LindQST ICD Calculation methods and disclaimers

Using LindQST Indoor Climate Designer requires basic knowledge of ventilation technique. Terms as throw length, velocity in the occupied zone, sound effect and sound pressure level are used to describe the thermal comfort and acoustic indoor climate.

Content

Determining ventilation parameters

In LindQST ICD you must type in the parameters for the ventilation in the room. The necessary ventilation rate must be determined. Most often this is done on basis of cooling or heating needs or the need for outdoor air determined by the dilution equation from occupant generated CO_2.

Heat balance

$$\Phi_v = q_v \cdot \rho \cdot c_p \cdot (T_s - T_e)$$

Φ_v is the effect for cooling or heating (cooling will be shown with negative sign) [W]
q_v is the air flow [m^3/s]
ρ is the air density [kg/m^3]
c_p is the specific heat capacity of air [J/(kg·K)]
T_s is the supply temperature [°C]
T_e is the extract temperature [°C]

At normal air conditions (20°C og 101,3 kPa) the formula can be shortened to

$$\Phi_v = q_v \cdot 1{,}2 \cdot (T_s - T_e)$$

q_v is the air flow [l/s]

Mixing ventilation

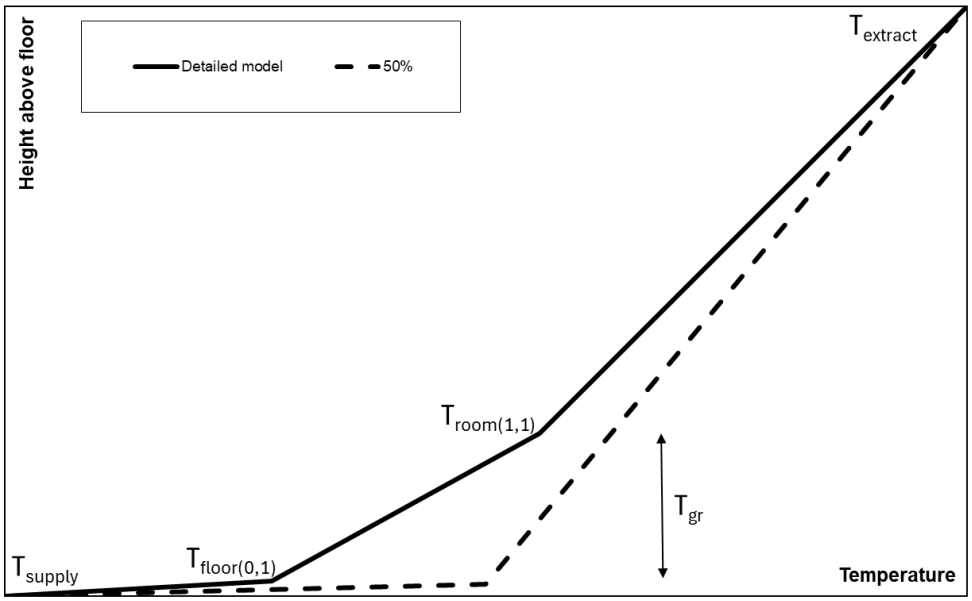
Applies to ceiling integrated diffusers, free hanging diffusers and active chilled beams.

Fully mixing in the room is assumed, which implies that the room temperature is equal to the extract temperature ($T_r = T_e$). The three parameters (q_v, T_s, T_r) always must be specified, the required Φ_v can also be specified and will then be held up against the calculated Φ_v.

If activating CO_2 calculations in Advanced Inputs, the max allowed CO_2 concentration can be specified and from the dilution equation, you can evaluate if the specified q_v is sufficient.

Displacement ventilation

By displacement ventilation cold air is supplied at floor level, which will give occasion for a vertical temperature gradient in the room. The vertical temperature distribution depends on many factors – placement, extent and convection flow of heat sources, room geometry and air flow. The 50%-rule, which applies that 50% of the temperature increase from supply to exhaust takes place at floor level, is a simple model. But laboratory tests at Lindab have shown that the temperature gradient most often is larger in the lower part than in the upper part of the room. LindQST ICD uses the following model to give a better prediction of the temperature gradient in the occupied zone, which is defined as the temperature in 1.1 m – the temperature in 0.1 m (T_r-T_f).



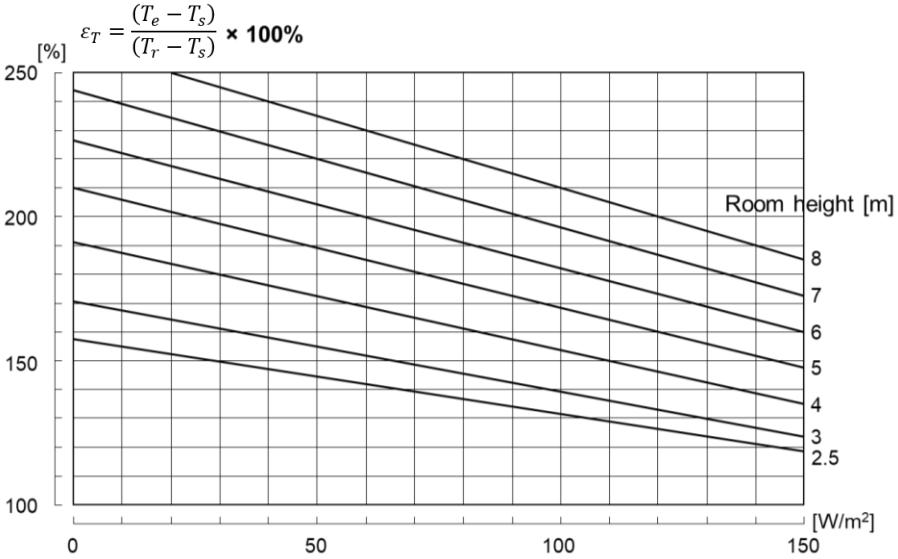
The assumption is, that the air temperature close to the floor (0.1 m) T_f is the mean value of room temperature T_r (in 1.1 m) and the supply temperature T_s. Hereby follows that the temperature increase between 0.1 m and 1.1 m (which is the temperature gradient in the occupied zone T_{gr}) can be determined as

$$T_{gr} = \frac{(T_r - T_s)}{2}$$

In addition, comes the temperature efficiency ε_T

$$\varepsilon_T = \frac{(T_e - T_s)}{(T_r - T_s)}$$

The temperature efficiency depends on the room height and the thermal load.



Combining the formula for temperature gradient in the occupied zone with the diagram, the formula for the temperature efficiency and the general formula for the heat balance gives us connection between the parameters (Φ_v, q_v, T_{gr}) and the room temperature T_r. Like for mixing, the three parameters (q_v, T_s, T_r) always must be specified, the required Φ_v can also be specified and will then be held up against the calculated Φ_v.

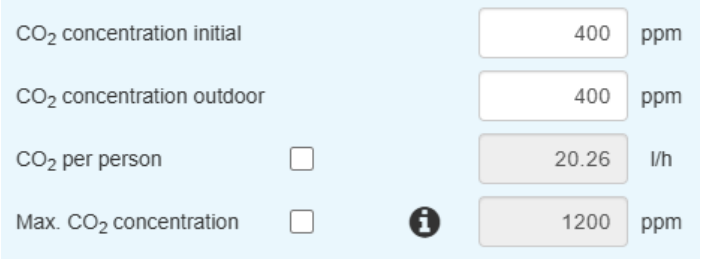
If activating CO_2 calculations in Advanced Inputs, the max allowed CO_2 concentration can be specified and from the dilution equation, it is evaluated if the specified q_v is sufficient.

CO_2 dilution equation

$$c = \left(\frac{q_m}{q_v} \cdot \left(1 - e^{-\frac{q_v}{V} \cdot \tau}\right) + (c_0 - c_i) \cdot e^{-\frac{q_v}{V} \cdot \tau}\right) \cdot 10^6 + c_i$$

c is the concentration of CO_2 in the room at the time τ implying ideal mixing [ppm] (ppm = cm3/m3)
q_m is added amount of CO_2 (depending of activity level) [m^3/h]
q_v is the air flow [m^3/h]
V is the volume of the room [m^3]
τ is the time [h]
c_0 is the start concentration (if set equal to c_i, this part of the equation is zero) [ppm]
c_i is the concentration of CO_2 in the supply air (outdoor air is normally 350-450) [ppm]

As default the added CO2 amount is calculated from number of persons and the metabolic rate selection (see the chapter about Thermal comfort), but you can also manually type in CO_2 per person.



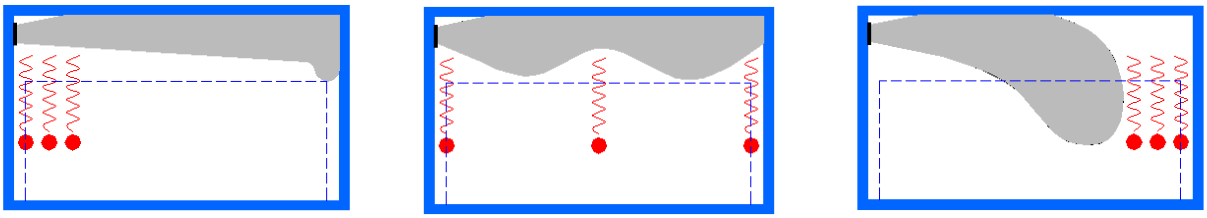
For comparing the result with the Max CO_2 requirement, you can select the categorization from B.3.1.3 Method 2 in EN 16798 or type in your own requirement.

Velocity in the occupied zone

In the following chapter, different methods and equations are described for products used in mixing ventilation and for products used in displacement ventilation.

Mixing ventilation air diffusers

Velocity in the occupied zone is calculated as the impulse velocity (from supply air throws and the supply temperature compared to room temperature). It will always be dependent of the placement/distribution of the thermal loads in the room.



Taking thermal upwards convection air flows into account requires very detailed information of all thermal conditions in the room and advanced tools to solve comprehensive and complicated equation systems (e.g. CFD). In LindQST ICD simpler formulas are used based on air jet theory using flow elements and experience from laboratory tests. Therefore, calculations must be considered as a realistic estimate of the maximal velocity in the occupied zone, which is important to consider in the design phase of a ventilation project.

The occupied zone is defined in the Room setup tab with the following defaults for mixing ventilation: height h_{occ} = 1,8 m, height of wall obstacle x_{occ} = 0.0 m (floor level). The value in Offset set from wall (default 0,6 m) is only a visual definition; this value will not affect the calculation for mixing ventilation.

Air jet velocity

The air jet velocity v_x of a diffuser decreases with the distance from the diffuser. The general formula of the velocity decay for a free and wall jet is:

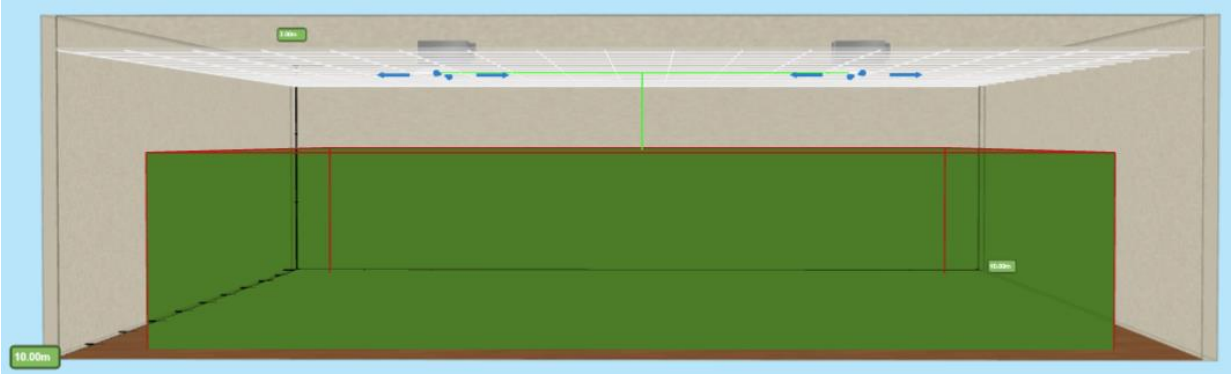
$$\text{free jet } v_x = \frac{K_a}{\sqrt{2}} \cdot \frac{\sqrt{A_0}}{x} \cdot v_0 \qquad \text{wall jet } v_x = K_a \cdot \frac{\sqrt{A_0}}{x} \cdot v_0$$

K_a is a diffuser constant, determined from diffuser specific data, e.g. throw length
A_0 is the freearea of the diffuser [m^2]
x is the distance from the diffuser [m]
v_0 is the supply velocity [m/s]

When the diffuser is placed ≤ 300 mm from the ceiling the formula for wall jet will be used.

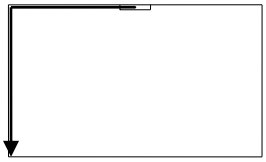
Diffusers with horizontal air pattern

The air jet velocity is calculated from the formulas of free and wall jet with the critical length l_{crit} is inserted instead of x. The critical length can be shown as a green/red indicator in the 3D-view.

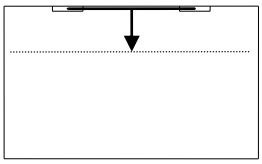


l_{crit} will be determined as the <u>minimum</u> of following distances:

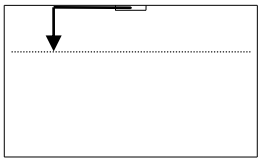
- Horizontal distance to the nearest wall + vertical distance to a possible obstacle at the wall (obstacle at wall is as standard the floor)



- Horizontal distance to the nearest air jet (weighted by local air flow and corrected by a mixing factor) + vertical distance to the occupied zone



- (When cooling) Penetration length + vertical distance to the occupied zone



Penetration length x_m is determined as $x_m = \frac{K_{sa} \cdot K_a}{\sqrt{Ar}} \cdot \sqrt{A_0}$ [m]

K_{sa} is determined from laboratory tests

Ar is Archimedes number defined as $Ar = \frac{g \cdot \beta \cdot \sqrt{A_0} \cdot |T_e - T_s|}{v_0^2}$

β and g is coefficient for thermal expansion [$°C^{-1}$] and gravity [m/s^2] respectively

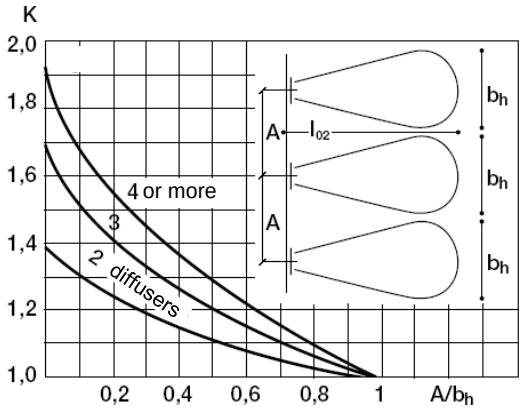
T_e and T_s is extract and supply temperature respectively [°C]

Wall diffusers:

When two or more diffusers with parallel directed supply is placed with a mutual distance A, which is less than b_h, the throw is lengthened by

$l_{0.2}$(corrected) = K · $l_{0.2}$

K is a correction factor, which can be read from the diagram

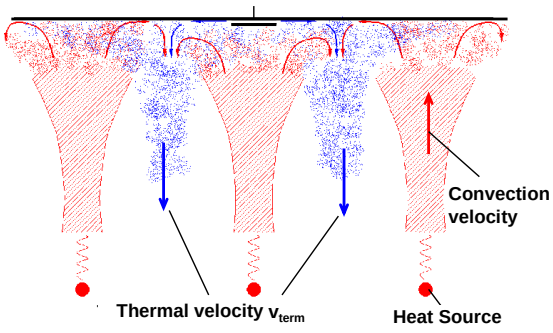


Supply with cold air:

The velocity calculations are based on the above-described flow elements, and for ceiling diffusers with horizontal dispersal pattern, Coanda effect is a prerequisite. Therefore, LindQST ICD cannot be used for predicting “drop” from a diffuser, if air flow is low and temperature difference is high. Free hanging diffusers with horizontal dispersal pattern has the same restriction, although without a ceiling there cannot be Coanda effect, but it is a prerequisite for the calculations, that the air flows always have horizontal dispersal (also if the dispersal pattern goes upwards in reality).

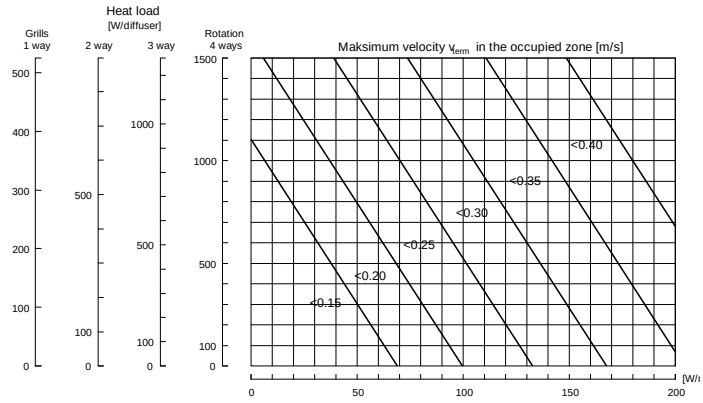
For calculating penetration length, i.e. where the air current will detach from the ceiling (for free hanging diffusers from a non-existing ceiling) and start going downwards against the occupied zone, the calculations imply Archimedes no., but it is also a prerequisite that all heat sources are evenly distributed over the whole floor area. I.e. in ICD it is not possible to simulate the penetration length extended or shortened by local intense heat sources or local heated surfaces, like a e.g. glass facade.

If “Include thermal velocity calculations” is activated in Advanced Inputs / Calculation, then for diffusers used in mixing, an additional empirical thermal velocity will be calculated additionally. Heat sources in the room generate convection air flows, which intensifies the drop effect of the cold supply air. These thermal conditioned downwards air flows cause a draught risk separately from the velocity of the supply air jet.

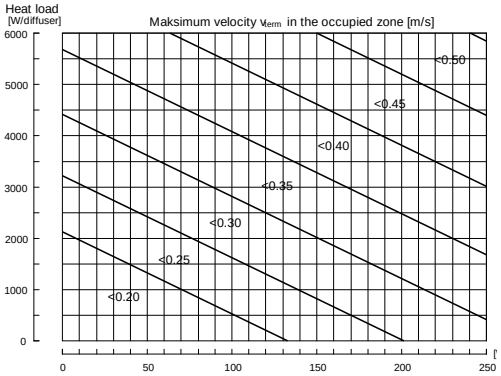


This velocity is called ”thermal velocity” to avoid confusion between the concept of convection velocity, which normally is perceived as the up going flow above heat sources. The thermal velocity is determined from an empirical model with the heat load [W/m^2], number of diffusers [W/diffuser], air pattern (1-, 2-, 3-, 4-way, rotation etc.), and the height of the room.

Height = 2,5



Height ≥ 4 m:



If thermal velocity has a higher value than the air jet velocity, then the thermal velocity is shown as the result.

<u>Supply with heated air at ceiling height:</u>

For diffusers with *horizontal air pattern*, in ICD there is no implementation of the impact when supplying with heated air. The air jet velocity calculations will be calculated as if it was isothermal conditions. Supplying with heated air means that the air current has buoyancy, and the practical impact is that the supply air will rise upwards against the ceiling, if the diffuser is not already integrated in the ceiling. The heated supply air can enter the occupied zone when it is fully mixed with the room air, if colliding with another air jet, or the most likely; “climb” down a surface, especially if the surface is cold, e.g. an outdoor window.

The ICD velocity calculations for now have its focus on showing potential draft risk, and that is rarely an issue for heated supply air. Therefore, ICD is not yet suitable for assessing whether the supply air reaches the occupied zone or not during heated air supply with horizontal air pattern.

Diffusers with vertical air pattern

Some supply air diffusers can be set to a vertical distribution pattern. That is mainly a function used for spaces where people stay for a short time, typically with low thermal comfort requirements, e.g. in shops, corridors etc. And it is often spaces which has its heating from supply air, although part of the year maybe also with cooling.

The air jet velocity is calculated by a variant of the formula for a free air jet

$$\text{Hot jet } v_y = v_0 \frac{K_a}{\sqrt{2}} \frac{\sqrt{A_0}}{y} \left(1 - \frac{Ar}{{K_{type,v}}^2 \cdot K_a} \left(\frac{y}{\sqrt{A_0}}\right)^2\right)^{1/3}, \; y_m = K_{type,v} \sqrt{A_0 \frac{K_a}{Ar}}$$

$$\text{Cold or isothermal jet } v_y = v_0 \frac{K_a}{\sqrt{2}} \frac{\sqrt{A_0}}{y} \left(1 + \frac{Ar}{{K_{type,v}}^2 \cdot K_a} \left(\frac{y}{\sqrt{A_0}}\right)^2\right)^{1/3}, \; y_m = H$$

Ar is Archimedes number defined as $Ar = \frac{g \cdot \beta \cdot \sqrt{A_0} \cdot |T_e - T_s|}{u_0^2}$

K_{type} is a type constant, which is determined from turning point data

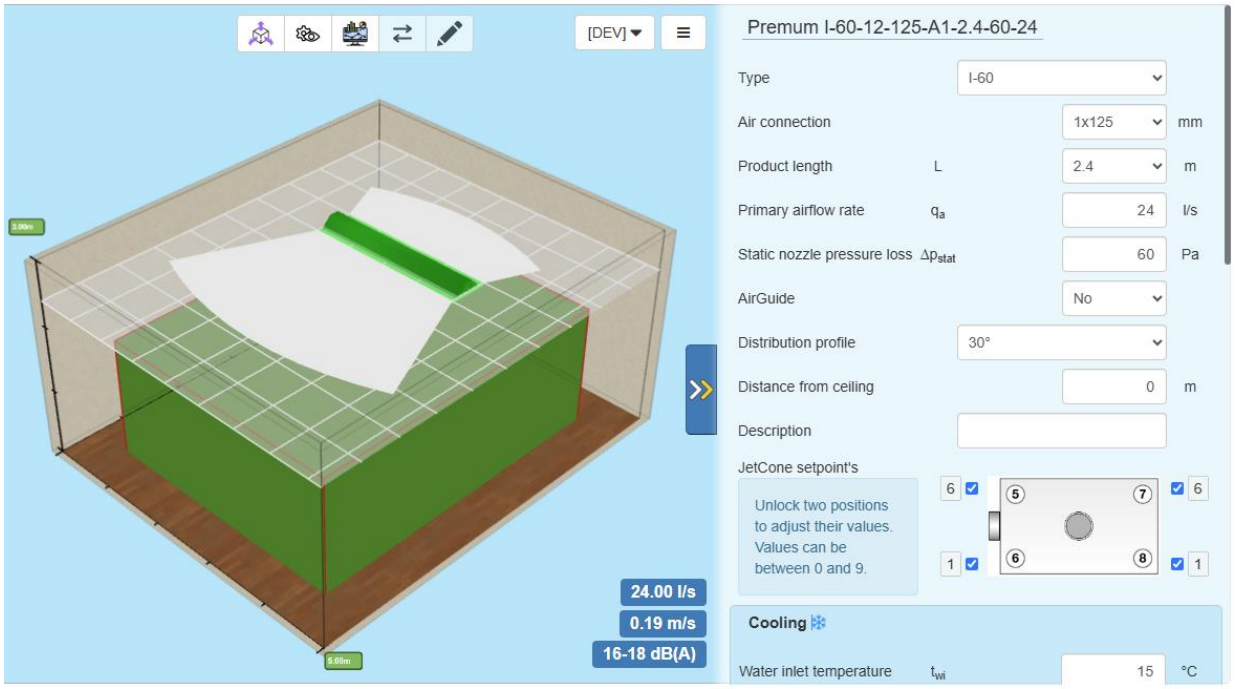
y is the vertical distance from the diffuser to the occupied zone [m]
y_m is the distance to where the air jet turns/stops [m]
H is the distance from diffuser to floor

The resulting velocity in the occupied zone is only determined from the air jet velocity.

Mixing ventilation Active Chilled Beams (ACB) and linear diffusers

Where supply air diffusers can be considered as point sources of air, ACBs and linear diffusers are line sources of air. So, in ICD they have a different calculation method for velocity in the occupied zone based on the cooling capacity of the air.

As ACBs and linear diffusers can have quite different air flows on each outlet side, the velocities are calculated from the local cooling effect from each side of the ACB / linear diffuser. On an ACB there are more possibilities regarding the air distribution, e.g. distribution profile, AirGuide as well as JetCone setpoints, which distributes the air from 4 fix points in each corner of the ACB.



Penetration length and vertical velocity

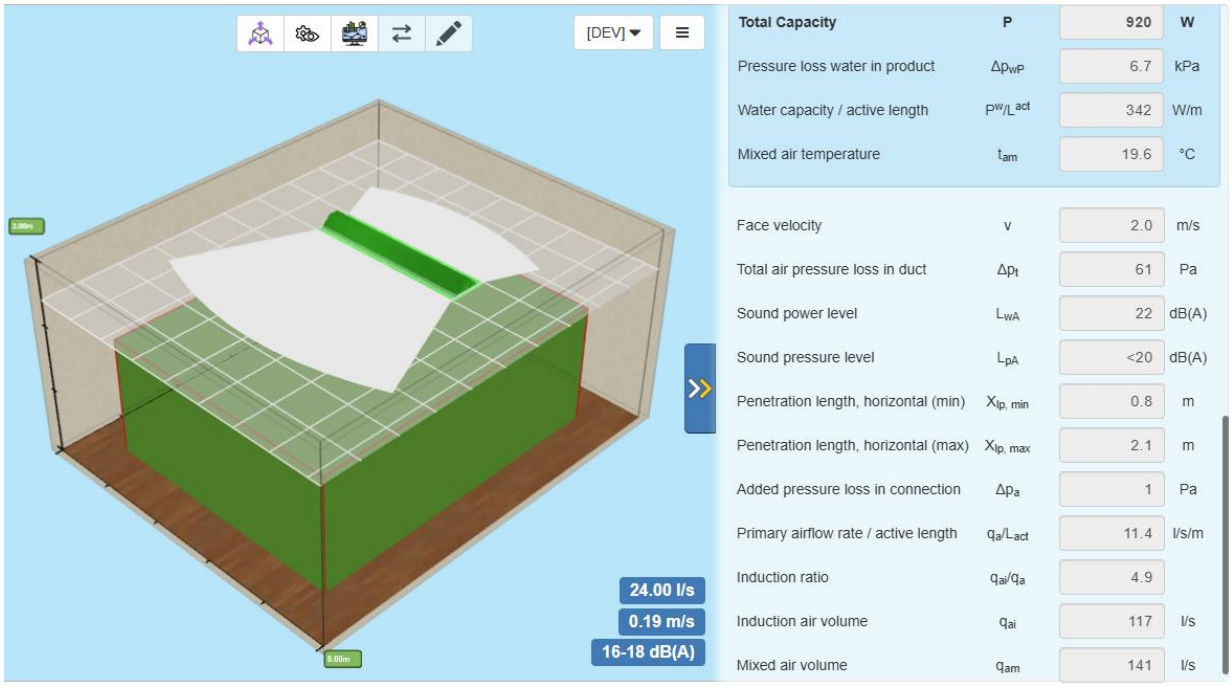
The penetration length, where the air currents detach from the ceiling and start going downwards against the occupied zone, is calculated by

$$x_{lp} = \left(\frac{K_{sp} \cdot K_p^2 \cdot q_{mix/m}^{4/3}}{\Delta T^{2/3} \cdot h_0} \right)$$

K_{sp} is determined from laboratory tests
K_p is a diffuser constant for a plane air jet, determined from diffuser specific data
$q_{mix/m}$ is the (mixed) air flow per outlet length [m^3/s]

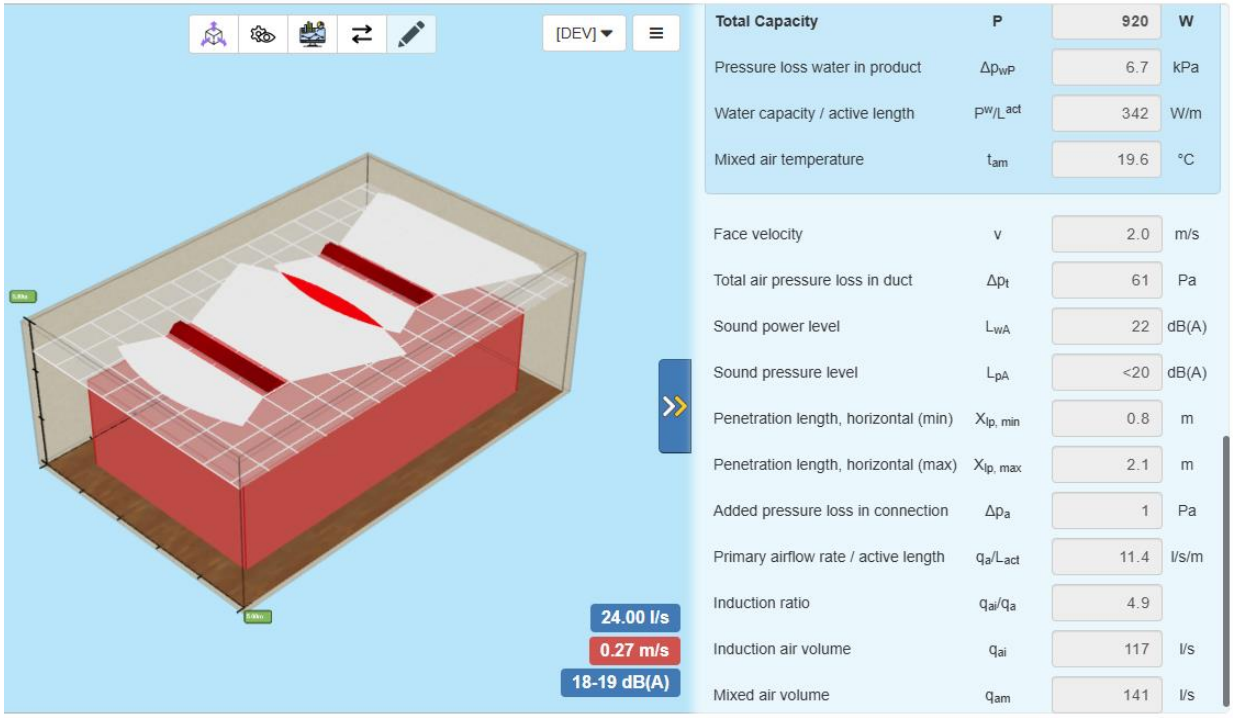
ΔT is the temperature difference between room temperature and the mixed air temperature [K]
h_0 is the outlet opening height

X_{lp} is calculated for each flow point and the minimum and maximum value is presented:

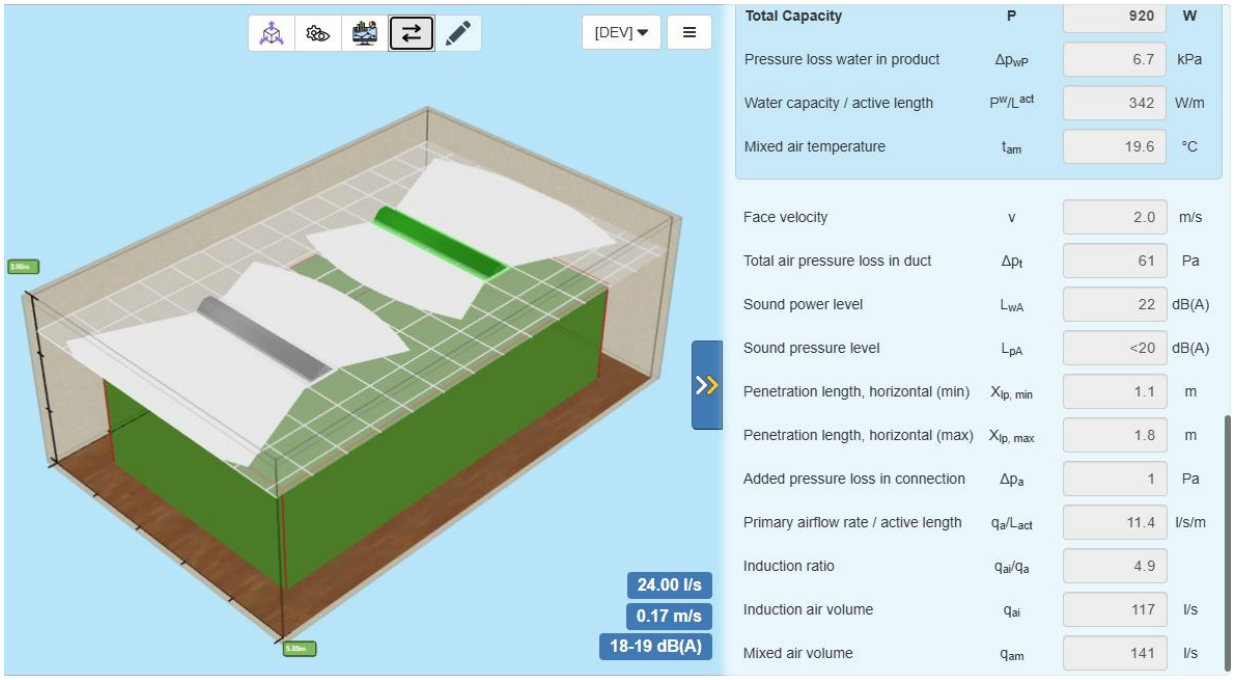


From each penetration length perimeter, a fall line length of the air current is determined, and a downwards vertical velocity against the occupied zone is calculated from the local cooling capacity per fall line length.

Cooling from intersecting fall lines from more ACBs / linear diffusers increases the velocity significantly.



However, for ACBs with JetCones you can adjust the air distribution more optimally, keeping the cooling capacity.

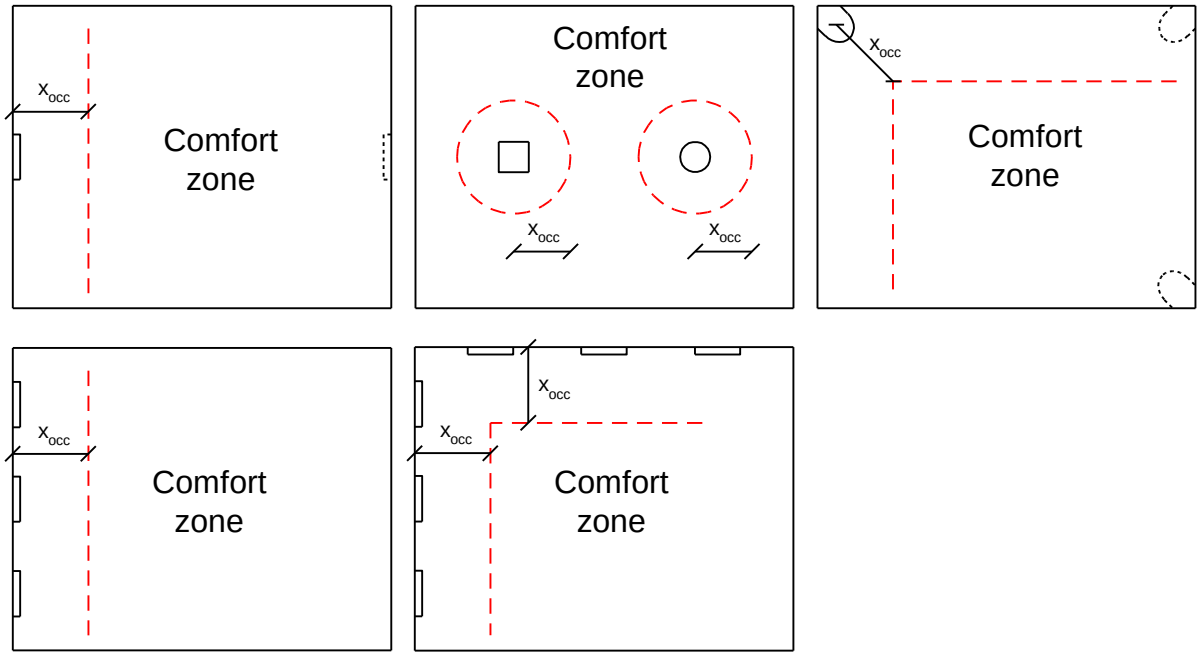


As cooling is the base for the velocity calculations, this calculation method cannot be used for isothermal (and nearby isothermal situations), as that will underestimate the velocities.

Activating “Include thermal velocity calculations” in Advanced Inputs / Calculation does not apply to ACBs and linear diffusers.

Displacement ventilation

The occupied zone (comfort zone) is defined in with the following default settings for displacement: distance from diffuser x_{occ} = 1.5 m and max velocity requirement of 0.2 m/s.



Air jet velocity

The direct velocity caused by the single diffuser is determined as

$$v_{x,\alpha} = K_{Dr,\alpha} \cdot \frac{q}{x_{occ,\alpha}}$$

the index α indicates the angle according to the centreline of the diffuser (symmetry axis)

q is the flow of the diffuser [m^3/s]

$x_{occ,\alpha}$ is the distance between diffuser and occupied zone in the angle α, where the velocity is highest [°] $x_{occ,\alpha} = \frac{x_{occ}}{cos(\alpha)}$

$K_{Dr,\alpha}$ is a velocity decay coefficient depending on diffuser specific data (near zone) and Ar_D

Ar_D is Archimedes number defined as $Ar = \frac{g \cdot \beta \cdot h \cdot |T_r - T_s|}{v_0^2}$

β and g is coefficient for thermal expansion [$°C^{-1}$] and gravity [m/s^2] respectively

h is the height of the diffuser [m]

T_r and T_s is room and supply temperature respectively [°C]

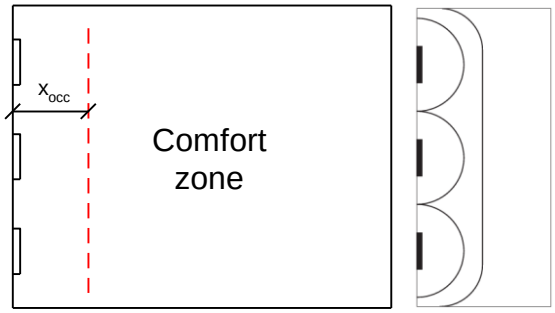
Lindab displacement diffusers (COMDIF) can be adjusted to both small diffusion (radial) and large diffusion (semi radial, where the supplied flow is largest in directions parallel to the wall and less in forward direction). At large diffusion will the velocity typically be highest in an angle of α = 45°.

Floor velocity

When more diffusers are placed at the same wall or around corner at two adjacent walls there is a risk of velocities at the floor in the occupied zone, which are higher than the air jet velocity from the single diffuser, where the air jet enters the occupied zone. When calculating this floor velocity, the programme distinguishes between if the diffusers are placed at the same wall or if they are placed around a corner.

<u>Diffusers placed at the same wall</u>

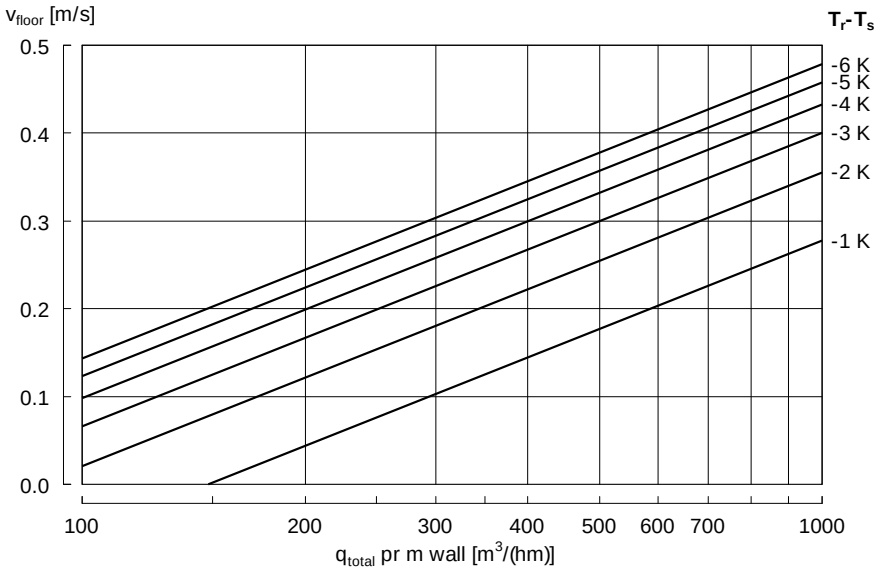
The air flow from closely placed (but evenly distributed) diffusers will at some distance run together to a plane flow regardless of the air pattern from the single diffusers is plane, radial or different shaped.



The velocity in this linear front is only dependent of the air flow per m wall and the temperature difference between room and supply but not of the supply velocity of the diffusers.

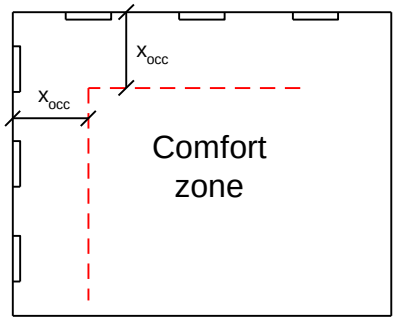
$$v_{floor} = k_1 \cdot log\left(\frac{q_{total}}{L}\right) + (k_2 \cdot ln(T_r - T_s) - k_3)$$

k_1, k_2 og k_3 are constants
q_{total} is the total air flow for all diffusers [m^3/h]
L is the length of the wall, which the diffusers are placed at [m]
T_r and T_s is room and supply temperature respectively [°C]



Diffusers placed around a corner

The max floor velocity in the occupied zone is dependent of the number of diffusers and how they are placed, as diffusers placed close to each other can give occasion for jets between diffusers and thereby relative high velocities in the occupied zone.



The velocity is calculated by

$$v_{floor} = K_{Dw} \cdot \left(\frac{q_{total}}{w}\right)$$

K_{Dw} is a coefficient dependent of $\frac{T_r - T_s}{q_{total}^2}$
q_{total} is the total air flow for all diffusers [m^3/s]
w is a characteristic (geometrical) unit, which is calculated from the length of a wall and the distance from the corner to the most distant diffusers
T_r and T_s is room and supply temperature respectively [°C]

Sound calculations

In Lindab sound diagrams and diffuser data the A-weighted *sound power level* L_{WA} is specified. It is specified for a diffuser and the plenum box (if any) connected with 1 m straight duct the same size as the diffuser. *Sound pressure level* is a measure for the intensity of the sound, that is the pressure vibrations we perceive, while *sound power level* is a parameter, which characterizes the sound source. Both things are normally specified in the unit dB (decibel), which can cause confusion.

Sound power (L_W)

Is the power a sound source (e.g. a machine) sends out in the shape of sound. The sound power is measured in Watt (W) and is most often specified as sound power level in decibel (dB) or dB(A).

Sound pressure (L_p)

Is a measure for the intensity of the sound, characterized by the pressure vibrations perceived by the ear or measured with a microphone on a sound level meter. Sound pressure is measured in Pascal (Pa) and is most often specified as sound pressure level in decibel (dB) or dB(A).

The sound property of a diffuser is specified as sound power.

$$\text{Sound power } L_W = 10 \cdot \log\left(\frac{N}{N_{re}}\right) [dB]$$

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

$$\text{Sound pressure } L_p = 20 \cdot \log\left(\frac{p}{p_{re}}\right) [dB]$$

p is the actual sound pressure [N/m^2]
$p_{re}=2\times10^{-5}$ N/m^2 is the reference sound pressure

The room attenuation D [dB] is the difference between sound power level and sound pressure level

$$L_p = L_W - D$$

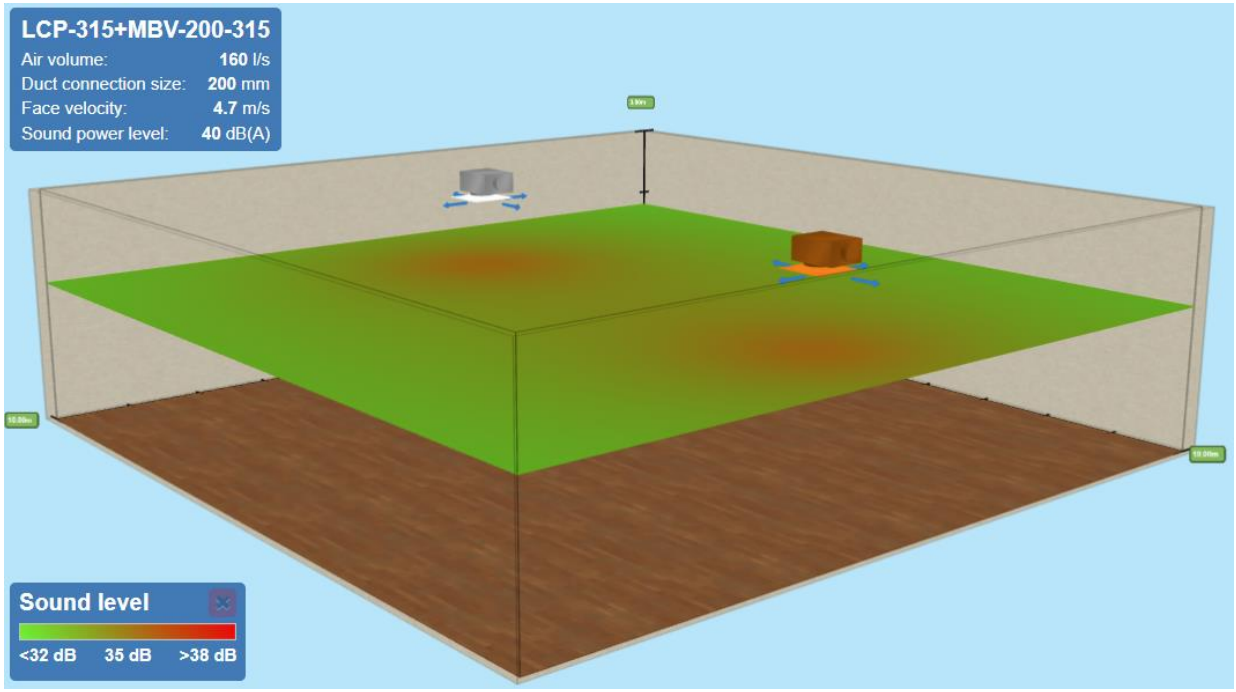
When the diffusers are placed in the room the sound pressure level is calculated in numerous points in the plane = height of the occupied zone. In each point the placement in the room of every diffuser is considered.

The sound pressure level from diffusers in one point is calculated by

$$L_{p,p1} = \left(L_{W,d1} + 10 \cdot log\left(\frac{Q}{4 \cdot \pi \cdot r_{p1-d1}^2} + \frac{4}{A}\right)\right) \oplus \left(L_{W,d2} + 10 \cdot log\left(\frac{Q}{4 \cdot \pi \cdot r_{p1-d2}^2} + \frac{4}{A}\right)\right) \oplus \dots$$

Q is the direction factor, which is dependent of the placement of the diffuser. In most cases it is assumed to be 2 (placed by ceiling)
n is the number of diffusers
A is the room absorption $A = 0{,}16 \cdot \frac{V}{T_s}$
V is the volume of the room [m^3]

T_s is the reverberation time of the room [s]
$L_{W,d1}$ and $L_{W,d2}$ is the sound power from each diffuser respectively
$r_{p1\text{-}d1}$ and $r_{p1\text{-}d2}$ is the distance from the point to the diffuser
$\oplus$ is logarithmic addition determined by: $L_1 \oplus L_2 = 10 \cdot log\left(10^{\frac{L_1}{10}} + 10^{\frac{L_2}{10}}\right)$



Thermal comfort

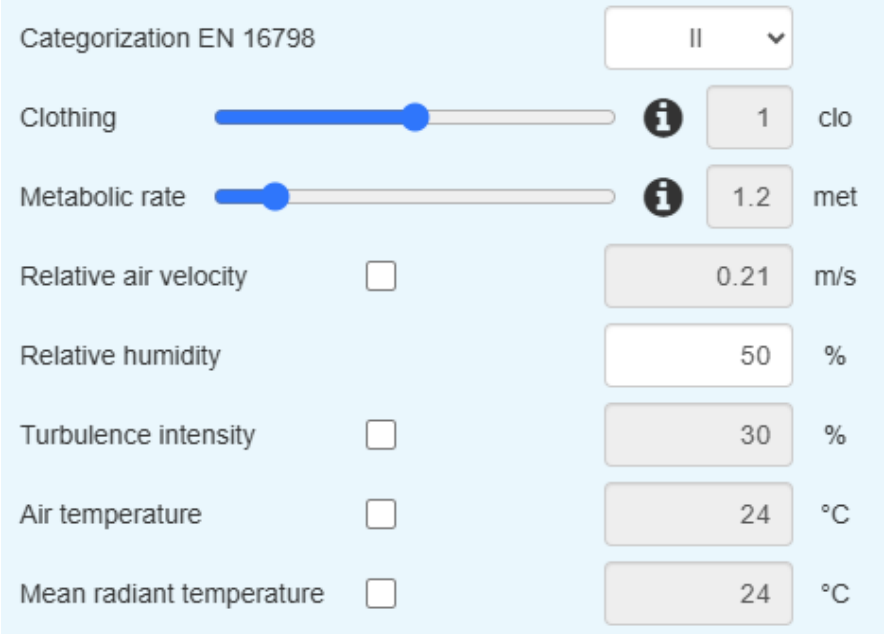
In LindQST ICD we have implemented calculations for thermal comfort parameters. If activating thermal comfort calculations in Advanced inputs, you will get results for local thermal discomfort factor DR (Draught Rate), but also the predictions for a thermal sensation for the body as a whole; PMV (Predicted Mean Vote) and PPD (Predicted Percentage Dissatisfied). Calculation of T_{op} (Operative temperature) is also included.

Input parameters for the calculations

In addition to the basic ICD inputs, like ventilation rate (supply air flow) and room size (volume), additional input parameters are necessary. It is thermal comfort factors, which can be grouped into two categories.

Personal factors – characteristics of the occupants: Clothing level and Metabolic rate

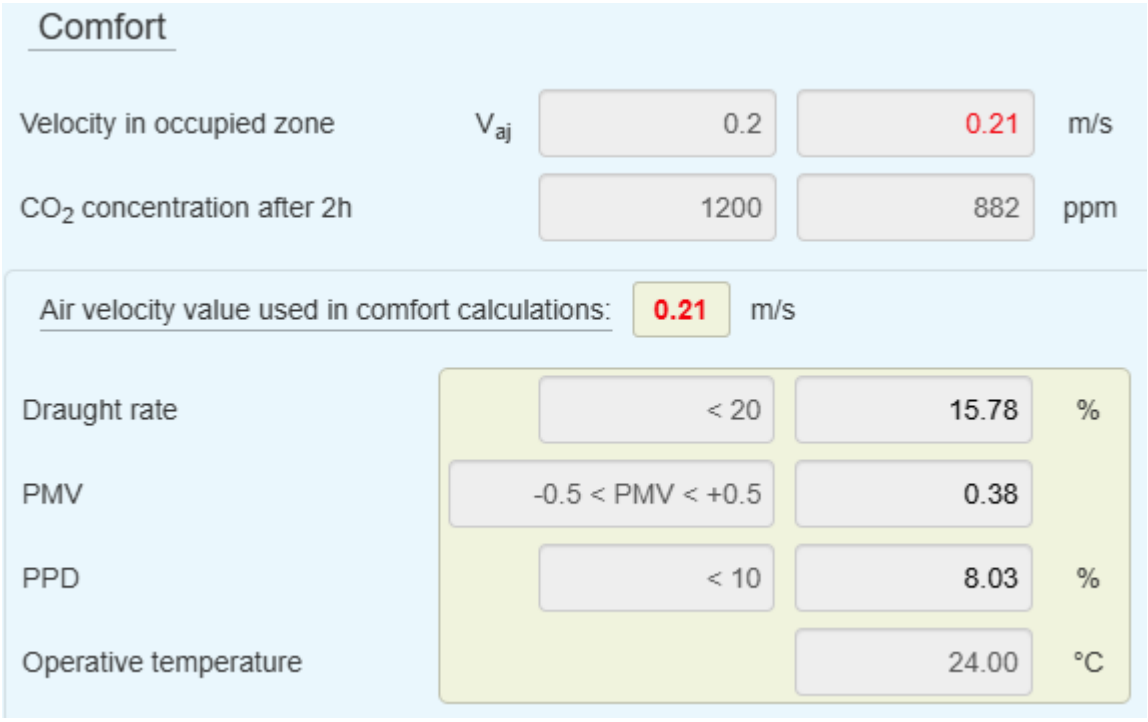
Environmental factors – conditions of the thermal environment: Relative air velocity, turbulence intensity, relative humidity, air temperature and mean radiant temperature.



The categorizations from the EN 16798 standard Annex B (Default criteria for the indoor environment) are used as requirements for comparison with the results. Each input parameter has either an info box or a helping text for suitable values, which appears at mouseover. For parameters with a tick mark selection, you can either use the defaults or manually type in the desired value.

Calculation results

The thermal comfort results are presented like this:



PMV (Predicted Mean Vote)

The Predicted Mean Vote (PMV) model stands among the most recognized thermal comfort models. The model is using heat-balance equations and empirical studies about skin temperature to define comfort. Standard thermal comfort surveys ask subjects about their thermal sensation on a seven-point scale from cold (−3) to hot (+3).

+3	Hot
+2	Warm
+1	Slightly warm
0	Neutral
-1	Slightly cool
-2	Cool
-3	Cold

Equations are used to calculate the predicted mean vote (PMV) of a group of subjects for a particular combination of air temperature, mean radiant temperature, relative humidity, air velocity, metabolic rate, and clothing insulation. PMV equal to zero is representing thermal neutrality, and the comfort zone is defined by the combinations of the six parameters for which the PMV is within the recommended limits (−0.5 < PMV < +0.5).

The equations for calculating PMV are found in EN/ISO 7730 Chapter 4.

Calculate the PMV using Equations (1) to (4):

$$PMV = \left[0{,}303 \cdot \exp(-0{,}036 \cdot M) + 0{,}028\right] \cdot \left\{ \begin{array}{l} (M - W) - 3{,}05 \cdot 10^{-3} \cdot \left[5\,733 - 6{,}99 \cdot (M - W) - p_a\right] - 0{,}42 \cdot \left[(M - W) - 58{,}15\right] \\ -1{,}7 \cdot 10^{-5} \cdot M \cdot (5\,867 - p_a) - 0{,}0014 \cdot M \cdot (34 - t_a) \\ -3{,}96 \cdot 10^{-8} \cdot f_{cl} \cdot \left[(t_{cl} + 273)^4 - (\bar{t}_r + 273)^4\right] - f_{cl} \cdot h_c \cdot (t_{cl} - t_a) \end{array} \right\} \quad (1)$$

$$t_{cl} = 35{,}7 - 0{,}028 \cdot (M - W) - I_{cl} \cdot \left\{3{,}96 \cdot 10^{-8} \cdot f_{cl} \cdot \left[(t_{cl} + 273)^4 - (\bar{t}_r + 273)^4\right] + f_{cl} \cdot h_c \cdot (t_{cl} - t_a)\right\} \quad (2)$$

$$h_c = \left\{ \begin{array}{ll} 2{,}38 \cdot \left|t_{cl} - t_a\right|^{0{,}25} & \text{for } 2{,}38 \cdot \left|t_{cl} - t_a\right|^{0{,}25} > 12{,}1 \cdot \sqrt{v_{ar}} \\ 12{,}1 \cdot \sqrt{v_{ar}} & \text{for } 2{,}38 \cdot \left|t_{cl} - t_a\right|^{0{,}25} < 12{,}1 \cdot \sqrt{v_{ar}} \end{array} \right. \quad (3)$$

$$f_{cl} = \left\{ \begin{array}{ll} 1{,}00 + 1{,}290\, I_{cl} & \text{for } I_{cl} \leq 0{,}078 \text{ m}^2 \cdot \text{K/W} \\ 1{,}05 + 0{,}645\, I_{cl} & \text{for } I_{cl} > 0{,}078 \text{ m}^2 \cdot \text{K/W} \end{array} \right. \quad (4)$$

where

M is the metabolic rate, in watts per square metre (W/m^2);

W is the effective mechanical power, in watts per square metre (W/m^2);

I_{cl} is the clothing insulation, in square metres kelvin per watt (m^2 · K/W);

f_{cl} is the clothing surface area factor;

t_a is the air temperature, in degrees Celsius (°C);

$\bar{t}_r$ is the mean radiant temperature, in degrees Celsius (°C);

v_{ar} is the relative air velocity, in metres per second (m/s);

p_a is the water vapour partial pressure, in pascals (Pa);

h_c is the convective heat transfer coefficient, in watts per square metre kelvin [W/(m^2 · K)];

t_{cl} is the clothing surface temperature, in degrees Celsius (°C).

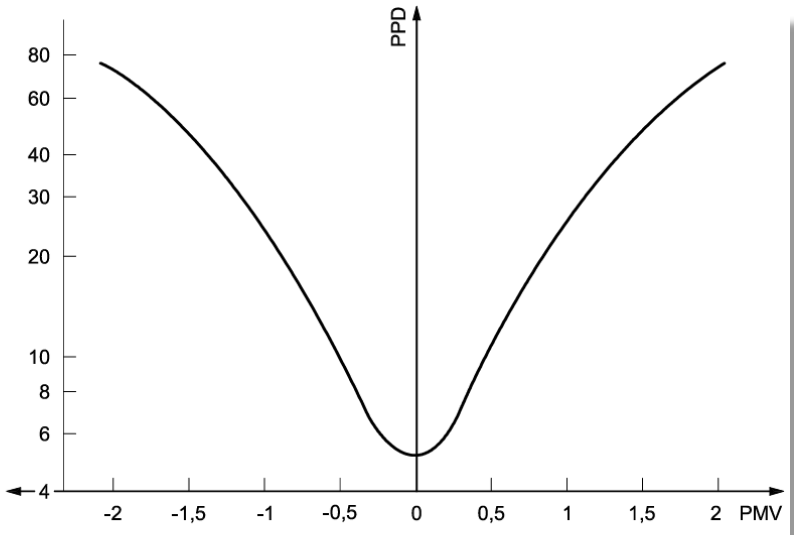
NOTE 1 metabolic unit = 1 met = 58,2 W/m^2; 1 clothing unit = 1 clo = 0,155 m^2 · °C/W.

PPD (Predicted Percentage Dissatisfied)

Although predicting the thermal sensation of a population is an important step in determining what conditions are comfortable, it is more useful to consider whether people will be satisfied. Therefore, another equation exists to relate the PMV to the Predicted Percentage of Dissatisfied (PPD).

The equation or PPD is found in EN/ISO 7730 Chapter 5.

$$PPD = 100 - 95 \cdot e^{(-0{,}03353 \cdot PMV^4 - 0{,}2179 \cdot PMV^2)}$$



Key
PMV predicted mean vote
PPD predicted percentage dissatisfied, %

Note that even when PMV is 0 = neutral, PPD will never be lower than 5%.

DR (Draught Rate) – local thermal discomfort

Where the PMV and PPD calculations express comfort/discomfort for the whole body, thermal dissatisfaction can also be caused by unwanted cooling or heating of one particular body part. This is known as local discomfort, and a common cause of local discomfort is draught. Other local discomfort factors: vertical air temperature difference, warm and cool floors and radiant asymmetry are not a part of LindQST ICD.

The equation for DR is found in EN/ISO 7730 chapter 6.

The discomfort due to draught may be expressed as the percentage of people predicted to be bothered by draught. Calculate the draught rate (DR) using Equation (6) (model of draught):

$$DR = (34 - t_{a,l})(\bar{v}_{a,l} - 0{,}05)^{0{,}62}(0{,}37 \cdot \bar{v}_{a,l} \cdot Tu + 3{,}14) \quad (6)$$

For $\bar{v}_{a,l} < 0{,}05$ m/s: use $\bar{v}_{a,l} = 0{,}05$ m/s

For $DR > 100$ %: use $DR = 100$ %

where

$t_{a,l}$ is the local air temperature, in degrees Celsius, 20 °C to 26 °C;

$\bar{v}_{a,l}$ is the local mean air velocity, in metres per second, < 0,5 m/s;

Tu is the local turbulence intensity, in percent, 10 % to 60 % (if unknown, 40 % may be used).

The model applies to people at light, mainly sedentary activity with a thermal sensation for the whole body close to neutral and for prediction of draught at the neck. At the level of arms and feet, the model could overestimate the predicted draught rate. The sensation of draught is lower at activities higher than sedentary (> 1,2 met) and for people feeling warmer than neutral. Additional information on the effect of air velocity can be found in Annex G.

Operative temperature t_o

The definition, measurement and calculation of operative temperature is described in EN ISO 7726 standard Annex G.

Defined as a uniform temperature of an imaginary black enclosure in which an occupant would exchange the same amount of heat by radiation plus convection as in the actual non-uniform environment.

The calculation of the operative temperature in ICD is based on air temperature, mean radiant temperature and air velocity.

$$t_o = \frac{t_a \cdot \sqrt{10} + t_r}{1 + \sqrt{10 \cdot v_a}}$$

t_a is the air temperature [°C]
v_a is the air velocity; either the occupied zone result or manually typed in [m/s]
t_r is the mean radiant temperature [°C]