6 Ventilation noise

ELEC-E5640 - Noise Control D

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Regulated values of HVAC noise in Finland

- HVAC = Heating, Ventilation and Air Conditioning (Building Service)
- Regulated values: Ministry of the Environment, Decree 796/2017
- They cover kitchens and other living rooms.
- They include also a penalty of 3–5 dB for impulsive and tonal sounds.
- Recommended values are given for other spaces. However, they are usually applied
- Quantities
  - A-weighted equivalent SPL, L_{Aeq}
  - A-weighted maximum SPL using fast time weighting F, L_{AFmax}
- Quantities cover 20-20000 Hz

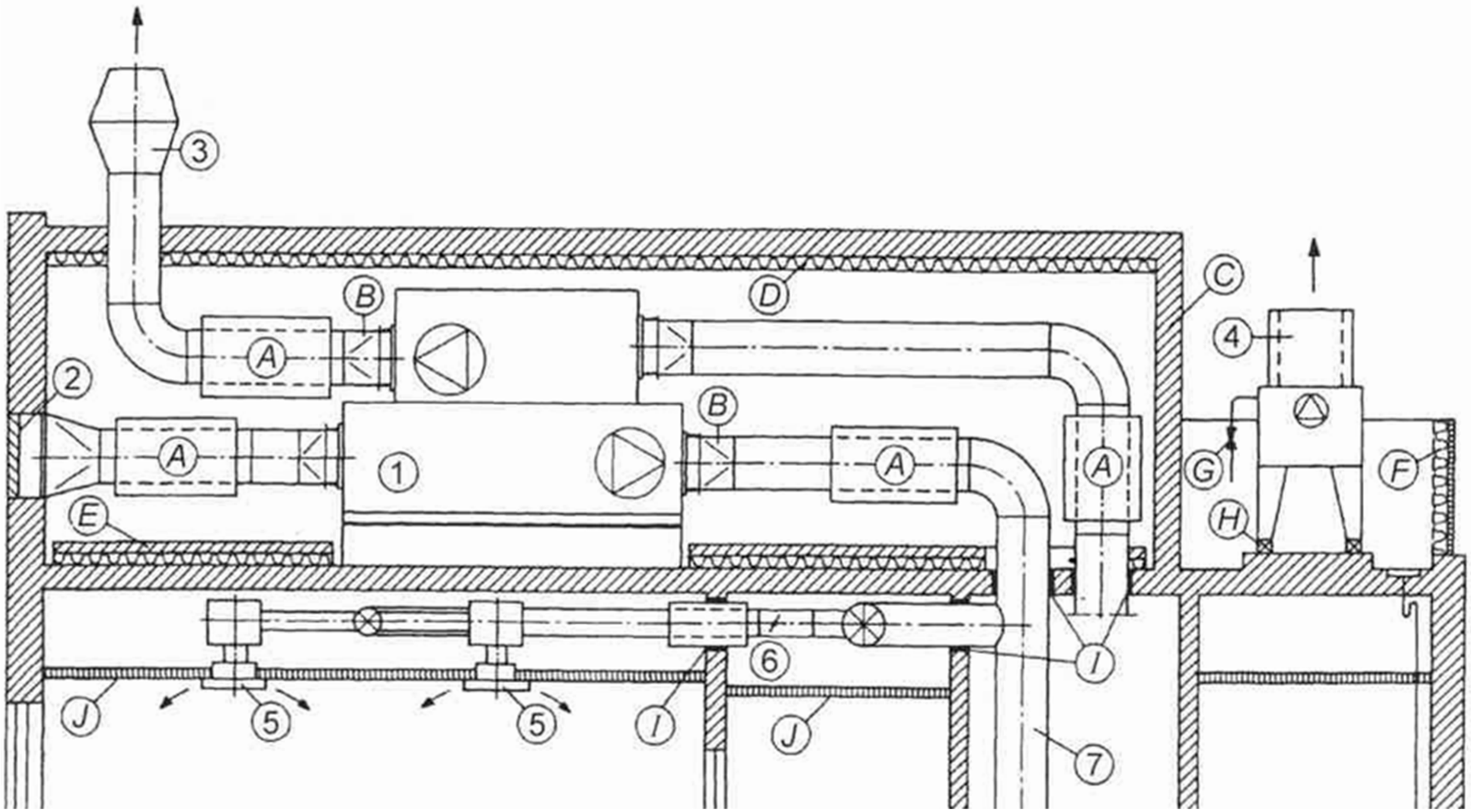
REGULATED VALUES

Room type	Largest allowed sound level			
	Continuous broadband sound		Impulsive or tonal sound	
	$L_{Aeq,T}$ [dB]	$L_{AFmax,T}$ [dB]	$L_{Aeq,T}$ [dB]	$L_{AFmax,T}$ [dB]
Residential room, accommodation room	28	33	25	30
Kitchen of an apartment, hobby room	33	38	30	35
Stairway	38	43	35	40
Yard, balcony or terrace	45	50	40	45

RECOMMENDED VALUES

Room type	Largest allowed noise level of building services	
	$L_{Aeq,T}$ [dB]	$L_{AFmax,T}$ [dB]
Teaching room	33	38
Teaching room in day-care center	28	33
Meeting room	33	38
Patient room, physician's room etc.	38	43
Operation room	33	38
Hobby room	33	38
Exercise room	38	43
Office room	33	38

Air conditioning room



Sound sources

1. Main fans/Ventilation unit
2. Fresh air terminal device
3. Exhaust air terminal device
4. Condenser blowers
5. Terminal devices
6. Air flow in dampers
7. Air flow in ducts

Noise control methods

A. Silencer
B. Streamlined air flow
C. Walls
D. Absorbents
E. Floors
F. Barriers
G. Flexible supports
H. Vibration isolation
I. Sealed holes
J. Linings

Halme&Seppänen, 2002

Calculation of SPL in the room for three main noise sources

- Fan (flow and engine noise):

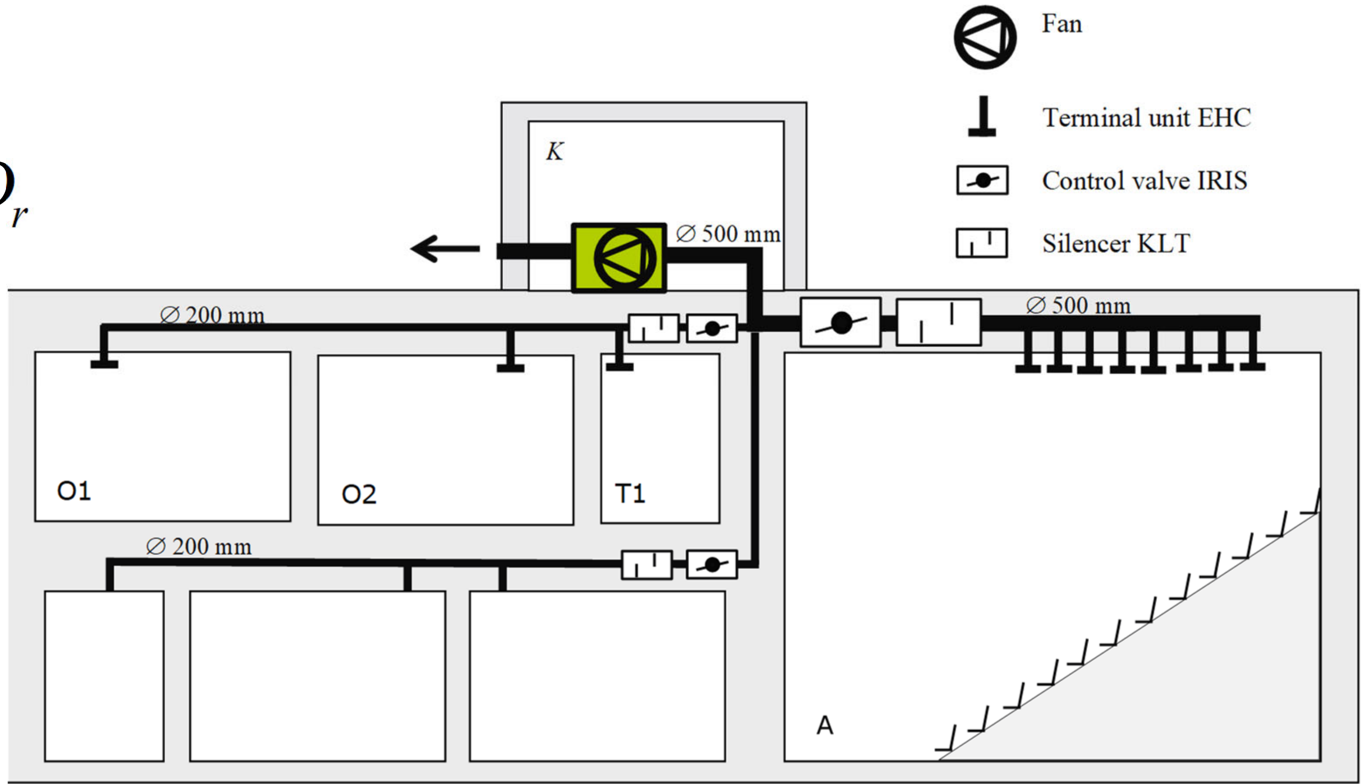
$$L_p = L_W - D_d - D_s - D_t + D_r$$

- Damper (flow noise):

$$L_p = L_W - D_d - D_s - D_t + D_r$$

- Terminal device (flow noise):

$$L_p = L_W + D_r$$



Attenuation due to

- L_W is the sound power level of fan/damper/terminal device
- Distribution in duct divisions/branches D_d
- Silencer D_s
- Duct termination or terminal unit D_t
- Room attenuation D_r (usually D_r=-4 dB is assumed)
- Ignored factors: Duct wall absorption and attenuation in bends

Calculations

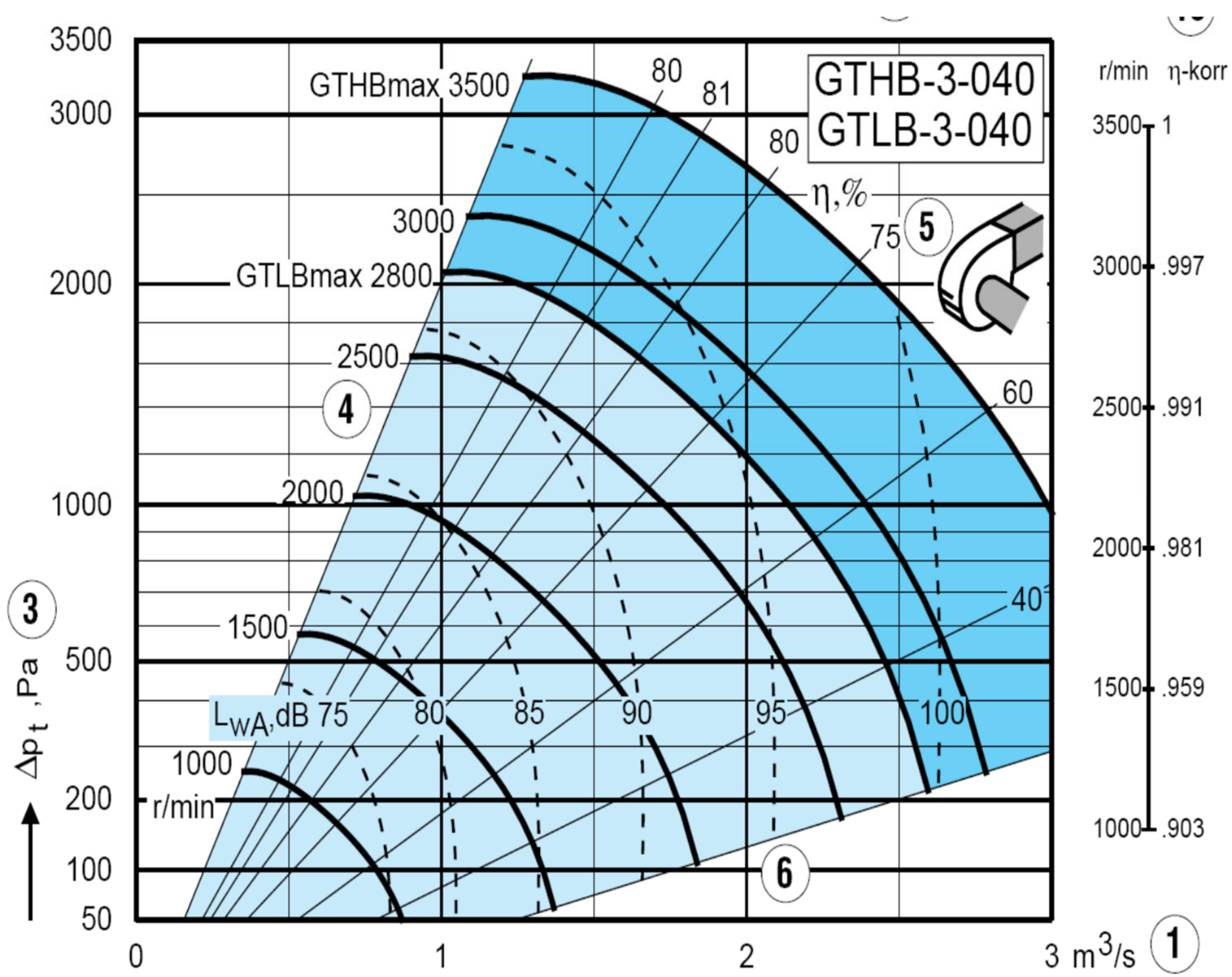
- Calculations are conducted for linear SPL in octave bands 63-8000 Hz
- Lacking octave band data is estimated
- First calculate separately the SPL caused by
  1. fan
  2. damper
  3. terminal device,
  4. ventilation room sound transmitted through the wall or floor
- Thereafter, take the logarithmic sum of uncorrelated sources
- Perform A-weighting in octave band values
- **Report total L_{Aeq} of all bands 63-8000 Hz**

$$L_{tot} = 10\lg \sum_{i=1}^{4} 10^{L_i/10}$$

Sound power levels L_W

Noise source: Fan

- Manufacturers declare the SWL of fans at different flow rates and pressures
- L_{WA} can be taken from the graph and the octave band values are obtained by applying a tabulated correction factor K



Fläktwoods
centrimaster
GT-3

Fan and K_{okt}

- Unweighted SWL at octave band i is obtained by

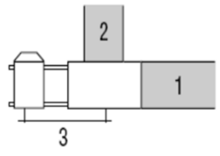
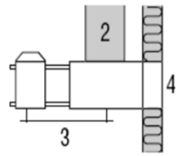
$$L_{W,okt,i} = L_{WA} + K_{okt,i}$$

- $L_{W,A}$ [dB] is the A-weighted SWL from the graph
- $K_{okt,i}$ [dB] is the correction at octave band i

- Three important values given

1. Pressure duct (*painekanavaan*)
2. Suction duct (*imukanavaan*)
3. Environment through the fan envelope (*ympäristöön*)

	Path Äänitie (s)	Rotation Speed range Pyörimisnopeusalue r/min	Korjaus K_{okt}, dB Oktaavikaista, keskitaajuus, Hz 63	125	250	500	1 000	2 000	4 000	8 000	$L_{WA(s)} - L_{WA}$ dB	$L_{Wt(s)} - L_{WA(s)}$ dB
	Painekanavaan (1) Pressure duct	0 – 964 965 – 1928 1929 – 3200	0 -2 -3	5 -1 -4	2 3 -4	-3 -3 -1	-6 -6 -6	-9 -9 -8	-14 -14 -13	-18 -17 -16	0 0 0	8,2 6,4 4,1
	Imukanavaan (2) Suction duct	0 – 964 965 – 1928 1929 – 3200	4 2 -2	3 -1 -5	0 0 -6	-3 -3 -2	-4 -5 -4	-9 -8 -7	-12 -10 -9	-14 -13 -14	0,4 0,3 0,8	7,8 6,2 3,3
Environment, duct installed	Ympäristöön, kanavaan liitetty puhallin (3)	0 – 964 965 – 1928 1929 – 3200	-8 -10 -12	-5 -8 -14	-6 -6 -11	-8 -10 -8	-11 -12 -10	-15 -16 -16	-22 -25 -24	-33 -36 -35	-6,1 -7,3 -6,4	6,1 5,8 3,1
Environment, without duct	Puhaltimen paine-aukkoon, vapaasti puhaltava puhallin (4)	0 – 964 965 – 1928 1929 – 3200	-9 -13 -17	0 -6 -9	0 1 -6	-3 -3 -1	-6 -6 -6	-9 -9 -8	-14 -14 -13	-18 -17 -16	-0,6 -0,5 -0,1	5,5 4,5 2,3

Äänitien kuvaus	Koejärjestely
1 = imukanavaan 2 = imukanavaan 3 = ympäristöön (kanavaan liitetty puhallin)	
4 = puhaltimen paineaukkoon (vapaasti puhaltava puhallin)	

Flow noise in general

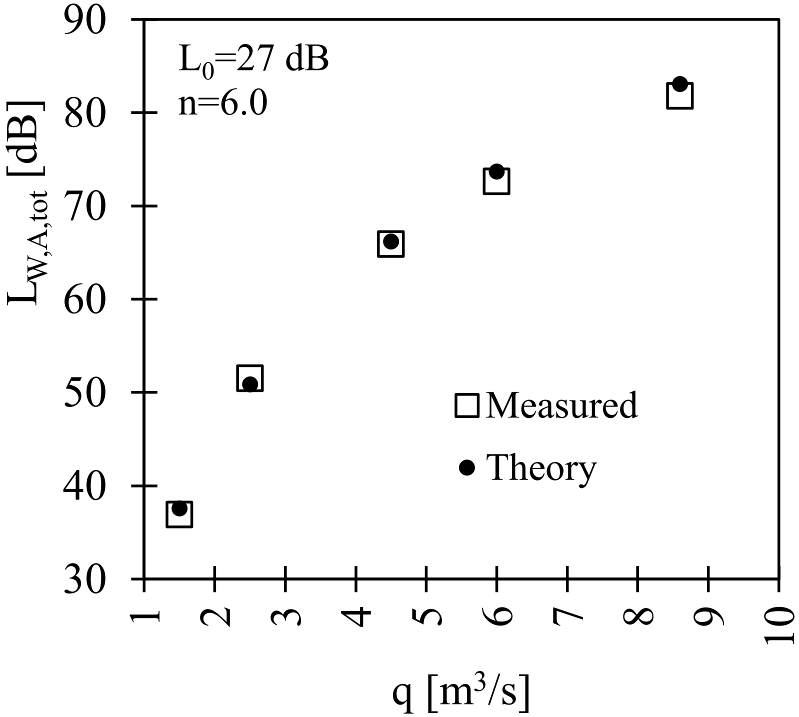
- SWL of flow noise:

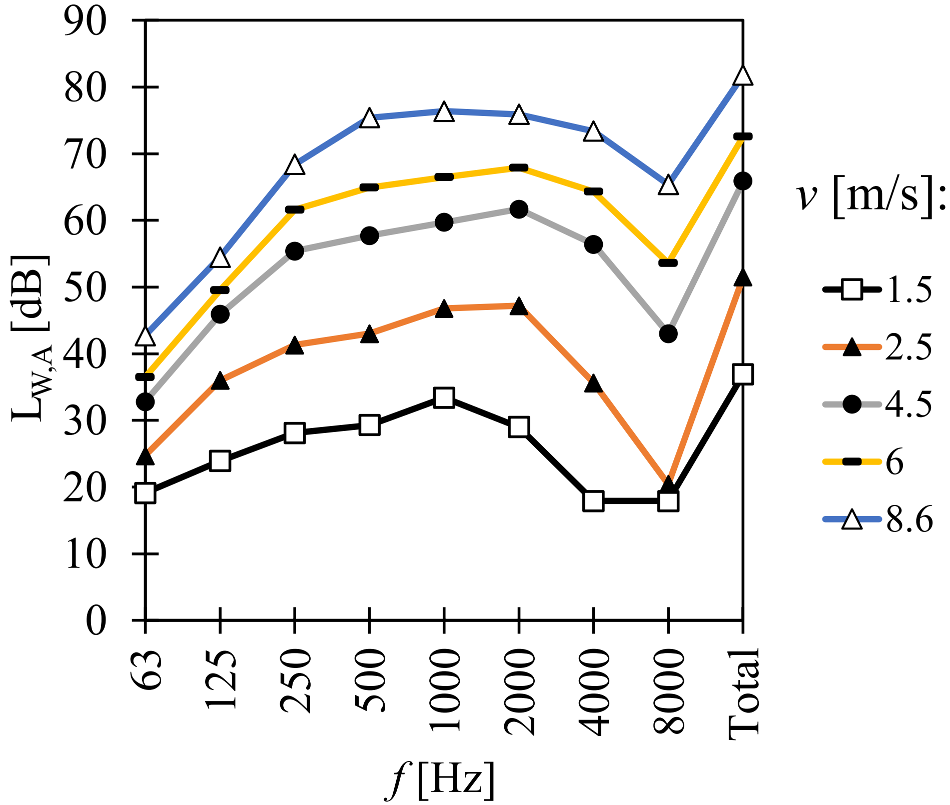
$$L_w = 10\lg S + 10\lg v^n + L_0$$

  - S [m²] is cross-sectional area of flow
  - v [m/s] is flow speed
  - L_0 [dB] is characteristic SWL when **q=1 m³/s** and **p=1 Pa.**
- Integer n expresses how the SWL is proportional to the flow speed:
  - Laminar flow: ***n*=5** (dipole radiation)
  - Turbulent flow: ***n*=6** (quadrupole radiation)
- The equation can be used to approximate the impact of v and S when L_0 is known
- The spectrum is constant although v or S changes.
- Doubling of flow speed v 15−18 dB SWL increment
- Doubling of flow area S: 3 dB SWL increment

$$L_w = 10\lg S + 10\lg v^n + L_0$$

- Outdoor terminal unit
- 200x200 mm





Flow noise in bare ducts

Highest recommended flow speed v [m/s] for three different target values of L_{Aeq}.

Square ducts	25 dB L_{Aeq}	30 dB L_{Aeq}	35 dB L_{Aeq}
Terminating duct	2.5	3	4
Branch duct	4	5	6

Round ducts	25 dB L_{Aeq}	30 dB L_{Aeq}	35 dB L_{Aeq}
Terminating duct	3.5	4	5
Branch duct	5	6.5	8

- Noise produced by flow in ducts is usually ignored.
- Designers usually apply the following absolute maximum flow speeds to avoid the exceedance of the target values of noise.

Streamlining reduces in Δp and L_{Aeq}

- Flow noise can transmit to the rooms
  - through terminal units, and
  - through the duct wall
- Square duct is noisier than round duct
- Discontinuous shapes are avoided
- Bends in square ducts are rounded
- Hindrances are avoided gently to achieve a small local Δp
- Slow changes in cross-sectional area: transition angles at most 30 degrees
- Perpendicular collisions to duct walls are avoided in T-junctions and duct terminations
- Ducts with large flow speeds are isolated with suspended ceilings and enclosures

Perpendicular joint should be rounded

Transition is too steep.

Benefits of one size larger duct diameter

Suure	Diameter A e.g., 125 mm	Diameter B e.g., 160 mm
Flow speed	v	» v/2
Pressure loss	Δp	» Δp/3
SWL of flow noise	L_w	» L_w - 7 dB
SWL of leak air noise	L_{w1}	» L_{w1} - 8… - 20 dB

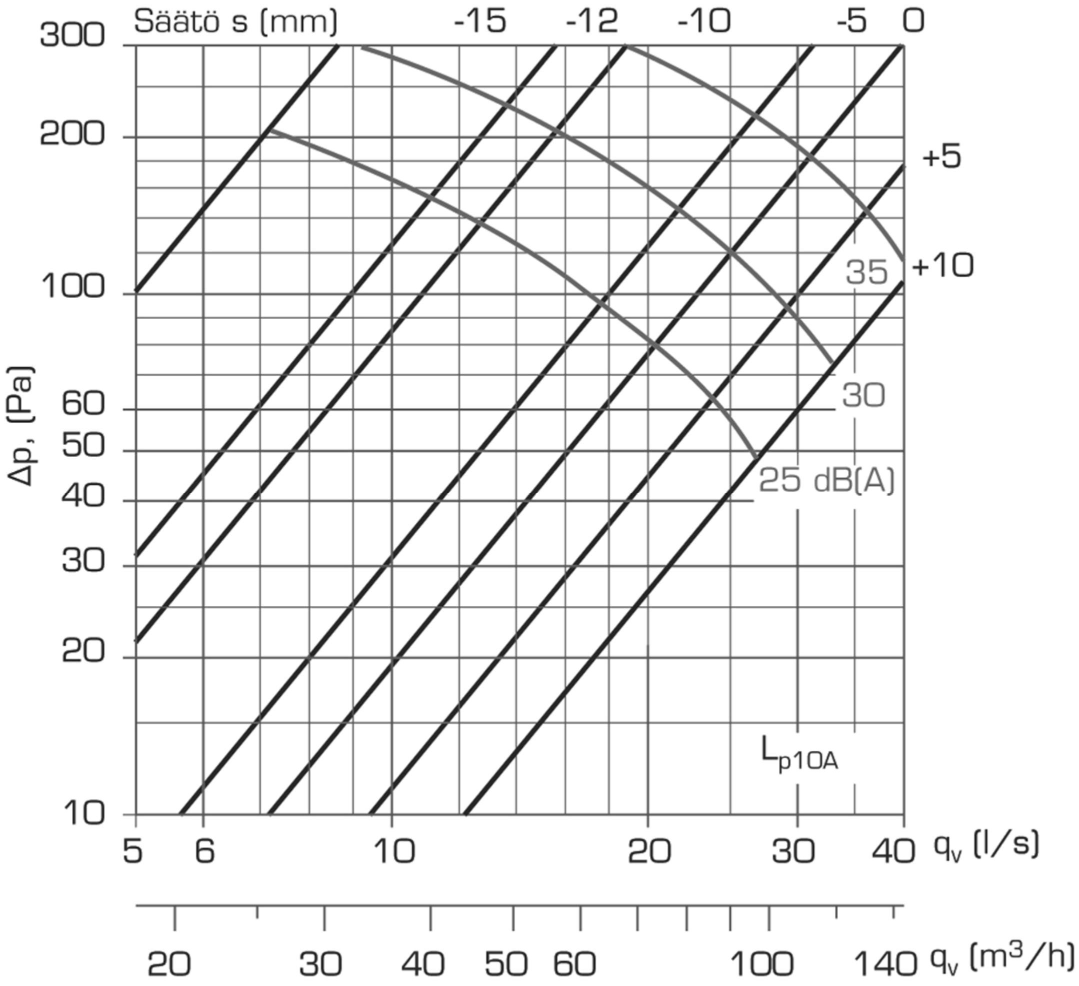
- Round duct diameters are 80, 100, 125, …600, 800 and 1000 mm.
- B is one step larger than A
- Small ducts have disadvantages:
  - Large electicity consumption due to larger Δp
  - Larger noise, more silencers are needed
  - Balancing problems
  - Increment of air flow rates is more difficult

Terminal devices

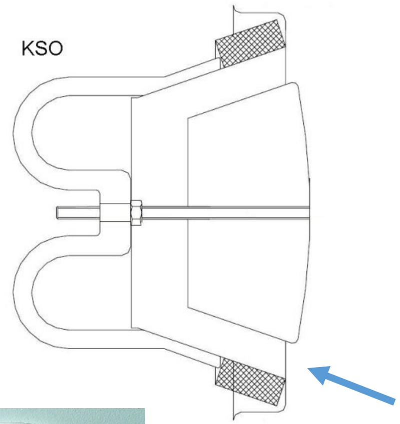
- Noise specifications are flow noise L_{p10A} and attenuation DL.
- Flow noise is caused by airflow through the component
- Attenuation concerns the terminal attenuation that the component produces for all noise produced before the component
  - Fan
  - Dampers
  - Flow noise in duct

Top: active chilling beam and outlet terminal unit.
Middle and bottom: different kinds of inlet terminal units having different airflow patterns and installation properties (Fläkt Woods Oy 2006).

Noise source: terminal device



$$L_{Wokt} = L_{p10A} + K_{okt}$$



s

L_{p10A}
25

	K	L_{p10}	L_W
125	-2	19.2	23.2
250	1	22.2	26.2
500	1	22.2	26.2
1000	0	21.2	25.2
2000	-5	16.2	20.2
4000	-9	12.2	16.2
8000	-23	-1.8	2.2
A		25	29

$$L_p = L_W + 10\log_{10}\left[\frac{4}{A}\right]$$

Fläkt Woods Oy 2007

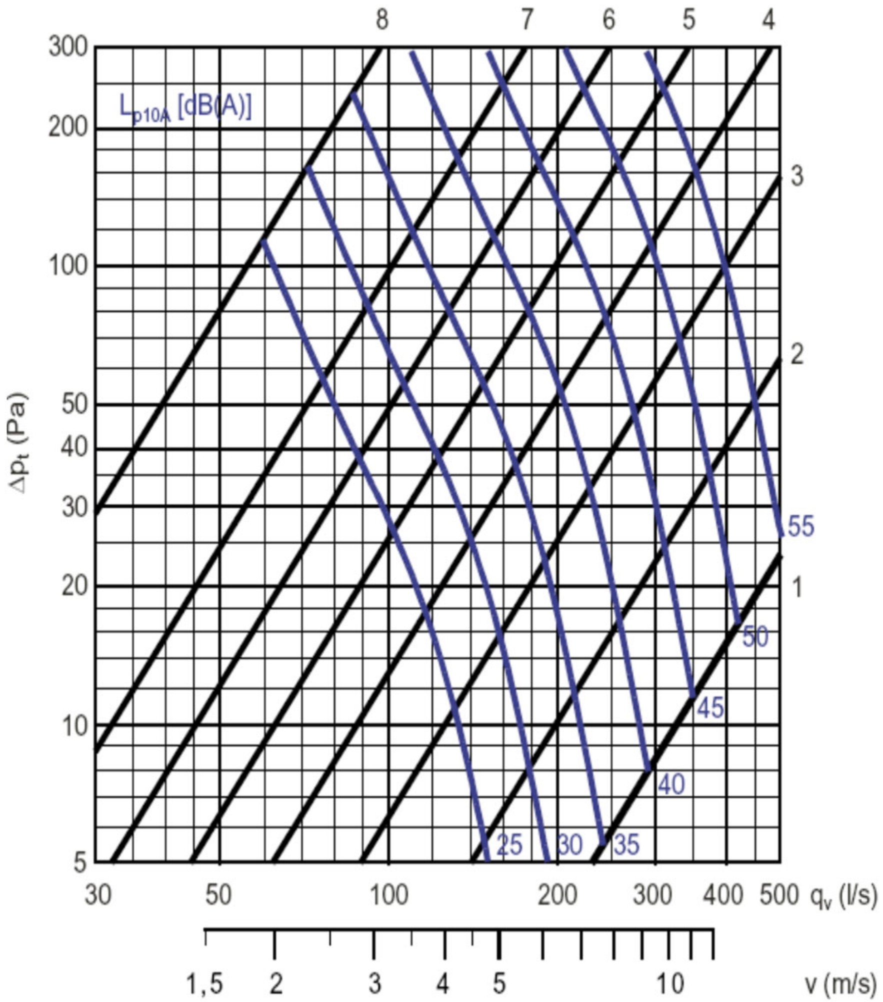
L_{p10A} is the A-weighted SPL in a room with absorption area A=10 m^2.

Dampers

- Used to adjust the airflow rate in the duct serving a group of rooms
- Manual or automated versions
- Automated versions are driven by, e.g., CO_2 signal

Noise source: damper

IRIS / IRIS-S 200



$$L_{Wokt} = L_{p10A} + K_{okt}$$

Äänen tehotaso L_W

IRIS	KORJAUS K_{okt} (dB)							
	Oktaavikaistan keskitaajuus (Hz)							
	63	125	250	500	1000	2000	4000	8000
80	10	16	12	9	5	-1	-6	-23
100	25	21	16	9	4	-6	-12	-25
125	17	17	13	7	1	-4	-6	-17
150	21	20	14	8	0	-6	-16	-29
160	19	18	14	6	-1	-6	-13	-25
200	20	17	12	5	-2	-5	-14	-26
250	16	12	8	3	1	-4	-17	-32
315	24	12	5	0	1	-2	-13	-27
400	15	9	6	2	-1	-4	-9	-13
500	14	7	4	1	-1	-4	-8	-11
630	15	7	3	2	-1	-5	-9	-11
800	9	5	3	3	-1	-6	-10	-13
Tol.±	6	3	2	2	2	2	2	3

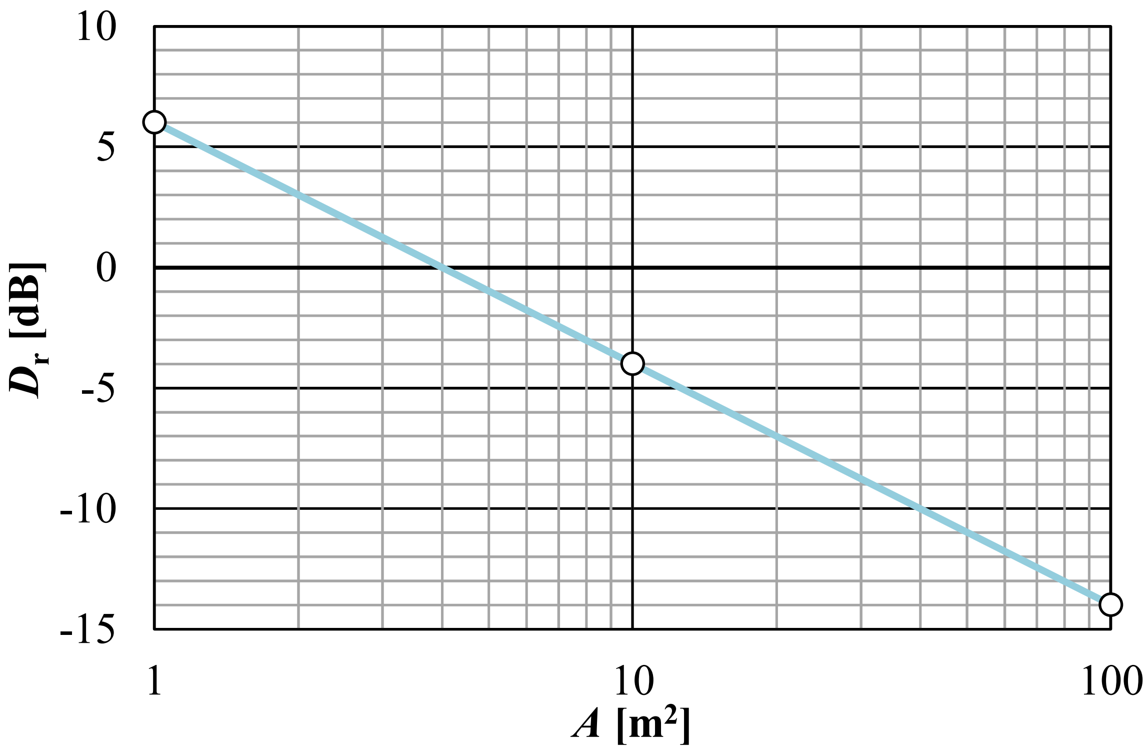
- Duct diameters from 80 to 800 mm are usually applied in ventilation.

Attenuations D

Room attenuation D_{r}: diffuse field

- Diffuse field is always assumed in ventilation noise calculations
- SPL can be calculated by the familiar equation
  - L_p [dB] is the SPL in room
  - L_W [dB] is the SWL of noise source
  - A [m^2] is the absorption area in the room
  - Dr = -4 dB usually
- It is usually assumed that A=10 m^2. Other values are applied in specially designed rooms such as in auditoria.

$$L_p = L_W + 10\log_{10}\left[\frac{4}{A}\right] \qquad D_r = 10\log_{10}\left(\frac{4}{A}\right)$$



7.2

Sound power level of ventilation terminal is LWA=35 dB. Calculate the LpA in an empty toilet and in a furnished living room for which the absorption areas are 1 and 20 m2, respectively.

$$L_p = L_W + 10\log_{10}\left[\frac{4}{A}\right]$$

Cut-off frequency in duct

- Attenuation in ducts and silencers depends strongly on frequency.
- At f_c (*cut-off frequency*), the largest cross-sectional dimension of the duct, d [m], equals with half wavelength:

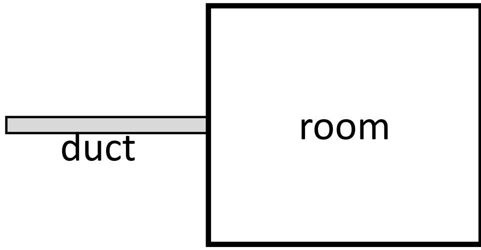
$$2d = \lambda = \frac{c_0}{f} \qquad f_c = \frac{c_0}{2d}$$

- Below f_c, sound field is one-dimensional (plane wave).
- Above f_c, sound field is 3-dimensional.

Circular ducts

d [mm]	f_c [Hz]	S [m^2]
100	1715	0.008
125	1372	0.012
160	1072	0.020
200	858	0.031
250	686	0.049
315	544	0.078
400	429	0.126
500	343	0.196
630	272	0.312
800	214	0.503
1000	172	0.785

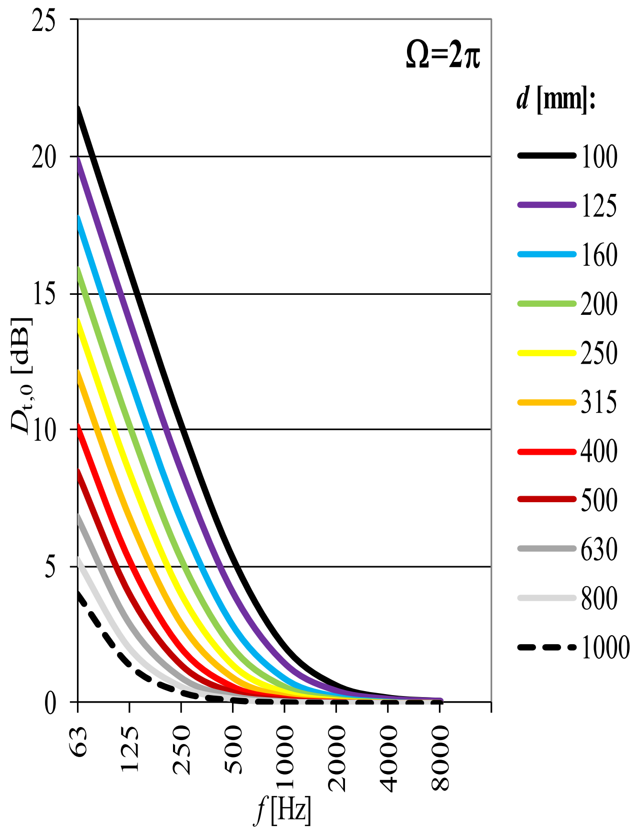
Attenuation of a terminating duct, D_t

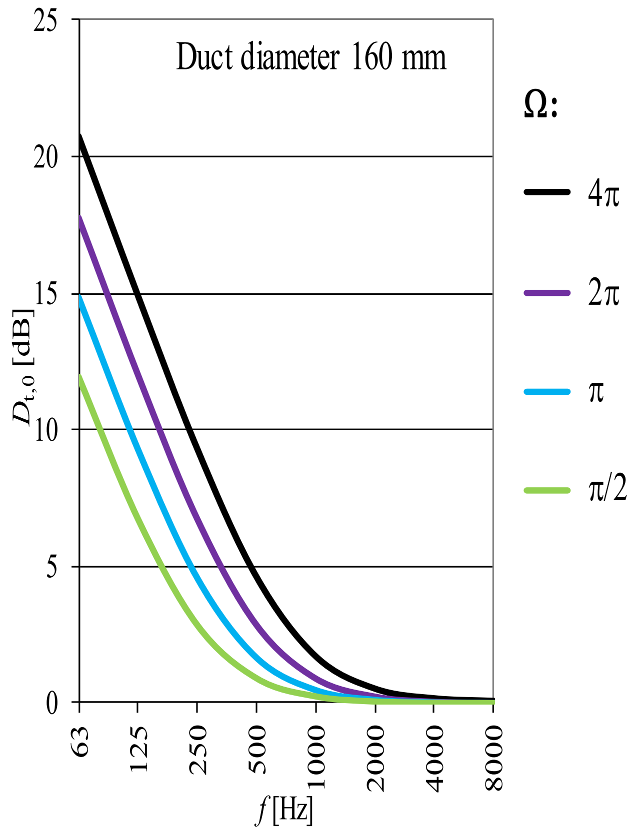


- A duct enters the room.
- Attenuation due to the area reduction of flow

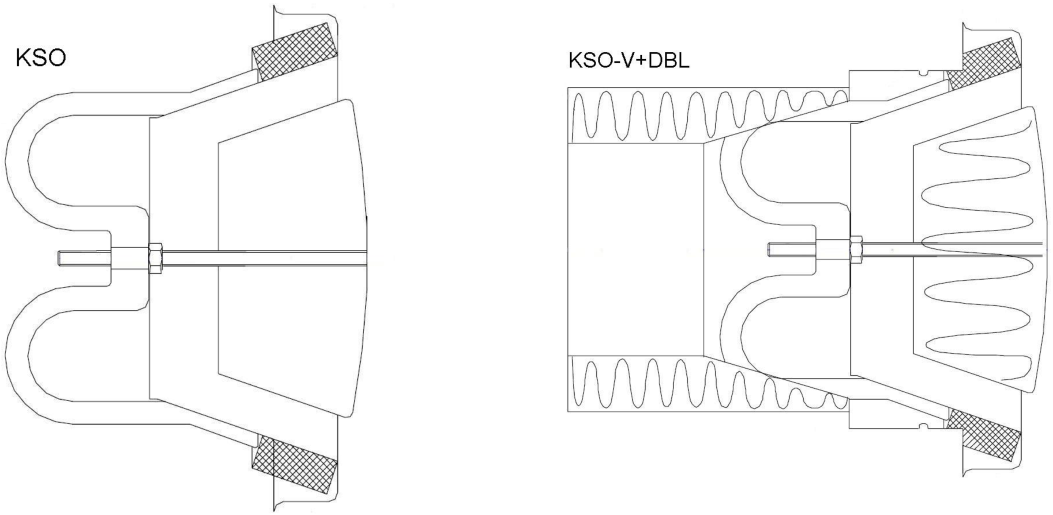
$$D_t = 10\lg\left(1+\left(\frac{c_0}{4\pi f}\right)^2 \frac{\Omega}{S}\right)$$

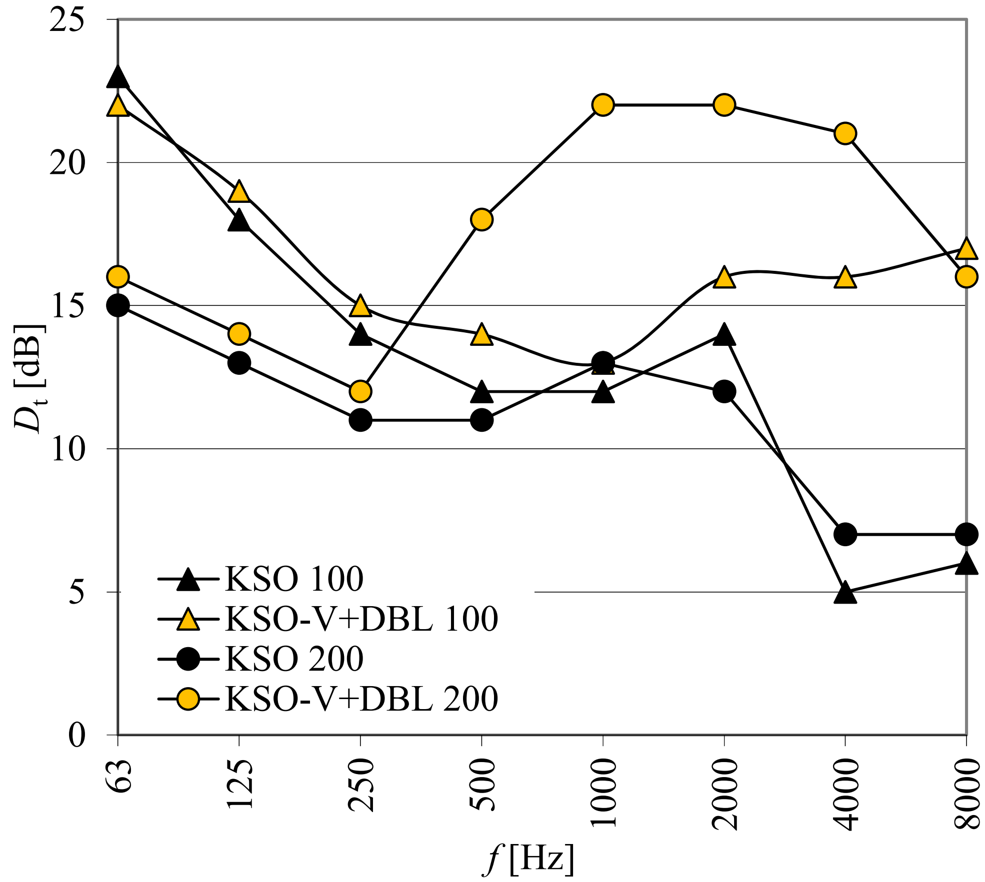
  - S [m²] is the cross-sectional area of duct
- *Below cut-off frequency*, the change of cross-sectional area means very large impedance: the hole on the wall acts as a monopole source and the diffraction increases with increasing wavelenght
  - Most of energy reflects back to the duct
- *Above cut-off frequency*, the sound field in the duct is already three dimensional and the change of cross-sectional area does not mean a large impedance
  - Most of the energy is transmitted to the room
- This is an inherent attenuation property of an open terminating duct ending to the room





Attenuation caused by a terminal unit, D_t

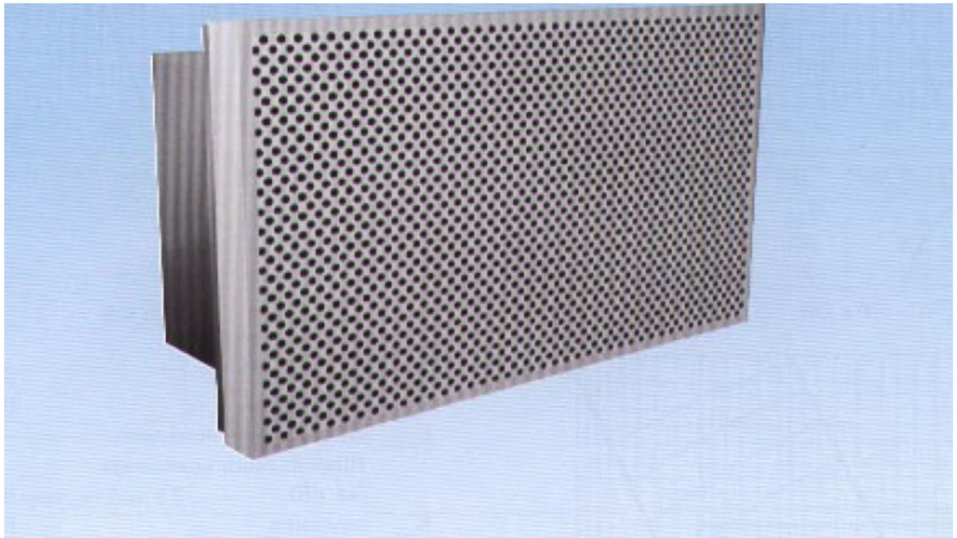


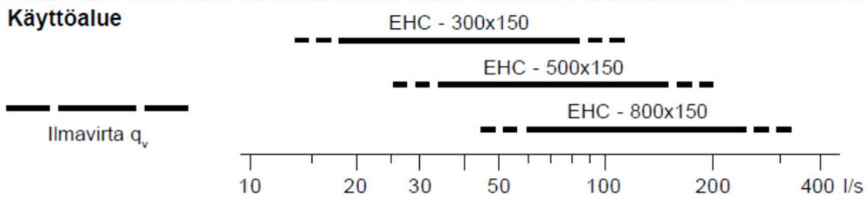


- D_t reported by manufacturers is the sum of
  - physical attenuation of an terminating duct (previous page)
  - attenuation due to the grille/obstacle/absorbers
- Terminal unit attenuation reduces the noise of both fan and damper but not the noise produced by the terminal unit itself.

Terminal unit attenuation example

- Terminal unit attenuates the noise produced by dampers and fans but not the flow noise produced by the terminal unit itself





Sound attenuation ΔL [dB]

EHC		ÄÄNENVAIMENNUS ΔL (dB) Oktaavikaistan keskitaajuus (Hz)					
		125	250	500	1000	2000	4000
300 x 150	O	10	7	7	7	3	5
	C	13	11	15	15	10	10
500 x 150	O	8	6	7	7	3	4
	C	12	10	14	14	9	9
800 x 150	O	6	4	6	5	3	4
	C	10	9	14	14	8	9
Tol.±		3	2	2	2	2	2

EHC:n keskimääräinen äänenvaimennus ΔL kanavasta huoneeseen sisältäen liittyvän kanavan päätevaimennuksen seinäasennuksessa saadaan yllä olevasta taulukosta.

ΔL [dB] contains the attenuation of the open terminating duct

Attenuation in duct branches D_d

- Duct branch attenuation is important for the fan or damper noise that arrives to the room
- If the duct has a branch, the total air flow rate maintains after the branch.
- The first approximation is that noise distributes to the branches in the same relationship as air flow rate.
- The attenuation due to duct branches from the fan to a room is approximated by:

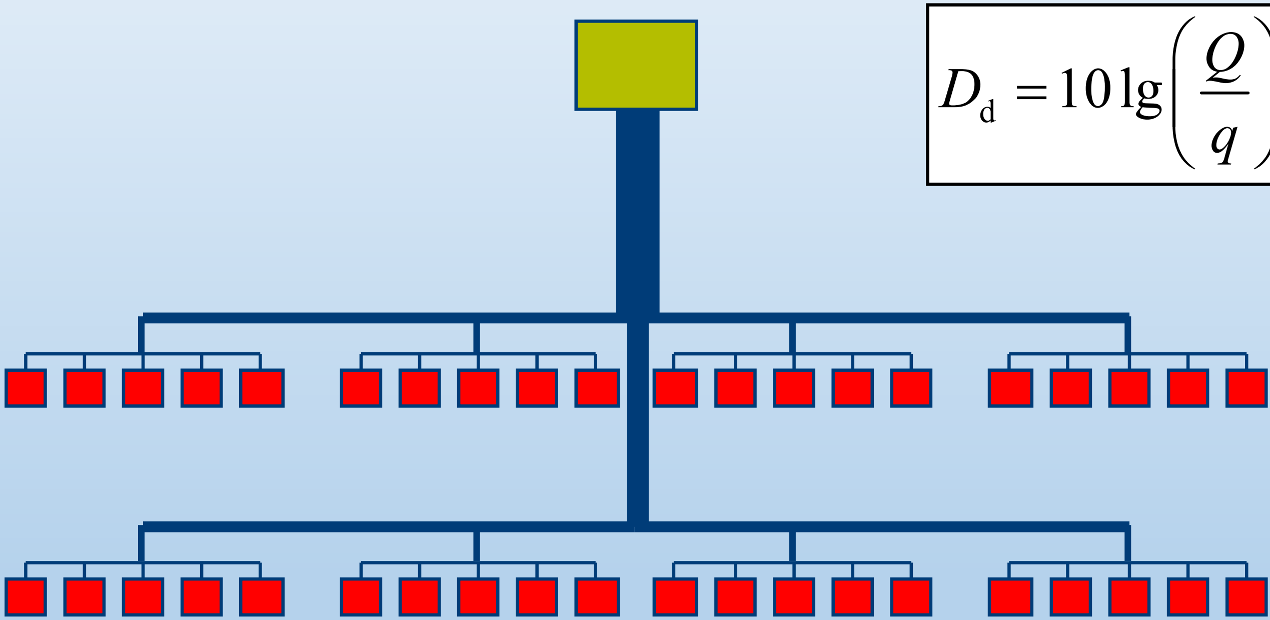
$$D_{\mathrm{d}} = 10\lg\left(\frac{Q}{q}\right)$$

  - q [m³/s] is the air flow rate to the room under inspection
  - Q [m³/s] is the air flow rate of the fan
- The value is independent on frequency.

7.3

Fan total outlet air flow rate is 2 m3/s from 40 rooms of a building.
What is the mean attenuation of branches, $\mathbf{D_q}$, from the fan to the terminal unit?

$$D_d = 10\lg\left(\frac{Q}{q}\right)$$



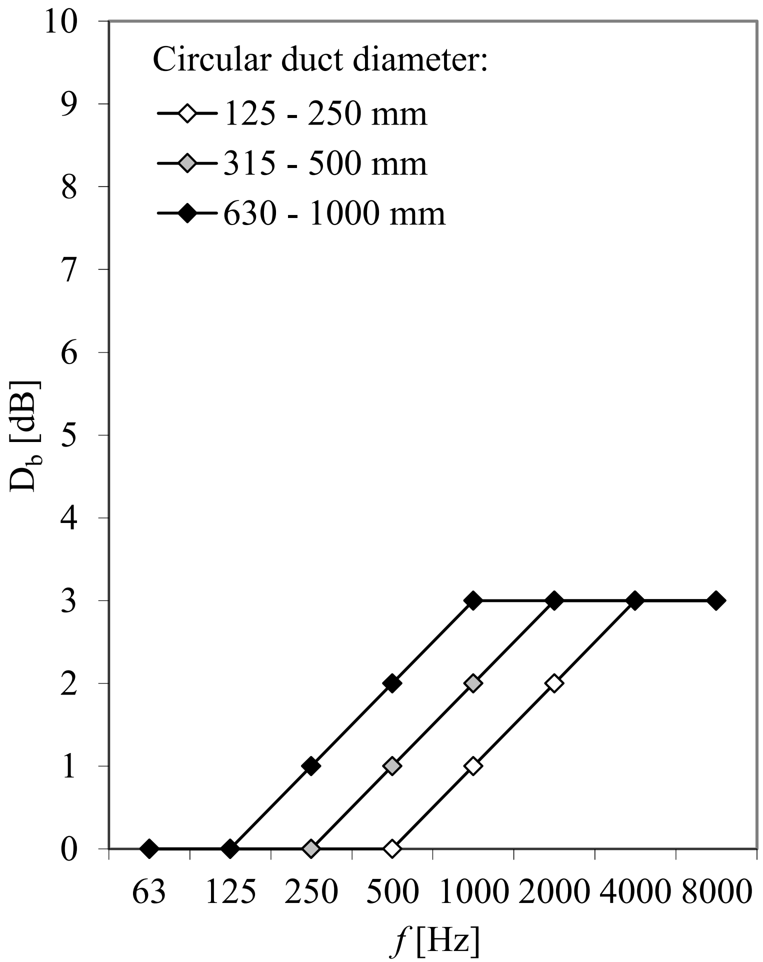
Attenuation of duct walls, D_w

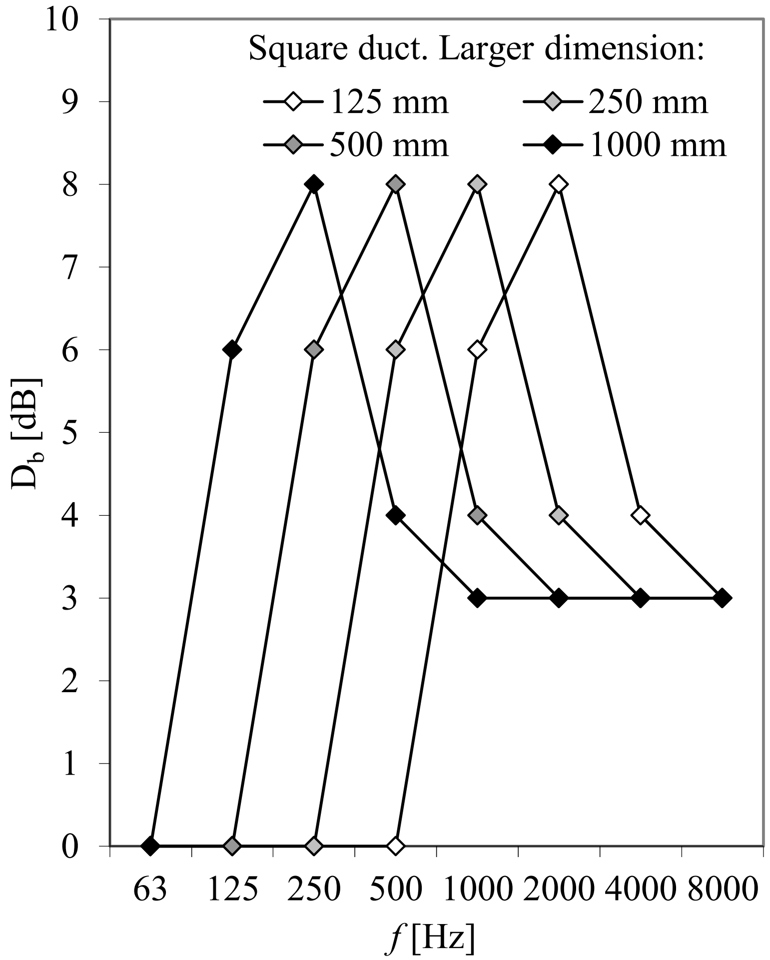
- Attenuation is caused both by sound absorption and sound transmission through the duct walls
  - Sound reduction index of R=10 dB means sound absorption of α=0.10
- Circular ducts: < 0.1 dB/m
- Square ducts: 0.2−0.6 dB/m
  - low frequency absorption is higher than high frequency absorption because of the resonances of flexible walls (circular ducts are very rigid)
- Attenuation of the duct is usually not considered in calculations. This provides some reserve for the calculations.

VDI 2081:1983

Attenuation in 90 degree duct bends, D_b

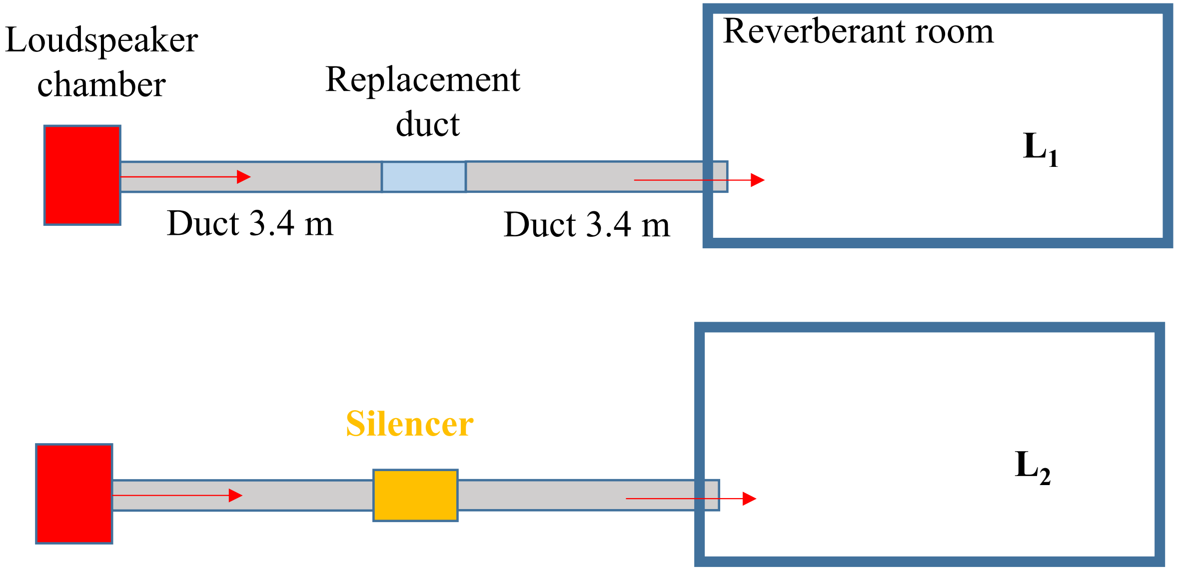
- Due to diffraction, small frequencies can easily transmit behind the bend.
- Instead, high frequencies do not bend easily, hit the duct wall and reflect back.
- Attenuation of the bends is usually not considered in calculations because the attenuation is usually sufficient at high frequencies where the bends provide some attenuation. This provides some reserve for the calculations at high frequencies.





Attenuation due to silencer, D_s

- Silencers are built so that the perimeter is covered with porous sound-absorbing material, such as wool.
- Wool is usually covered by perforated steel to avoid the desorption of wool fibres and to provide mechanical protection during sweeping
- Attenuation values are given by the manufacturer.
- $D_s = L_1 - L_2$ where L_1 and L_2 [dB] are the SPLs caused by a loudspeaker to the test room without the silencer (silencer replaced by straight duct) and with silencer, respectively.



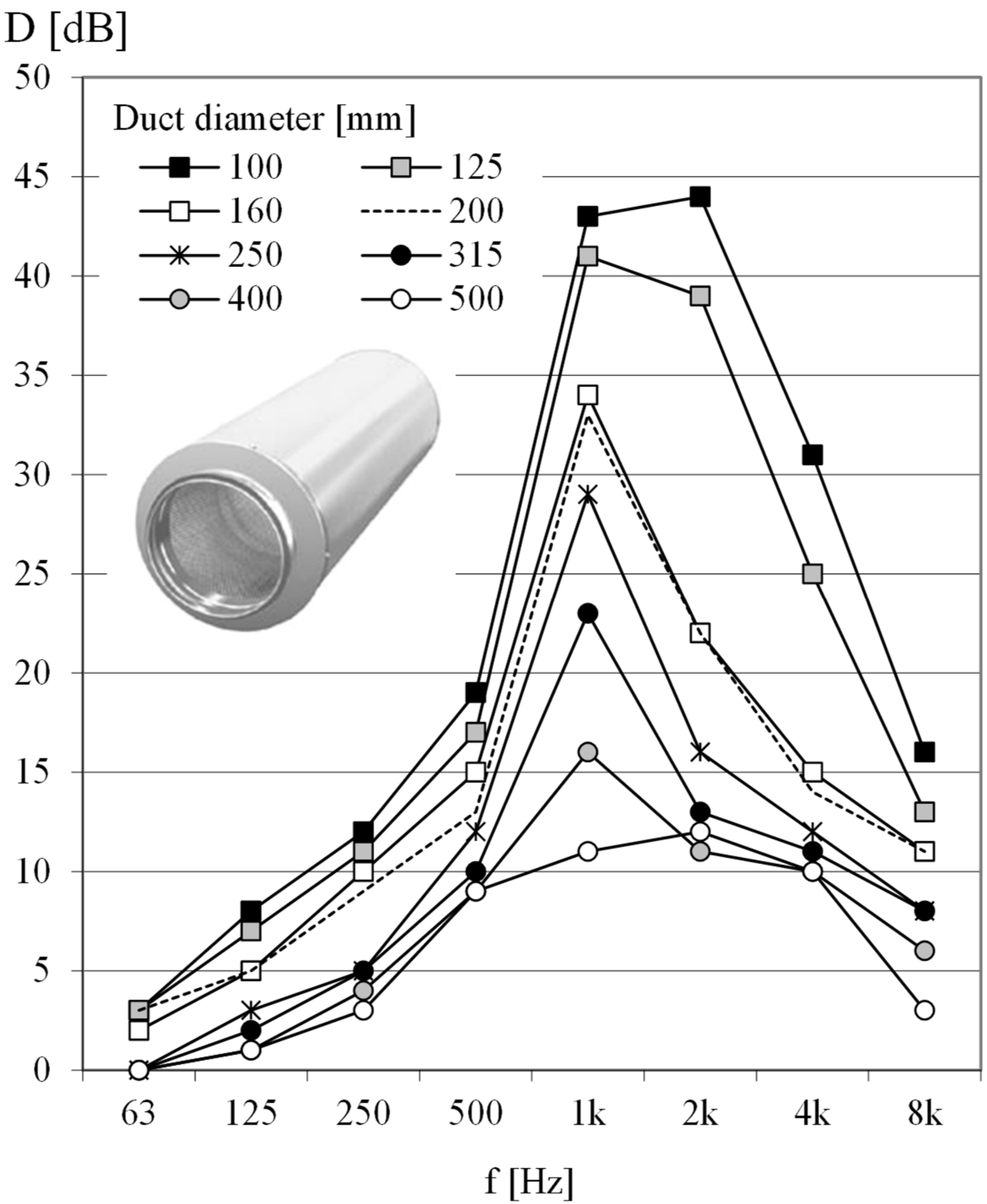
Sound attenuation ΔL [dB]
Length 600 mm

KLT	ÄÄNENVAIMENNUS ΔL (dB) Oktaavikaistan keskitaajuus (Hz)							
	63	125	250	500	1000	2000	4000	8000
100	9	17	19	36	47	44	40	31
125	11	15	16	35	44	45	42	32
160	15	13	12	32	40	43	37	26
200	15	6	11	23	30	39	29	22
250	12	4	8	17	24	28	15	16
315	5	3	6	15	20	18	14	14
400	5	2	5	15	18	16	15	19
Toler.±	6	3	2	2	2	2	2	3

- Jatkuu 20.11.

Direct silencer – effect of duct diameter

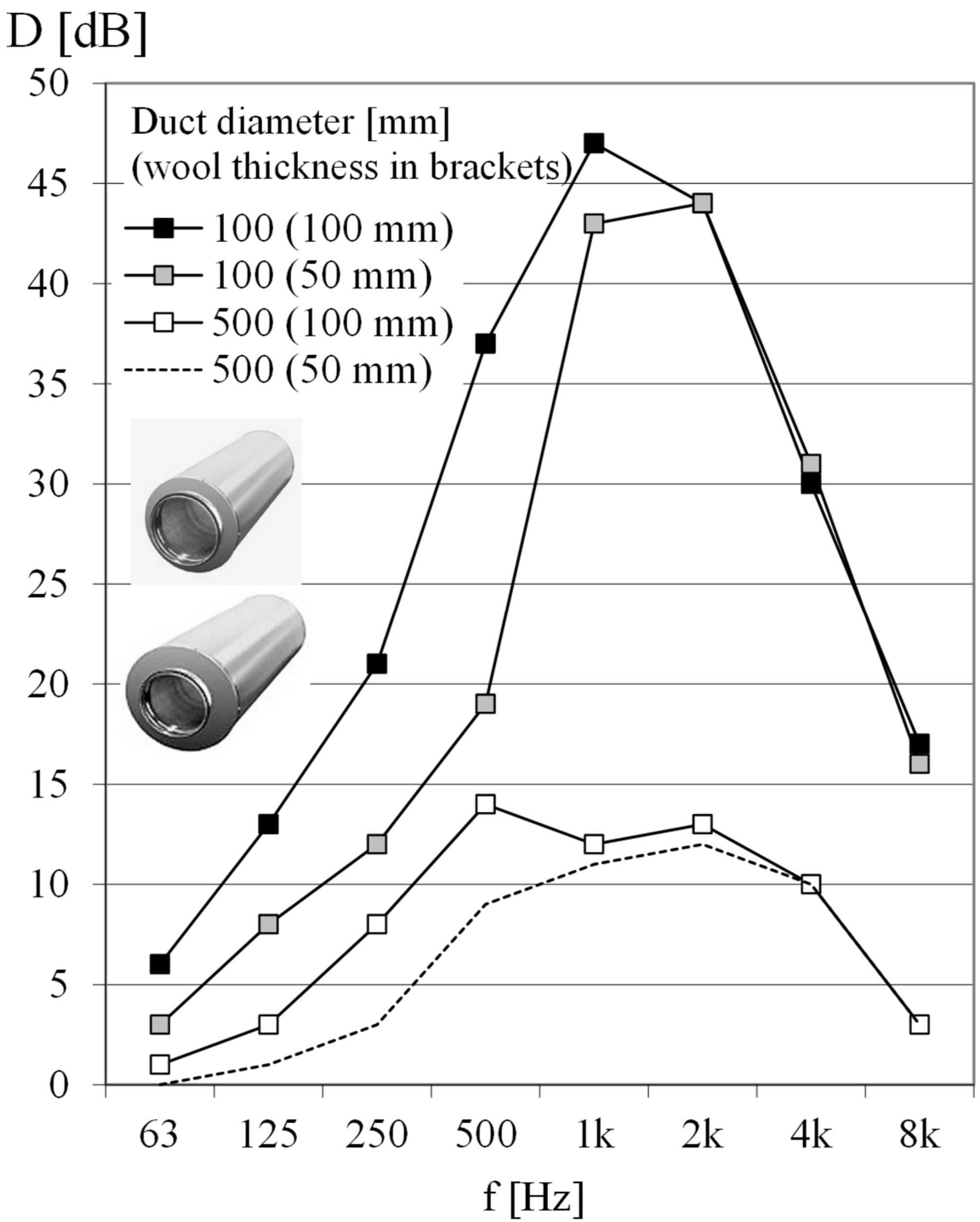
- Wool thickness 50 mm
- Silencer length L=600 mm.
- Attenuation reduces with increasing diameter.



Data: IVK-Tuote Oy

Direct silencer – effect of wool thickness

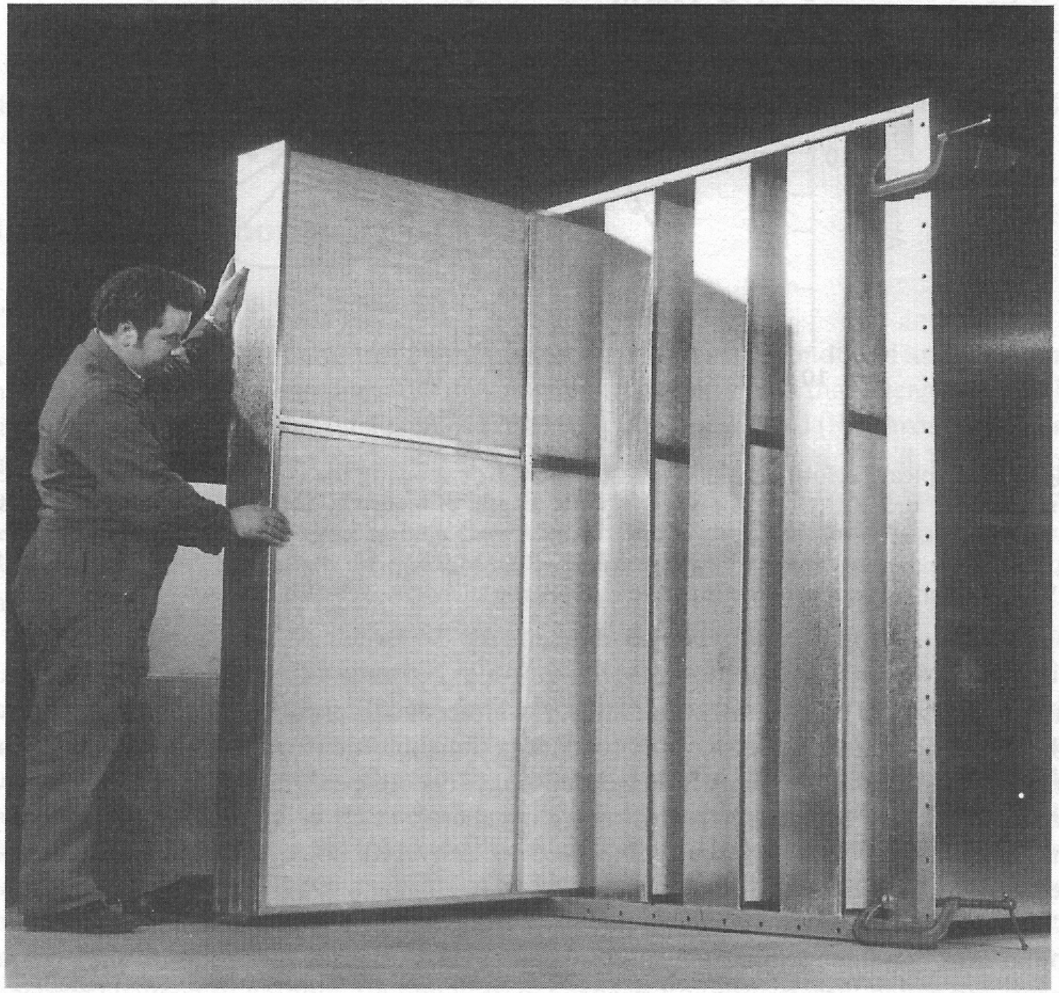
- Silencer length *L*=600 mm.
- Thicker wool improves attenuation at low frequencies both with small and large silencer diameters



Data: IVK-Tuote Oy

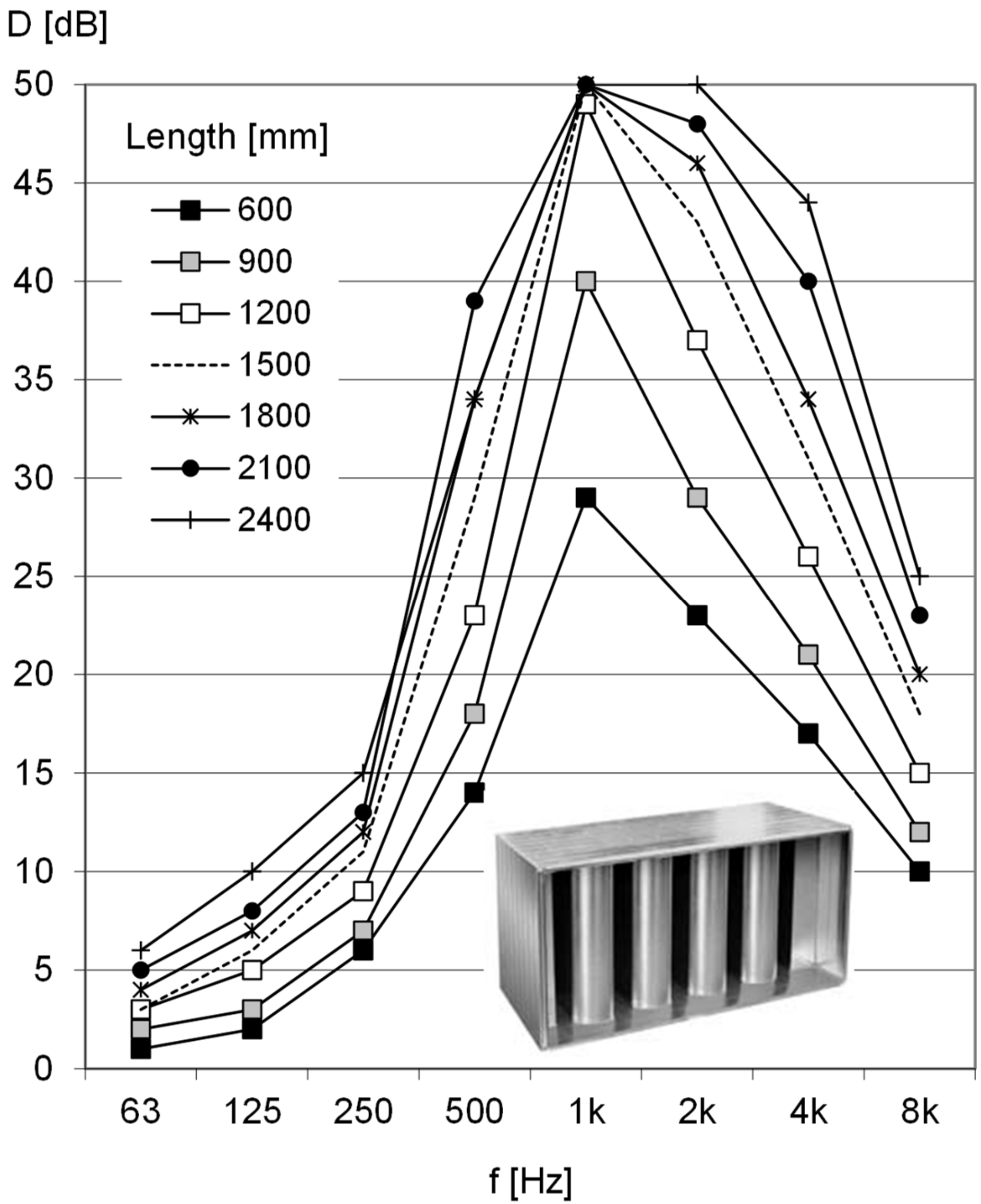
Primary silencers

- Primary silencer is located right after the fan
  - Lamella construction is usual
- Secondary silencers are located close to the rooms
  - Previous slides deal with secondary silencers
- D_s of a large lamella silencer exceeds 50 dB in most octave bands
- The flanking transmission via silencer body limits the performance to 50−60 dB.



Direct silencer – effect of length

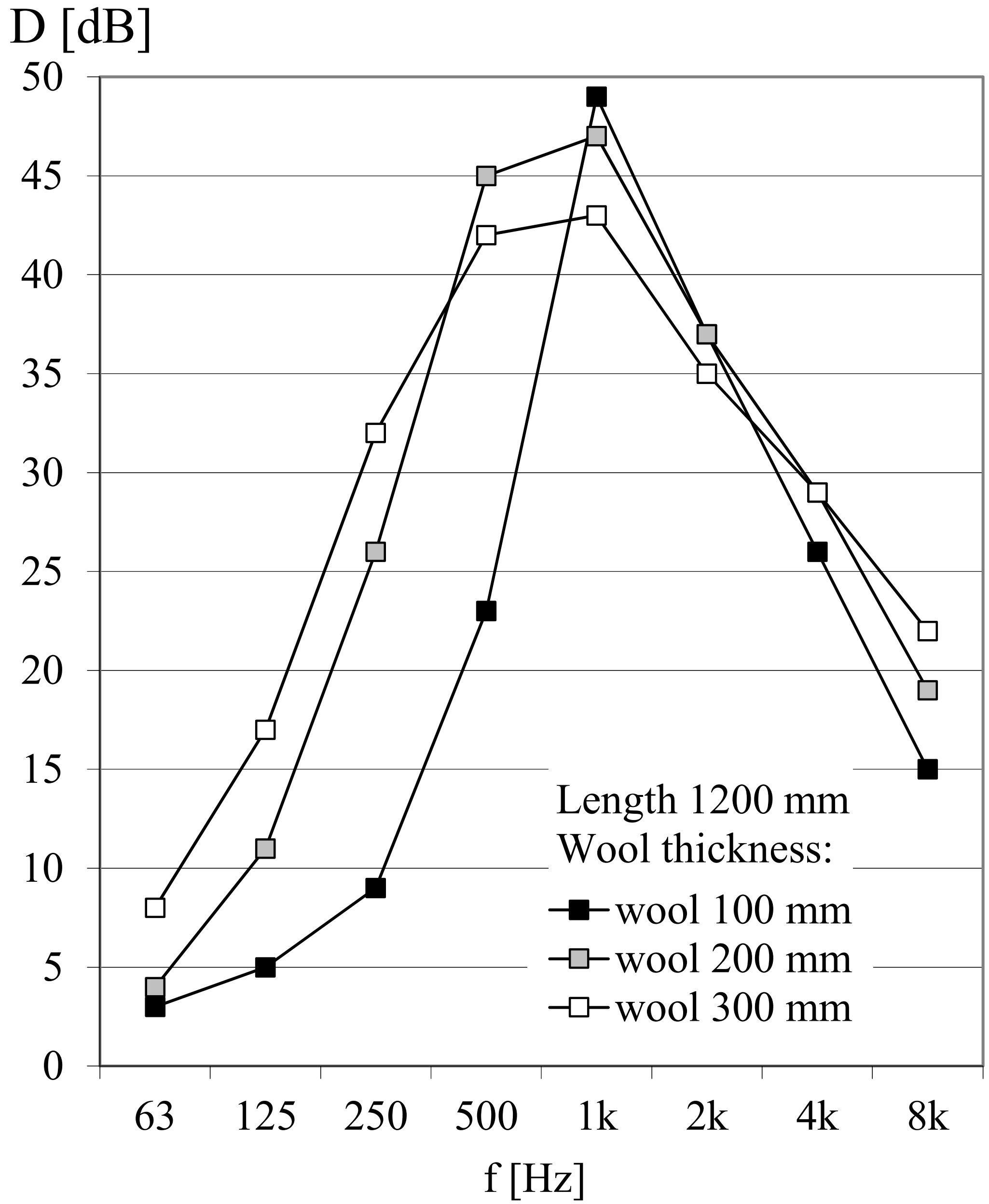
- Lamella silencer
- Width of the free path is 100 mm.
- Quadruple length means double attenuation



Data: IVK-Tuote Oy

Direct silencer – effect of wool thickness

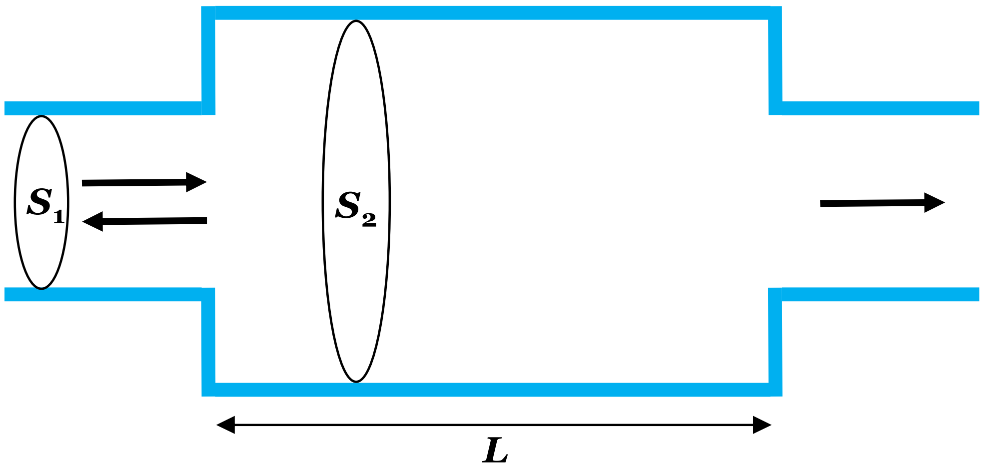
- Lamella silencer
- Width of the free path is 100 mm.
- Effect of wool is evident at low frequencies

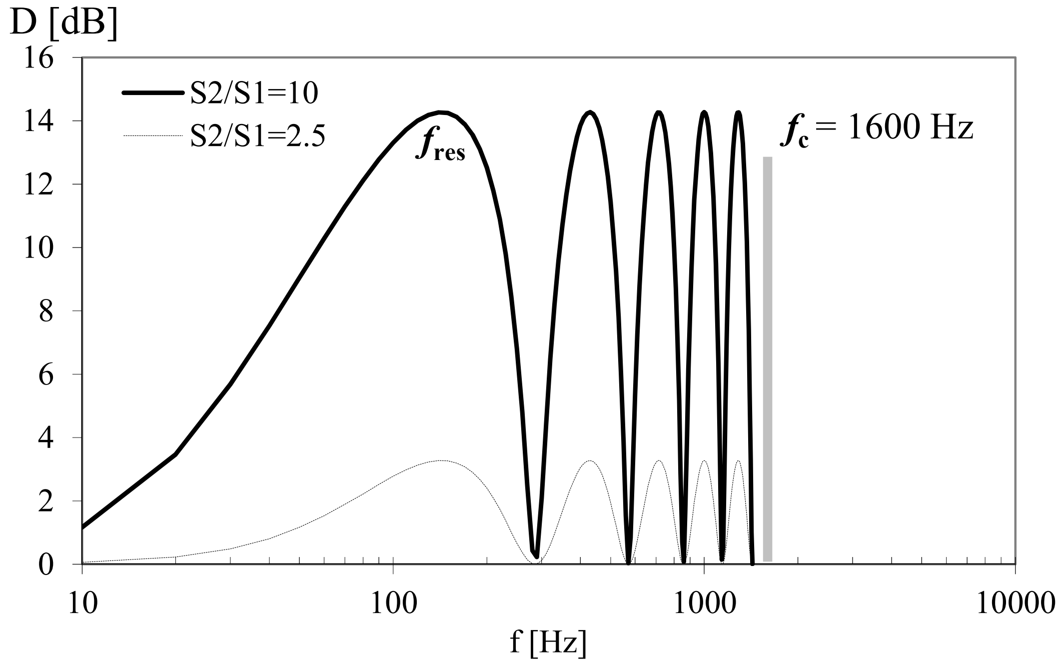


Data: IVK-Tuote Oy

Reactive silencer: chamber

- Reactive silencer means that
- Resonator behavior when $f < f_c$
- Figure:
  - duct diameter 125 mm
  - chamber diameter 200/400 mm
  - chamber length L=600 mm





$$f_{res} = \frac{nc_0}{4L}; \; n = 1,3,5,...$$

$$D = 10\lg\left[1 + \left(\frac{S_1}{2S_2} - \frac{S_2}{2S_1}\right)^2 \sin^2 kL\right]$$

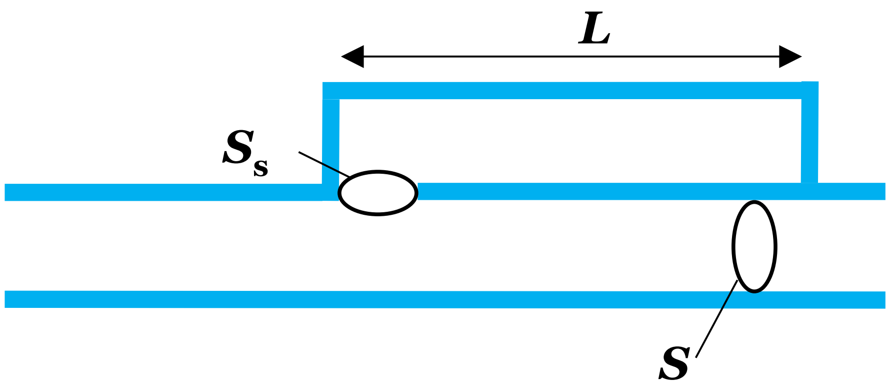
- Assume that $f < f_c$. The attenuation caused by a side branch is:

$$D = 10\lg\left|1 + \frac{S_s \rho_0 c_0}{2S\mathbf{Z}_s}\right|^2$$

Quarter wave resonator

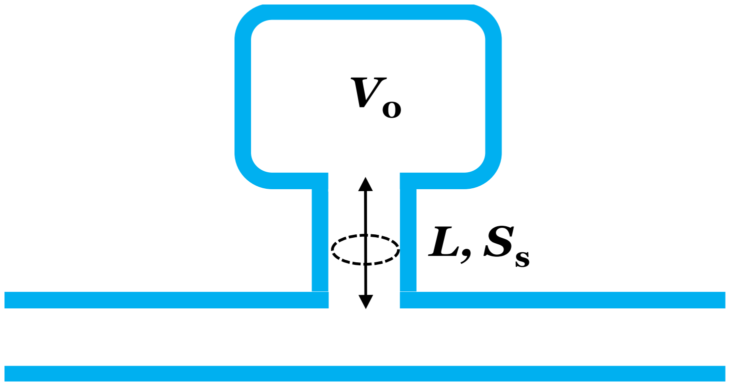
$$f_1 = \frac{4c_0}{L}$$

$$\mathbf{Z}_s = -i\rho_0 c_0 \cot kL$$



Helmholz resonator

$$f_{res} = \frac{c_0}{2\pi}\sqrt{\frac{S_s}{LV_0}} \qquad \mathbf{Z}_s = i\omega\rho_0 L + \frac{\rho_0 c_0^2 S_s}{i\omega V_0}$$



Airborne sound insulation of machine room

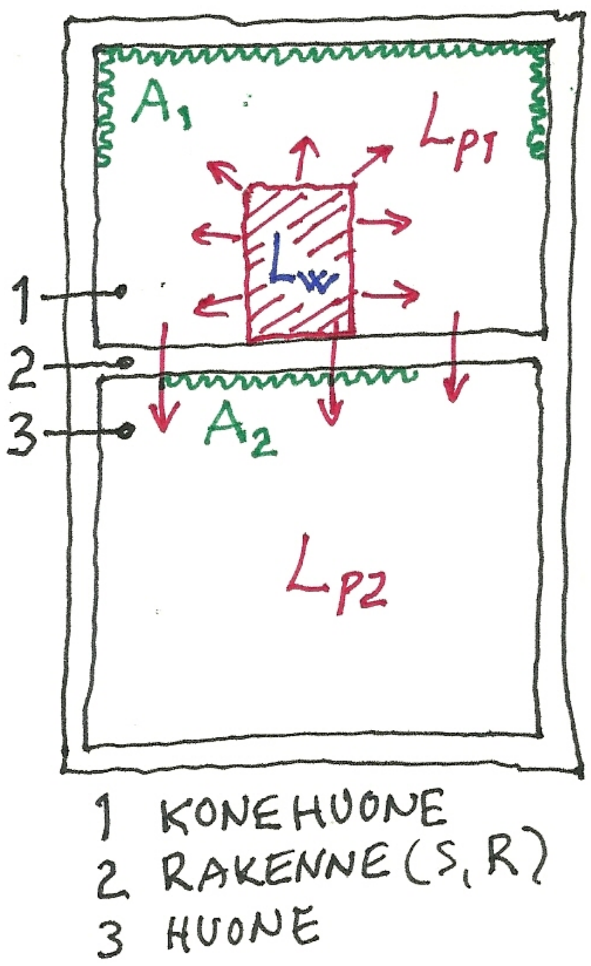
- SPL in machine room, L_{p1}:

$$L_{p1} = L_W + 10\log_{10}\left(\frac{4}{A_1}\right)$$

  - L_W [dB] is sound power of the fan to the environment (see the fan graph and related K_{oct} terms))
  - A_1 [m^2] is absorption area of machine room
- SPL in nearby room L_{p2}:

$$L_{p2} = L_{p1} - R + 10\lg\frac{S}{A_2}$$

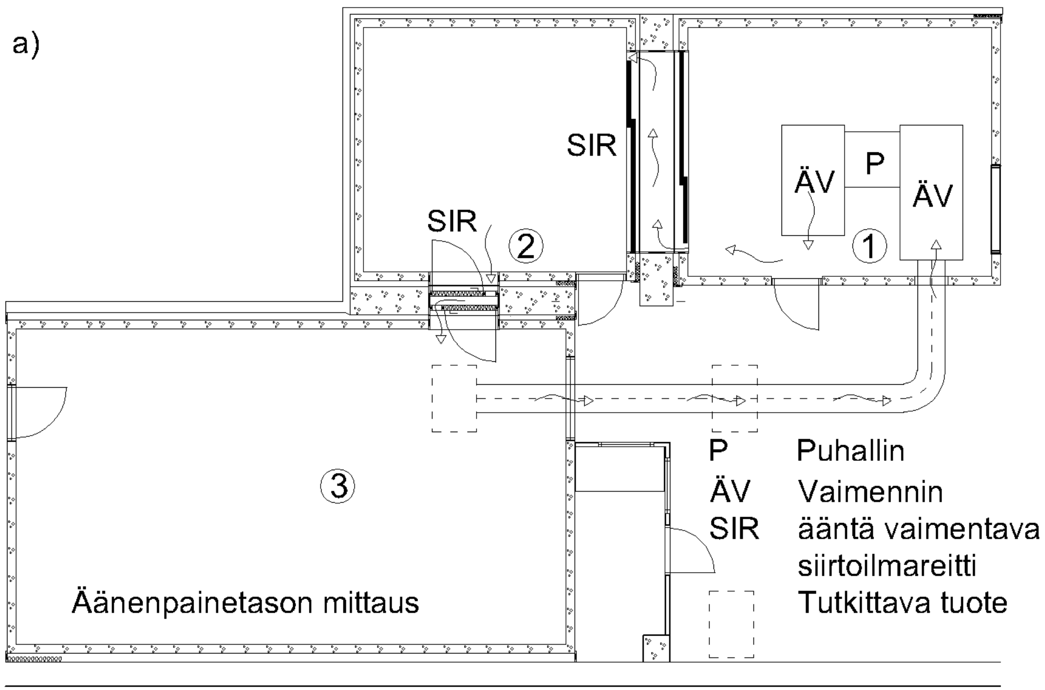
  - R [dB] is the SRI of the construction between the two rooms
  - S [m^2] is the area between the rooms
  - A_2 [m^2] is absorption area of room 2

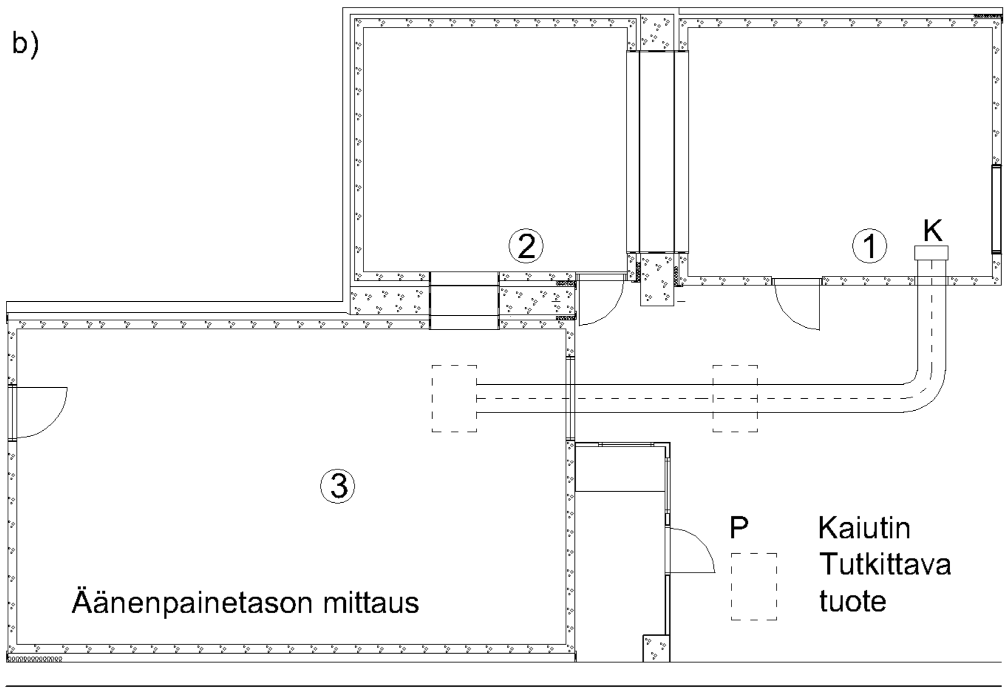


Laboratory measurements of duct components

Sound power level
- Terminal devices
- Fans
- Dampers
- 22 dB L_{WA}

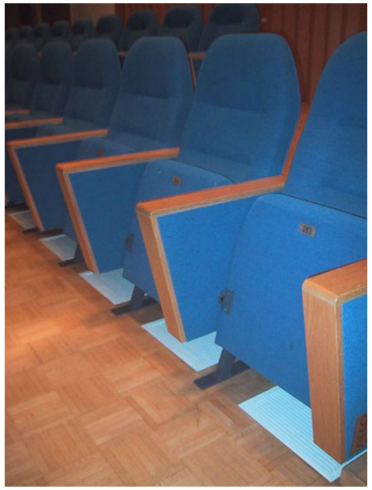
Insertion loss
- Silencers
- Terminal devices





Risto Ryti –hall

- Huittinen
- 404 seats
- A low-velocity terminal device under each seat
- Speech amplification
- BG noise 30 dB L_{Aeq}
- Ten foldable elements on side walls provide adjustable RT



open

closed

Risto Ryti, 1889-1956
President of Republic
1940-1944