ \text{K}$$

From *Figure 24* the temperature efficiency can be read at ε_t = 178% by a ceiling height of 4 m and a heat load of 48 W/m².
Consequently the temperature difference t_r-t_i can be determined by using the formula:

$$\varepsilon_t = \frac{t_u - t_i}{t_r - t_i} \quad <=> \quad t_r - t_i = \frac{t_u - t_i}{\varepsilon_t} = \frac{8{,}9\ \text{K}}{1{,}78} = 5\ \text{K}$$

which gives a temperature gradient in the occupied zone of 2.5 K/m (since the temperature gradient in the occupied zone can be set to the half of the cooling temperature t_r-t_i).

Lindab recommends a temperature gradient of <2 K/m and therefore the air flow should be increased.

A temperature gradient of 2 K/m gives t_r-t_i = 4 K and with unchanged temperature efficiency of 178% the acceptable temperature difference is t_u-t_i = 7,1 K.

To remove the thermal load of 2900 W the air flow must be changed to:

$$q = \frac{2900\ \text{W}}{7{,}1\ \text{K} \times 1{,}2\ \text{kg/m}^3 \times 1007\ \text{J/kg/K}} \times 1000\ \text{l/m}^3 = 337\ \text{l/s}$$

Planning of sound level Theory

Planning sound level

The diagrams in the product data sheets specify the A-weighted sound effect level L_{WA} for diffusers connected to a straight air duct with a length of 1 m and the same dimension as the diffuser.
The actual sound pressure level that we hear is determined as shown below.

Key

A	Total room absorption	[m²]
K_{ok}	Octave correction value for sound power level	[B]
L_A	A-balanced sound pressure level	[dB(A)]
L_{WA}	A-balanced sound power level	[dB(A)]
L_{Wok}	Sound power level in octave bands	[dB]
L_p	Sound pressure level	[dB]
D	Room attenuation	[dB]
L_w	Sound power level	[dB]
V	Room volume	[m²]
T_s	Reverberation time	[-]
D	Room attenuation	[dB]
Q	Direction factor	[-]
Δ	Increase in sound power level at a given number of identical units	[dB]
r	Distance to closest unit	[m]
α	Absorption factor	[-]
n	Number of units	[-]

Sound pressure level

The collective sound effect L_W from a number of similar diffusers is found through a logarithmic multiplication of the number of diffusers with the sound power level from an individual diffuser

$L_W = L_{W1} \otimes n$

where L_{w1} is the sound power level from an individual diffuser [dB] and n is the number of diffusers.
The collective sound power can, by help of *Figure 25* be calculated as $L_W = L_{W1} + \Delta$ where Δ is the increase of sound power level for a given number of identical diffusers.

n	1	2	3	4	5	6	7	8	9	10	15
Δ	0	3.0	4.8	6.0	7.0	7.8	8.5	9.0	9.0	10.0	11.8

Fig. 25, Increase of sound power level (logarithmic multiplication) by a number of identical sound sources.

With the knowledge of the sound sources and the absorption area of the room, the attenuation of the room is determined by *Figure 26, Figure 27 and Figure 28* at one or several identical sound sources in the room.
The actual sound pressure level is the difference between the sound power level and the room attenuation where L_p is the sound pressure level [dB], L_W is the sound power level [dB] and D is the room attenuation [dB].
In the case of different sound sources in the same room, the sound pressure level is found at a given point by a logarithmic addition of the sound pressure levels for the individual sound sources (*Figure 29*).

A can also be calculated from reverberation time by using the formula:

$$A = 0.16 \times \frac{V}{T_s}$$

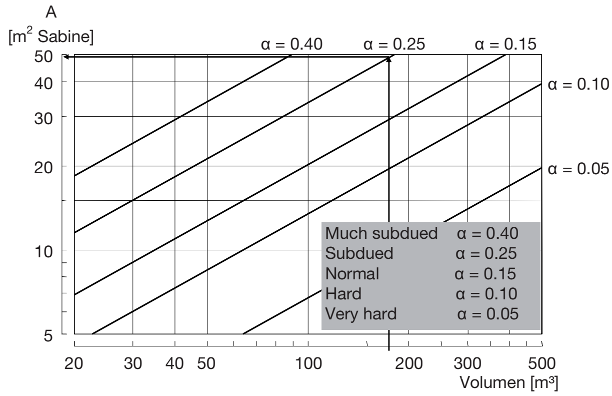


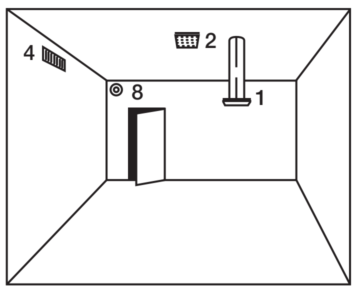
Fig. 26, Direction factor for different placements of sound sources and the relationship between the room volume and equlent sound-absorption area.

Calculation example

In a room with the dimensions L × B × H = 10 m x 7 m x 2,5 m four diffusers are mounted in the ceiling. Each diffuser gives off a sound power level of 29 dB(A). The room is attenuated, which gives an absorption area of A ~ 50 m² Sabine (*Figure 26*). The sound pressure level needs to calculated 1,5 m above the floor.
Sound power from the four diffusers: $L_W = 29 \otimes 4 = 29 + 6 = 35$ dB(A) (*Figure 25*).
For diffusers mounted in the ceiling the direction factor Q = 2 and consequently becomes (*Figure 27*).

√n / √Q = 1,4

At the height of 1.5 m over the floor the distance to the closest diffuser is r = 1 m, and therefore the room attenuation can be determined to be D = 9 dB via. *Figure 28*.
The sound pressure level in the room: L_A = 35 dB(A) - 9 dB = 26 dB(A).



n	1	2	3	4	5	6	7	8	9	10	15
Q					√n / √Q						
1	1.0	1.4	1.7	2.0	2.2	2.4	2.6	2.8	3.0	3.2	3.9
2	0.7	1.0	1.2	1.4	1.6	1.7	1.9	2.0	2.1	2.2	2.7
4	0.5	0.7	0.9	1.0	1.1	1.2	1.3	1.4	1.5	1.6	1.9
8	0.4	0.5	0.6	0.7	0.8	0.9	0.9	1.0	1.1	1.1	1.4

Fig. 27, Direction factor for different placements of sound sources and the ratio between √n / √Q as a function of number of sound sources and direction factor (picture).

Planning of sound level

Theory

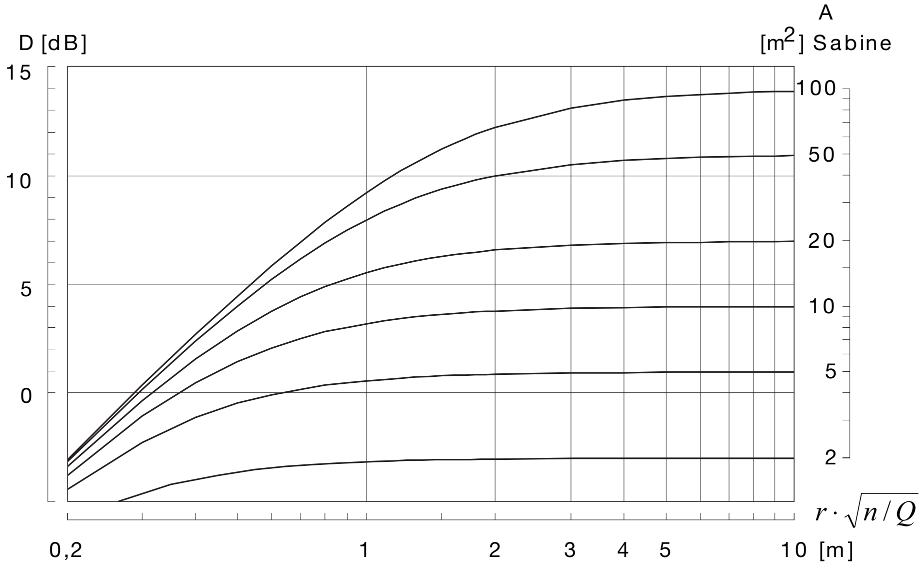


Fig. 28, Room dampening as a function of area of absorption and number of sound sources.

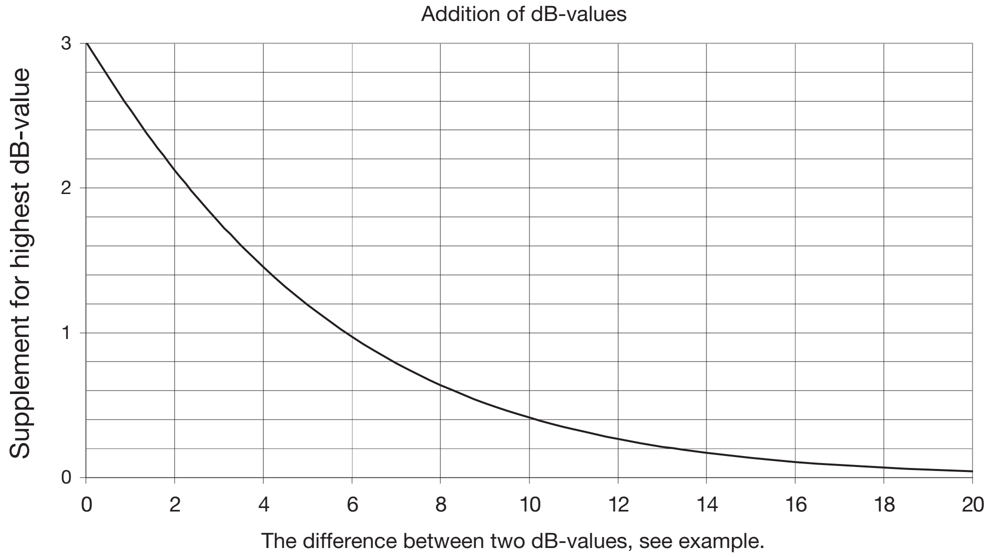


Fig. 29, Addition of sound levels (logarithmic addition of sound effect level or sound pressure level).

E.g. two sources at 41 dB and 47 dB;
difference is 47 - 41 = 6 ;
from graph: 6 on X-axis = 1 on Y-axis;
47 + 1 = 48 dB resultant level.

Most of us spend the majority of our time indoors. Indoor climate is crucial to how we feel, how productive we are and if we stay healthy.

We at Lindab have therefore made it our most important objective to contribute to an indoor climate that improves people's lives. We do this by developing energy-efficient ventilation solutions and durable building products. We also aim to contribute to a better climate for our planet by working in a way that is sustainable for both people and the environment.

Lindab | For a better climate

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