Rak-43.3415 Building Physics Design 2
ACOUSTICAL DESIGN
Autumn 2016

LECTURE 5
HVAC noise control, vibration isolation

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HELIMAKI ACOUSTICS

HVAC noise control

”When we consider all the noisy machinery that modern building techonology has created and, first of all, the inappropriate building methods which have been used out of lack of knowledge and experience, it is no wonder that the general problem of sound insulation still lies ahead of us without final solution.”

M. Sc. U. Varjo 1938

Effects of HVAC technology (LVIS) on acoustics

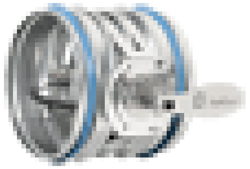
- Flow noise
  - Air conditioning sytems
  - Water and sewer systems
  - Heating systems
- Structural borne noise
- Sound insulation
  - Flanking transmission via ventilation shafts
  - Sealing of penetrations (läpiviennit)
  - Coupling of double structures by HVAC installations

Air conditioning

Air conditioning
Sound sources and effects on sound insulation

- Sound sources
  - Fan noise (puhallinääni)
  - Flow noise (ilmavirtauksen aiheuttama ääni)
  - Structure borne noise
  - Noise output of equipment to their surroundings (äänenkehitys ympäristöön)
- Effects on airborne sound insulation
  - Flanking transmission via ventilation shafts
  - Sealing details

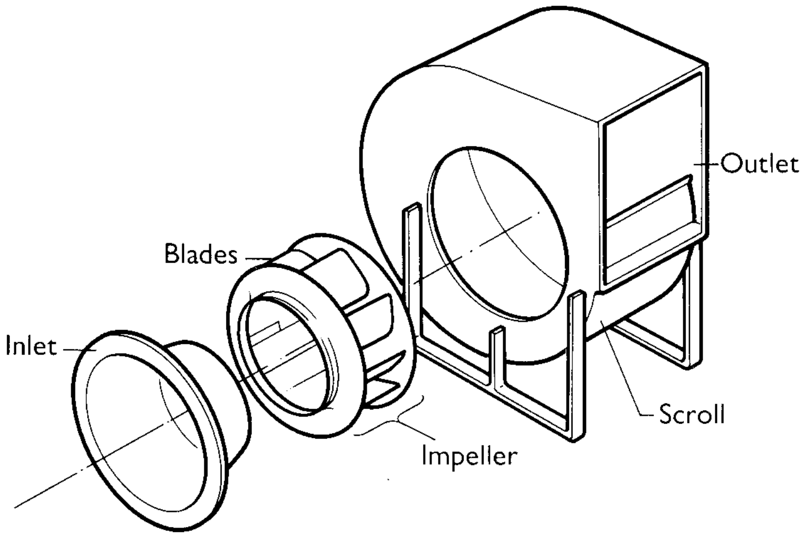
Sound sources

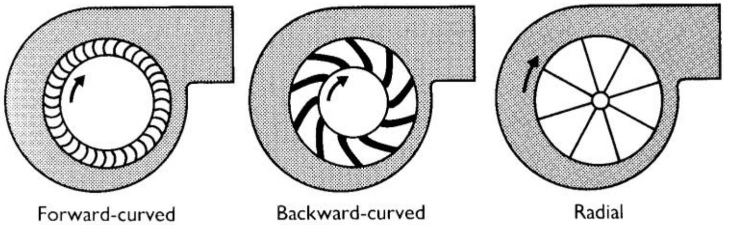


Sound sources
Fans

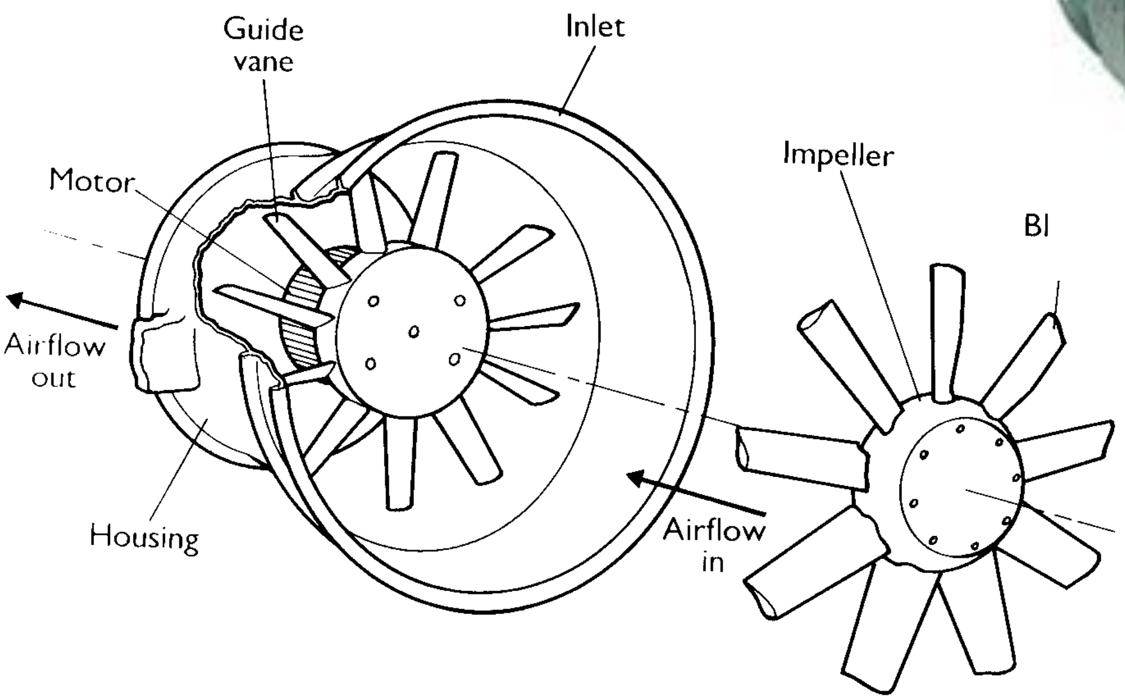
- Factors affecting sound level:
  - Air flow rate, ilmamäärä (m3/s)
  - Pressure levels (Pa)
  - Fan type and size
- Fan types
  - Axial fans (aksiaalipuhaltimet)
  - Mixed flow fans (sekavirtauspuhaltimet)
  - Enclosed fans (koteloidut puhaltimet)

Sound sources
Enclosed fan





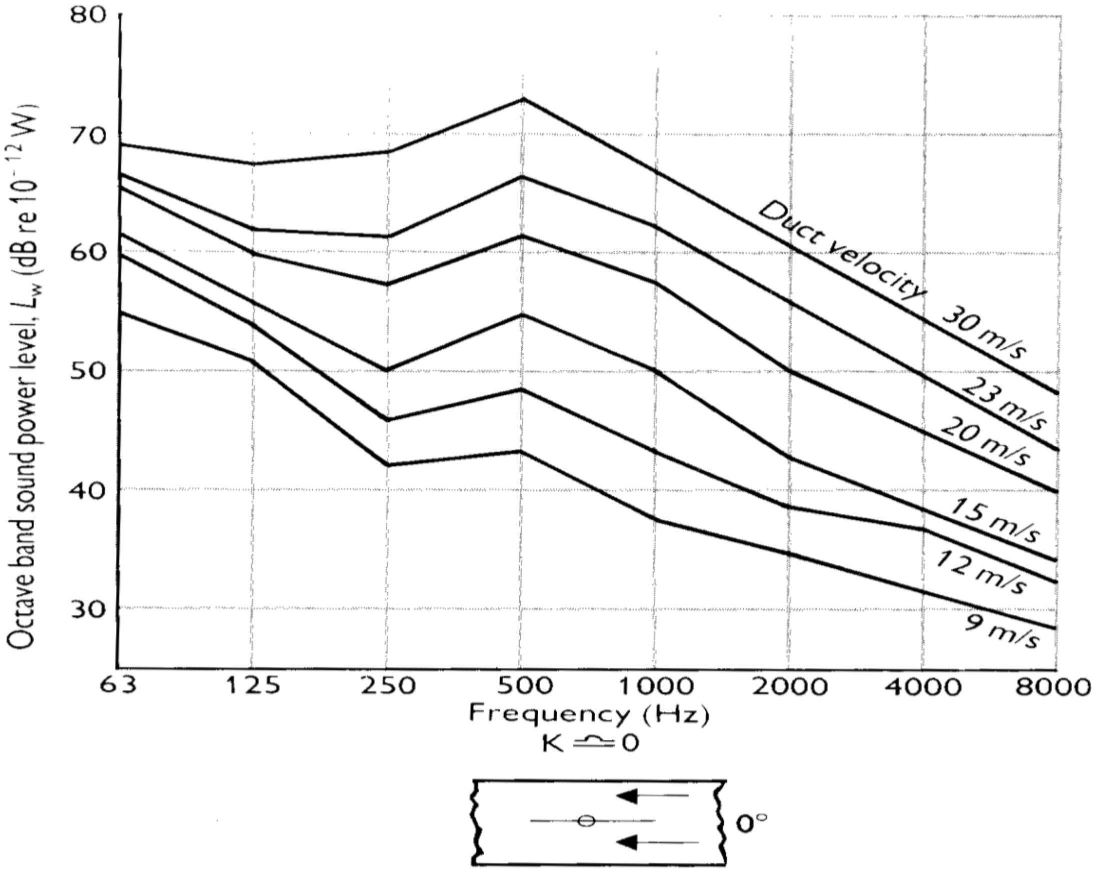
Sound sources
Axial fan



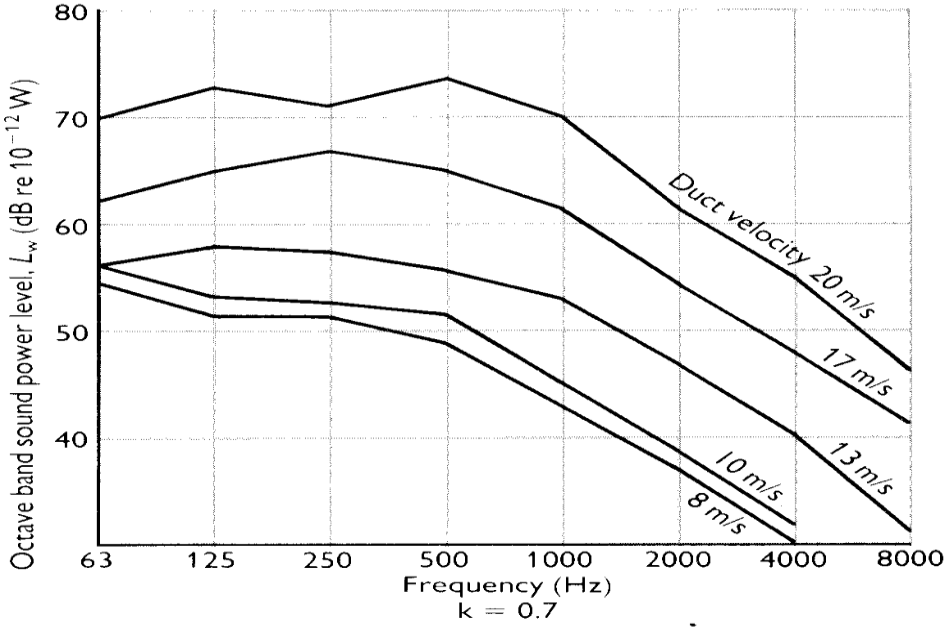
Sound sources
Control devices (säätölaitteet)

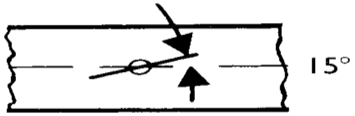
- Factors affecting sound level:
  - Air flow rate, ilmamäärä (l/s)
  - Pressure levels (Pa)
  - Type and size
- Control device types
  - Dampers (säätöpellit)
    - Iris type dampers
    - Pivot dampers (läppäpelti)
  - Air flow regulators (ilmamääräsäätimet, IMS)

Sound sources
Pivot damper 600 x 600 mm

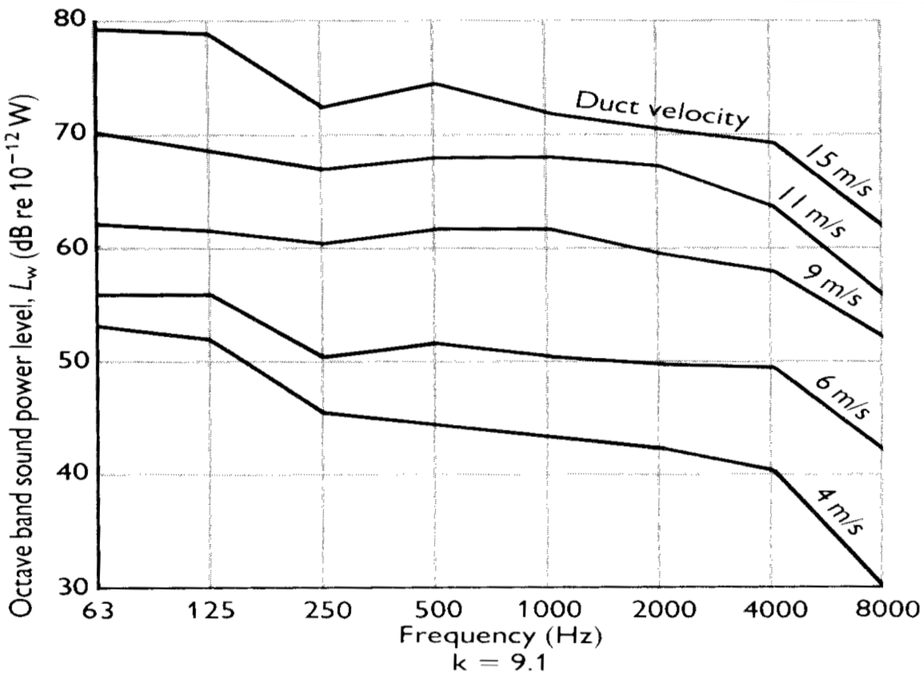


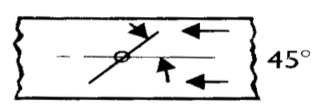
Sound sources
Pivot damper 600 x 600 mm





Sound sources
Pivot damper 600 x 600 mm





Sound sources
Air terminal units (pääte-elimet)

- Factors affecting sound level:
  - Air flow rate, ilmamäärä (l/s)
  - Pressure levels (Pa)
  - Type and size
- Types of air terminal units
  - Must always be checked depending on equipment / system
  - The effect of pressure level also has to be considered

Regulations

- Finnish Building Code section C1-1998:
  - Dewlling room: $L_{A,eq} \leq 28$ dB ja $L_{A,max} \leq 33$ dB
  - Kitchen: $L_{A,eq} \leq 33$ dB ja $L_{A,max} \leq 38$ dB
  - Noise caused by HVAC equipment outside the building in a noise sensitive area: $L_{A,eq} \leq 45$ dB
- Additional guideline values on, e.g.
  - Finnish Building Code section D2-2012
  - Asumisterveysohje (2003)

Calculating the sound level in a room

1. Find out the sound power level of the source, L_w
2. Subtract the attenuation values between source and room
3. Subtract room attenuation, D_{room} / D_{huone}
4. Do A-weighting
5. Calculate equivalent sound level $L_{A,eq}$ in the space
6. Sum up the sound levels of all sources affecting the total sound level

Note:

- Air is not the dimensioning factor of the air conditioning system, but rather sound (level)
- Sound level in a space depends on all HVAC sound sources

Sound power level

- Sound power levels L_W must be known in octave bands (63-4000 Hz)
- Stating sound power levels
  - L_W = non-weighted sound power level
  - $L_{W,A}$ = A-weighted level
  - $L_{W,A,10m2}$ = A-weighted level, including 4 dB room attenuation
- Before doing the calculation you have to know how the sound power level has been stated so that
  - A-weighting is not done twice
  - Calculation os based on true room attenuation
- A good policy is to convert the stated values of sound power level L_W to non-weighted (linear) values

Data stated by manufacturer
Case 1

- Case 1: sound level given directly as sound power level
  - Linear
  - A-weighted
  - Sometimes reference level can change: 10^{-12} W, 10^{-13} W
- Note: It is not uncommon that the sound levels are stated unclearly, it might not be clear whether octave band levels are linear or A-weighted values

Data stated by manufacturer
Case 2

- Case 2: sound level given in a space with room attenuation X dB
  - 4 dB (10 m2-Sab) is the normal room attenuation corresponding to, e.g., dwelling or office room
  - The room attenuation on which the reported level are besed can, however, vary considerably
  - The sound levels can be reported with up to 17 dB room attenuation
- Due to this variation the sound power level of each sound source needs to be checked separately:

  $L_w = L_p$ + room attenuation

Data stated by manufacturer
Case 3

- Case 3: sound level given as a measurement result in some space
  - Volume
  - Reverberation time
  - Measured sound pressure level Lp
- In this case absorption area has to be calculated:

$$T = \frac{0{,}16 \cdot V}{A} \qquad A = \frac{0{,}16 \cdot V}{T}$$

- …and the sound power level:

$$L_w = L_p + 10\lg\left(\frac{A}{4}\right)$$

Data stated by manufacturer
Case 4

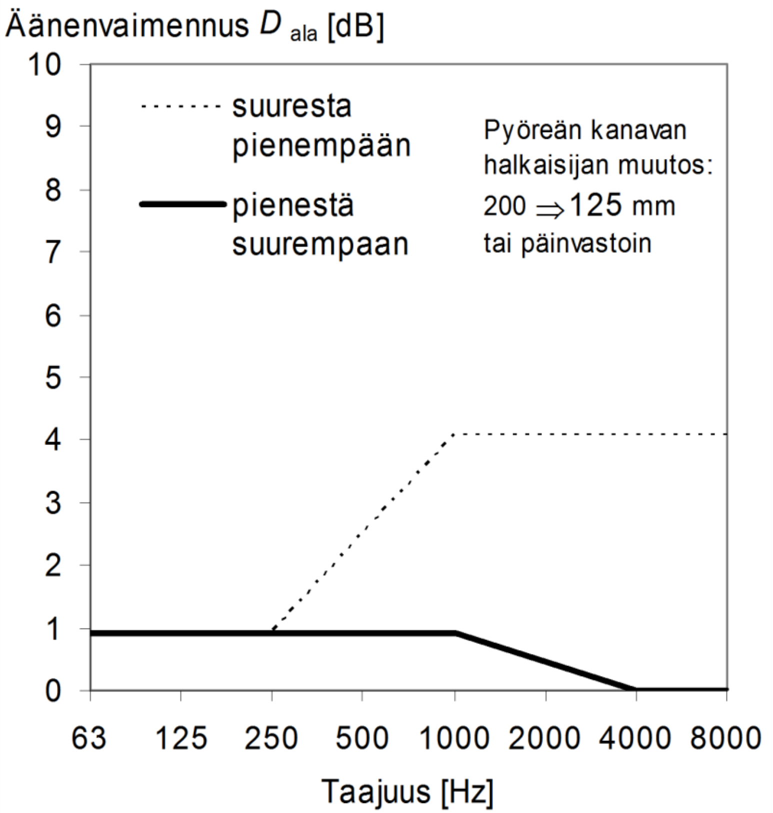
- Case 4: sound level given as a measurement result outside or in an anechoic chamber
  - Distance r
  - Sound pressure level Lp
- In this case the sound power level is calculated from the equation

$$L_p = L_w + 10\lg\left(\frac{1}{4\pi r^2}\right)$$

Attenuation in a ventilation shaft

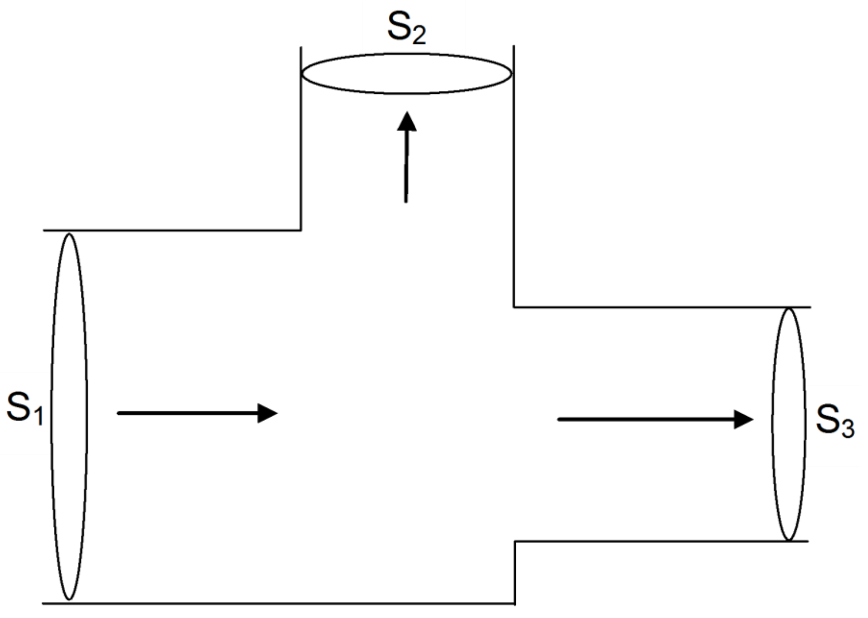
- Attenuation is caused by
  - Change in cross-sectional area of the shaft (poikkipinta-alan muutos), D_{ala}
  - Branching of the shaft (kanavan haaroittuminen), D_{haara}
  - Bends (mutkat), D_{mutka}
  - Silencers (äänenvaimennin), $D_{\text{äv}}$
  - At the end of the shaft: attenuation of air terminal units (päätelaitteiden päätevaimennus) or open shaft, $D_{\text{pääte}}$
- Note:
  - Sound is not attenuated in a long straight shaft
  - The designer has to consider which attenuation factors are significant and which should be left as ”safety margin”

Attenuation in a ventilation shaft
Change in cross-sectional area D_{ala}



- Attenuation is significant when the change in area is abrupt
- Attenuation is insignificant when the area changes gradually
- Usually D_{ala} can be neglected in calculations

Attenuation in a ventilation shaft
Branching of the shaft D_{haara}



- Attenuation due to branching:

$$D_{haara} = 10\log\left(\frac{S_2 + S_3}{S_2}\right),\ f > f_c$$

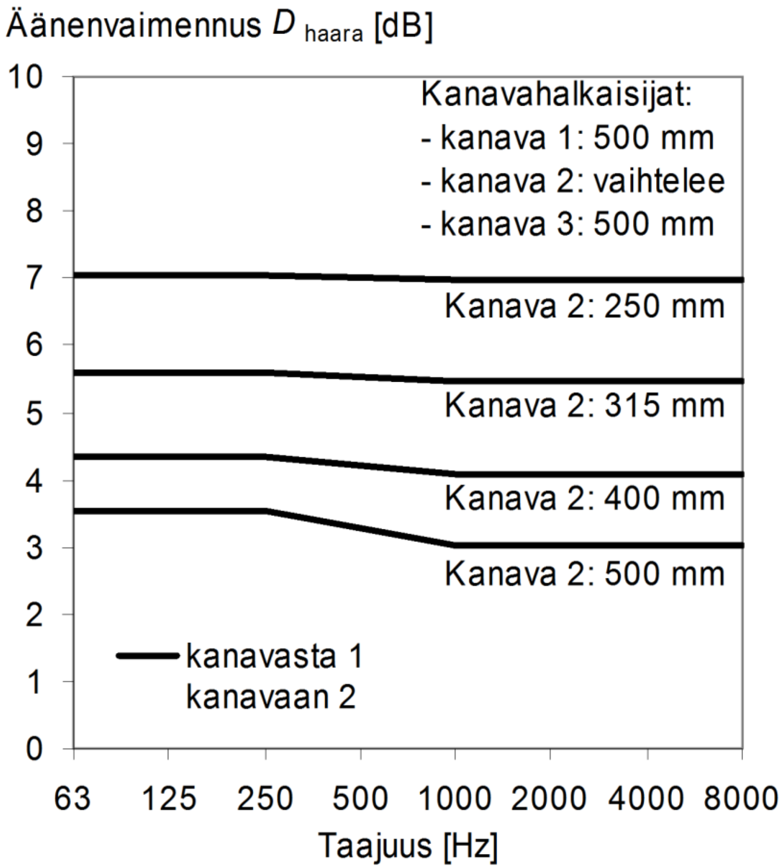
$$D_{haara} = 10\log\left(\frac{\left(\sum_i S_i\right)^2}{4S_1S_2}\right),\ f < f_c$$

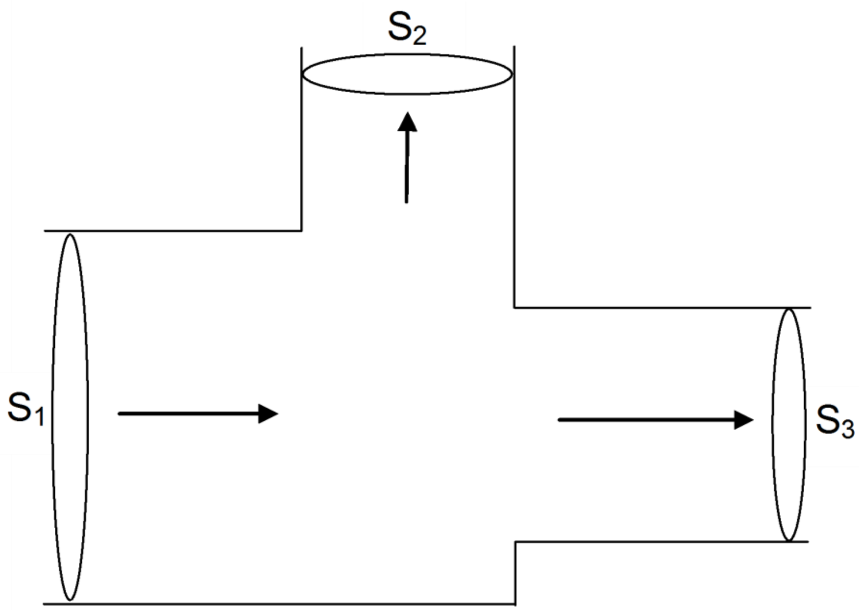
- Where S is the cross-sectional area of the shaft and f_c is a limiting frequency depending on the shaft diameter d:

$$f_c = \frac{c_0}{2d}$$

Attenuation in a ventilation shaft

Branching of the shaft D_{haara}





Attenuation in a ventilation shaft
Branching of the shaft D_{haara}

- Attenuation in relation to air flow rates:

$$D_{\mathrm{haara}} = 10 \lg \frac{Q}{q}$$

- Q = total air flow rate of the fan (l/s)
- q = air flow rate of the investigated room (inlet or exhaust) (l/s)

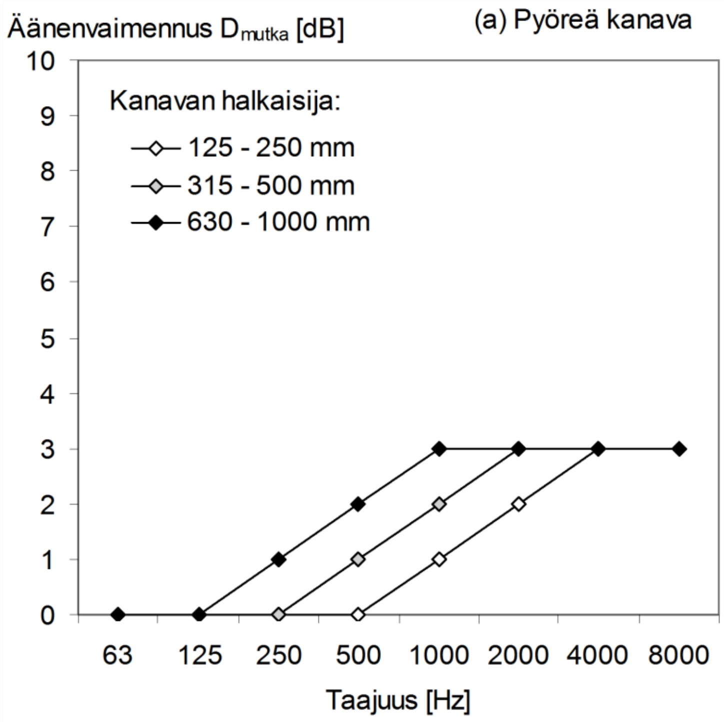
Attenuation in a ventilation shaft
Bends D_{mutka}

- Level of attenuation
  - The higher, the steeper the bend
  - In rectangular shafts higher than in round shafts
  - Improves if the bend is treated with absorption material
- Bends are done rounded so that flow noise does not increase in the bend
- Bends must not be done to achieve sound attenuation
- In calculations the attenuation of bends can often be neglected and left as safety margin

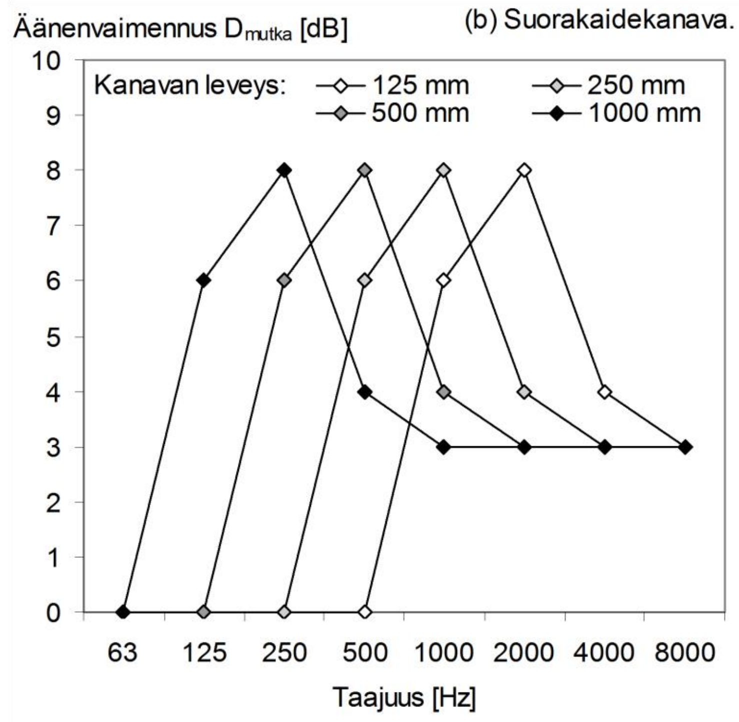
Attenuation in a ventilation shaft

Bends D_{mutka}

Round shaft, bend 90˚

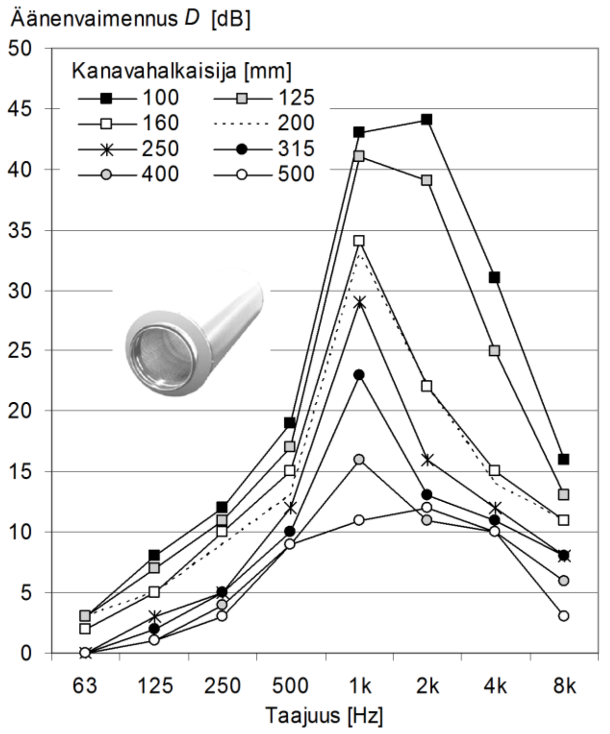


Rectangular shaft, bend 90˚

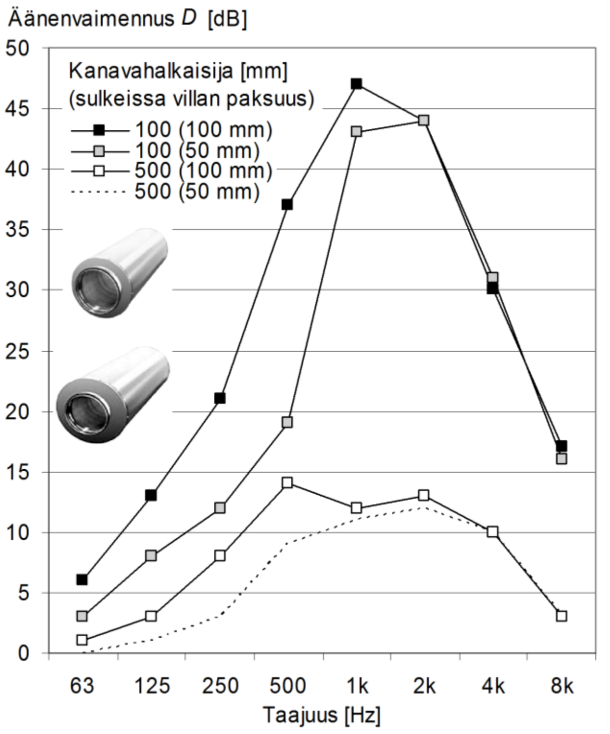


Silencers, $D_{äv}$
Round silencers

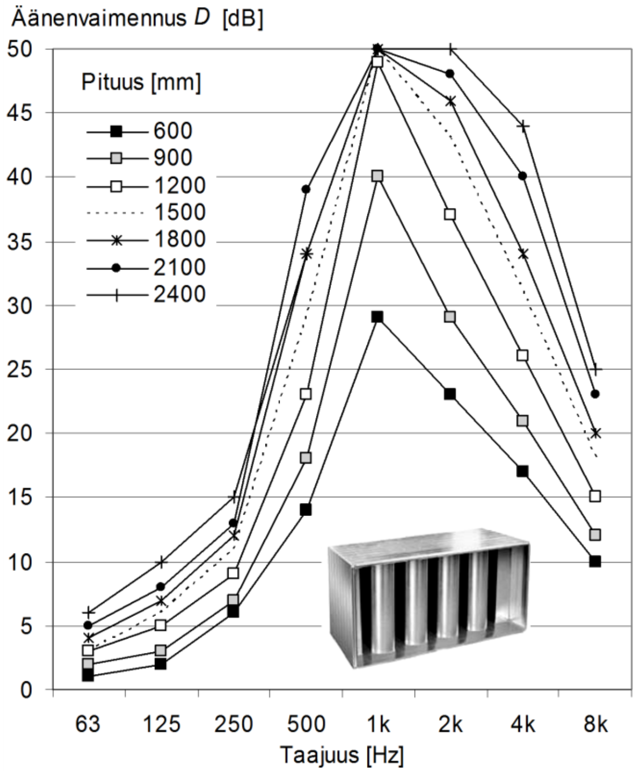
- L = 600 mm
- Mineral wool 50 mm

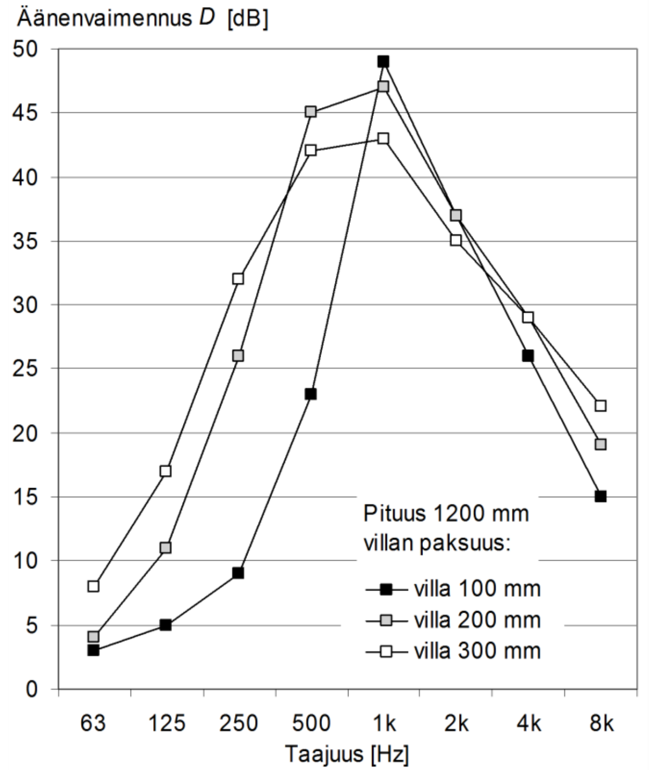


- L = 600 mm
- Mineral wool 50/100 mm



Silencers, $D_{äv}$
Lamel silencers (lamellivaimentimet)



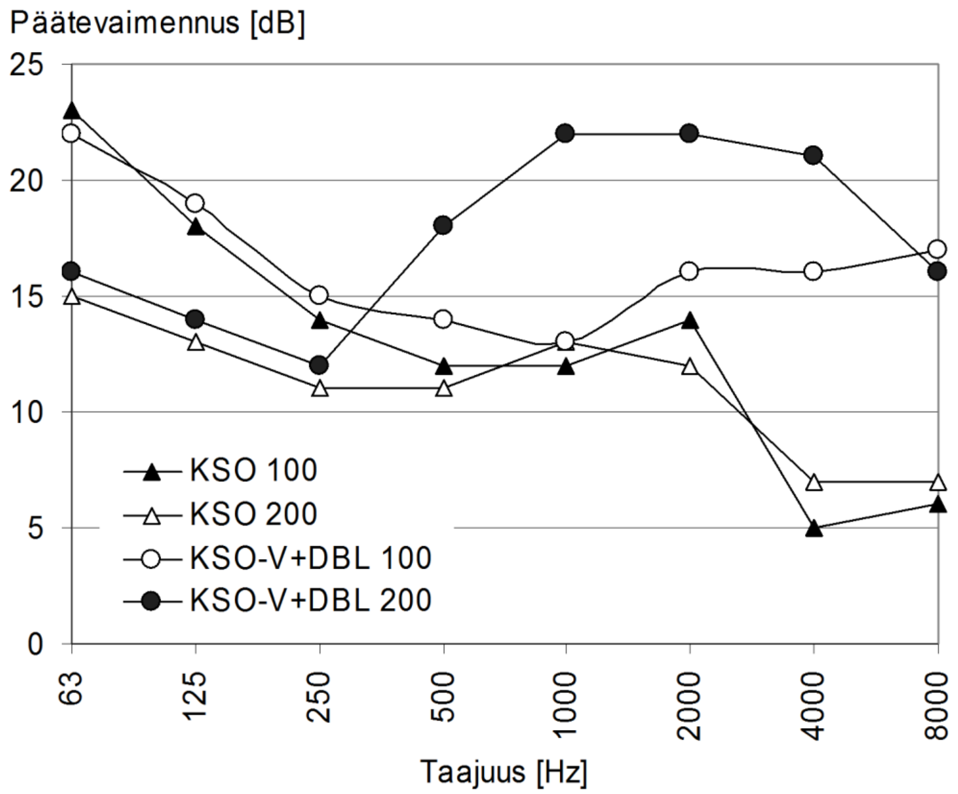


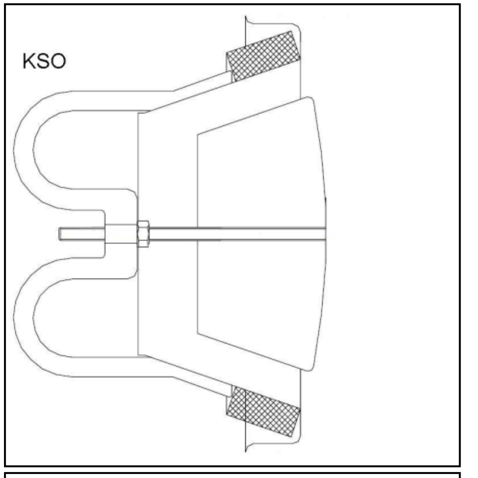
The airspace between adjacent mineral wool lamels is 100 mm

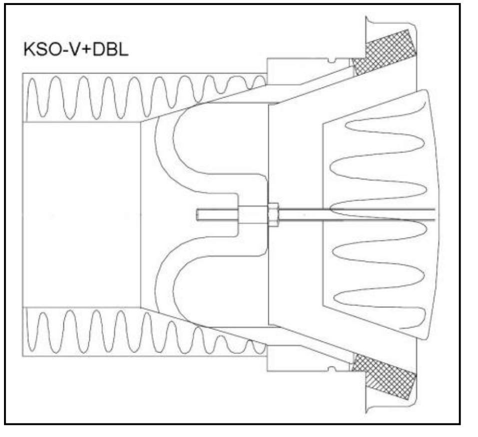
Silencers, $D_{äv}$
Installation

- Air flow speed must not increase at the silencer, because this causes noise and makes the silencer useless
- The cross-sectional area of silencers is large compared to the shaft → space requirements (width, length) need to be considered in design
- Silencer is always placed between the sound source and room

Attenuation in a ventilation shaft
Attenuation of terminal units, $D_{pääte}$







Attenuation in a ventilation shaft

Attenuation of open shaft, $D_{\text{pääte}}$

- When ventilation shaft terminates to a room surface, the cross-sectional area of the flow channel changes drastically and, thus, sound attenuates
- The attenuation caused by open shaft (no terminal unit) when sound propagates from the shaft to room can be calculated from the equation:

$$D_{\text{pääte}} = 10\lg\left(1+\left(\frac{c_o}{4\pi f}\right)^2\frac{\Omega}{S}\right)$$

- where Ω is the solid angle of sound radiation (π/2, π, 2 π or 4 π) and S is the surface area of the shaft opening [m2]

Room attenuation

- The last factor causing attenuation from the shaft to room:

$$D_{\mathrm{huone}} = 10\lg\left(\frac{A}{4}\right)$$

- Typical values of room attenuation:
  - Ventilation machinery room: 4…10 dB
  - Dwelling room: 4 dB
  - Kitchen, hallway: 0…2 dB
  - Washing room, WC: -4…-7 dB
- Remember: attenuation or amplification of sound power level depending of the absorption area of the room!

Calculation example

- Sound source: fan in an exhaust ventilation shaft leading to an office negotiation room
- Negotiation room:
  - Area: 20 m2
  - Height: 2,5 m
  - Ceiling treated with 20 m^2 mineral wool
- Total air flow rate of fan: 6000 l/s
- Exhaust air flow rate of room: 50 l/s
- Diameter of exhaust air shaft: 125 mm
- Control device and terminal unit in the shaft

Calculation example: fan

	yksikkö	63	125	250	500	1000	2000	4000
Fan, L_w	[dB]	86	88	84	79	74	68	64
Vaimennus ilmavirran jakautumisesta D_q	[dB]	-21	-21	-21	-21	-21	-21	-21
Äänenvaimennin $D_{äv}$	[dB]	-3	-7	-10	-17	-41	-39	-25
Päätevaimennus $D_{pääte}$	[dB]	-20	-16	-12	-13	-12	-9	-6
Absorptiosuhde α	-	0,11	0,22	0,69	0,96	0,96	0,96	0,96
Absorptioala A	[m^2-Sab]	2	4	14	19	19	19	19
Huonevaimennus D_{huone}	[dB]	3	0	-5	-7	-7	-7	-7
Lineaarinen äänitaso L_p = 47 dB	[dB]	45	44	36	21	-7	-8	5
A-painotus	[dB]	-26	-16	-9	-3	0	1	1
Äänitaso huoneessa $L_{A,eq}$ = 31 dB	[dB]	19	28	27	18	-7	-7	6

Calculation example: control device

	yksikkö	63	125	250	500	1000	2000	4000
Control device L_w	[dB]	64	61	59	55	49	46	46
Äänenvaimennin $D_{äv}$	[dB]	-3	-7	-10	-17	-41	-39	-25
Päätevaimennus $D_{pääte}$	[dB]	-20	-16	-12	-13	-12	-9	-6
Huonevaimennus D_{huone}	[dB]	3	0	-5	-7	-7	-7	-7
Lineaarinen äänitaso L_p = 45 dB	[dB]	44	38	32	18	-11	-9	8
A-painotus	[dB]	-26	-16	-9	-3	0	1	1
Äänitaso huoneessa $L_{A,eq}$ = 26 dB	[dB]	18	22	23	15	-11	-8	9

Calculation example: terminal unit

	yksikkö	63	125	250	500	1000	2000	4000
Exhaust air terminal unit L_w	[dB]	22	27	28	29	26	30	22
Huonevaimennus D_{huone}	[dB]	3	0	-5	-7	-7	-7	-7
Lineaarinen äänitaso L_p = 32 dB	[dB]	25	27	23	22	19	23	15
A-painotus	[dB]	-26	-16	-9	-3	0	1	1
Äänitaso huoneessa $L_{A,eq}$ = 27 dB	[dB]	-1	11	14	19	19	24	16

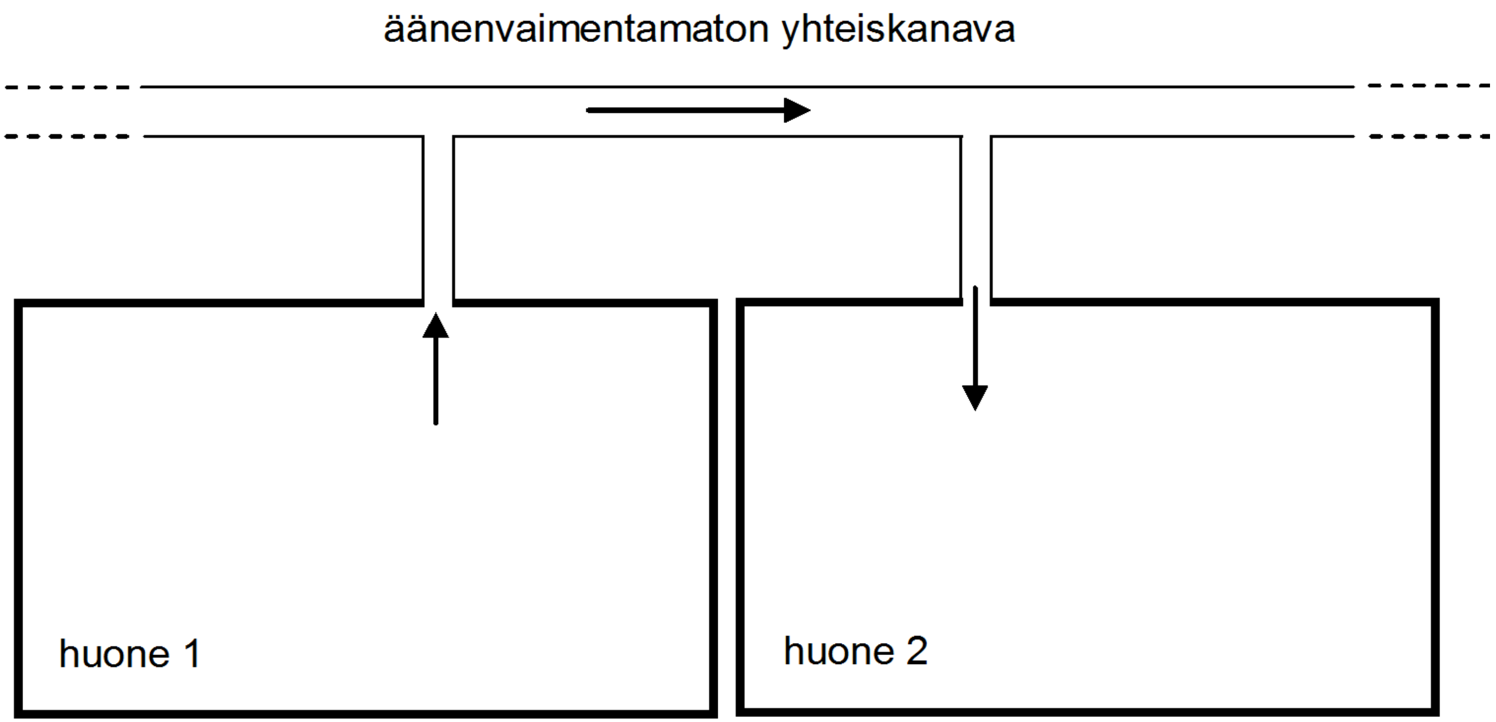
Calculation example: total SPL

	yksikkö	63	125	250	500	1000	2000	4000
Fan L_p (linear)	[dB]	45	44	36	21	-7	-8	5
Control device L_p (linear)	[dB]	44	38	32	18	-11	-9	8
Exhaust terminal unit L_p (linear)	[dB]	25	27	23	22	19	23	15
Total SPL in room, linear	[dB]	47	45	37	26	19	23	16
A-painotus	[dB]	-26	-16	-9	-3	0	1	1
Total SPL in room $L_{A,eq}$ = 33 dB	[dB]	21	29	28	23	19	24	17

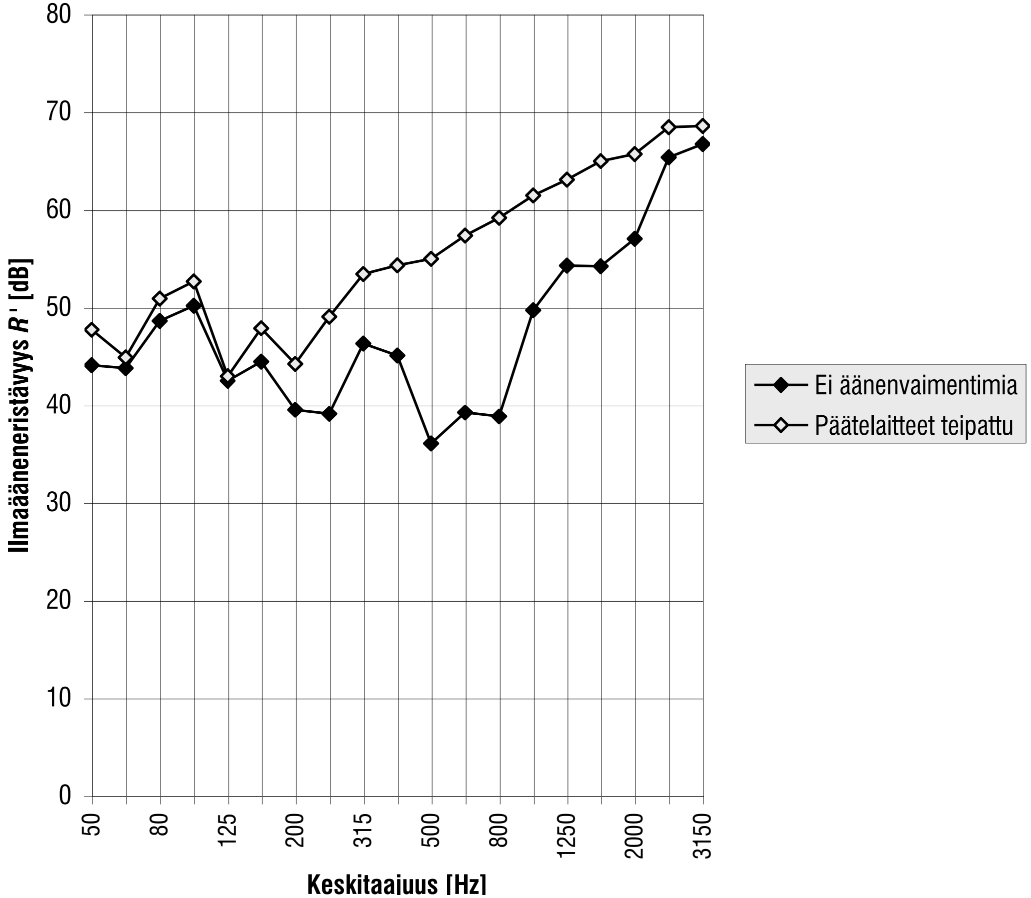
$$L_A = 10\lg\left(10^{L_{A(63\text{Hz})}/10} + 10^{L_{A(125\text{Hz})}/10} + 10^{L_{A(250\text{Hz})}/10} + 10^{L_{A(500\text{Hz})}/10} + \ldots\right)$$

$$L_{A,\text{huone}} = 10\lg\left(10^{L_{A,\text{puhallin}}/10} + 10^{L_{A,\text{säätö}}/10} + 10^{L_{A,\text{pääte}}/10}\right)$$

Flaking transmission via ventilation shaft



Flaking transmission via ventilation shaft
Example: measurement in apartment building



Flaking transmission via ventilation shaft
Sound insulation along the ventilation shaft

- ”Rule of thumb”: R'_w along the ventilation shaft must exceed the required sound insulation between spaces
  - by 10 dB, when the number of routes along shafts between spaces is 1
  - by 13 dB, when the number of routes is 2
  - by 16 dB, when the number of routes is 4
  - by 19 dB, when the number of routes is 8
- Each terminal unit constitutes one route

Flaking transmission via ventilation shaft
Sound insulation along the ventilation shaft

$$R = D_{\text{ala}} + D_{\text{lähtö}} + D_{\text{haara}} + D_{\text{mutka}} + D_{\text{äv}} + D_{\text{pääte}}$$

- D_{ala} = 10lg(4/S), S = cross-sectional area of the shaft
- $D_{\text{lähtö}}$ = starting attenuation (lähtövaimennus)
- D_{haara} = attenuation due to branching
- D_{mutka} = attenuation due to bends
- $D_{\text{äv}}$ = attenuation due to silencer
- $D_{\text{pääte}}$ = attenuation of terminal unit

Flaking transmission via ventilation shaft
Starting attenuation / lähtövaimennus

- Currently there is no accurate model for calculating the attenuation resulting from sound waves going from the room into the opening of a ventilation shaft, i.e. starting attenuation / lähtövaimennus
- Starting attenuation depends on the surface area of the shaft opening, position in the room and frequency
- Starting attenuation can be calculated with ”satisfactory accuracy” from the equation [1]:

$$D_{lähtö} = 10\lg\left(\frac{4}{S}\right) + n\lg\left(\frac{100}{df}\right)$$

- where S = surface area of the opening, d = diameter of round hole, in the case of a rectangular opening d = *squareroot(ab)*, n = 0 when terminal unit is away from room surfaces, n = 6,67 at a surface, n = 13,3 in the corner of two surfaces, n = 20 in the corner of three surfaces
- **NOTE:** When inserting $D_{lähtö}$ into the equation on the previous slide, set D_{ala} = 0 dB, otherwise the term 10lg(4/S) is calculated twice and attenuation becomes too large

[1]: Ilmastoinnin äänitekniikka (Halme, Seppänen 2002)

Flaking transmission via ventilation shaft
Sound insulation along the ventilation shaft

- When the number of paths along the ventilation shaft from room to room is *n* and the sound insulation between the rooms is $R`_{rak}$, the combined sound insulation of all the paths is

$$R` = 10\lg\left(\frac{1}{10^{-R'_{rak}/10} + \sum_{i=1}^{n} 10^{-R_i/10}}\right)$$

- where R_i is the sound insulation along the ventilation shaft of path *i* calculated with the previously-described equation

Flaking transmission via ventilation shaft
Calculation example

	yksikkö	63	125	250	500	1000	2000	4000
Starting attenuation = attenuation of the terminal unit (assumption)	[dB]	20	16	12	13	12	9	6
Haaravaimennus D_{haara}	[dB]	4	4	4	4	4	4	4
Äänenvaimennin $D_{äv}$	[dB]	3	7	10	17	41	39	25
Mutkavaimennus D_{mutka}	[dB]	0	0	0	0	1	2	3
Päätevaimennus $D_{pääte}$	[dB]	20	16	12	13	12	9	6
Kanavareitin ääneneristävyys R	[dB]	47	43	38	47	70	63	46
Vertailukäyrä (ISO 717-1)	[dB]	-	39	48	55	58	59	-
Vertailukäyrän asema, kun R_w = 51 dB	[dB]	-	35	44	**51**	54	55	-
Poikkeama vertailukäyrästä yht. < 10 dB	[dB]	-	-	6	4	-	-	-

Water and sewer systems

Noise sources of sewers

- The highest noise levels are associated with flow (water, waste) hitting the bottom angle of the sewer line
- Horisontal transitions of vertical sewer lines (pystyviemärien vaakasiirrot)
- Attachment of sewer pipes to building frame
- Noises caused by using the toilet
  - toilet seat
  - gurgling noises

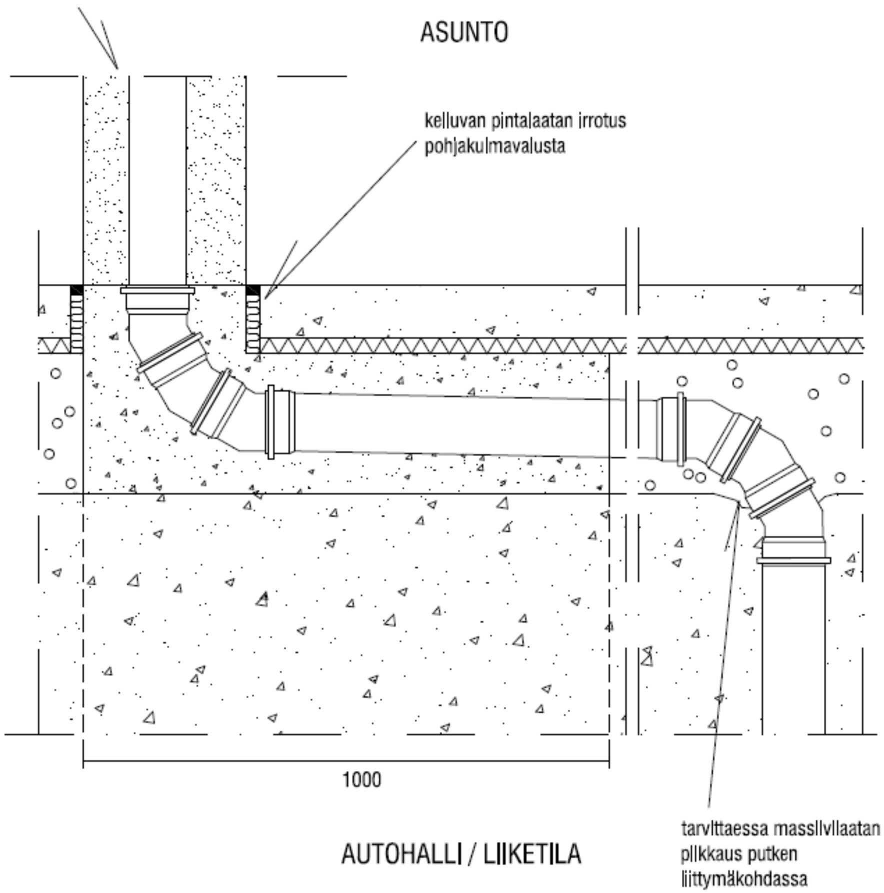
Installation types of sewers

- Acoustically the sewer system can be supported from the building frame in two different ways
- **IRTI-järjestelmä** (IRTI or ”decoupled” system): sewer pipes are attached to the building frame by using vibration isolated supports only
- **KIINNI-järjestelmä** (KIINNI or ”coupled” system): sewer pipes are firmly attached to massive concrete structures

KIINNI system
(Nowadays in use)

- Sewer pipes can be supported from massive concrete structures only
  - > 180 mm concrete
  - Or concrete floor structure
- When necessary separate support frames are used
- Sewer pipes must be alinged as straight line from top to bottom floor (no horisontal transitions in dwellings)
- Pipe penetrations through the floor are sealed with cement mortar or similar
- Mineral wool is inserted in the flues (hormit)
- At the bottom of the sewer line the pipe is cast inside concrete (so-called concrete bottom angle / betoninen alakulma), and the bottom angle is attached to massive floor structure
- The sewer must be disconnected from the ground floor slab or the possible floating floor slab of the bottom floor

KIINNI system
Bottom angle



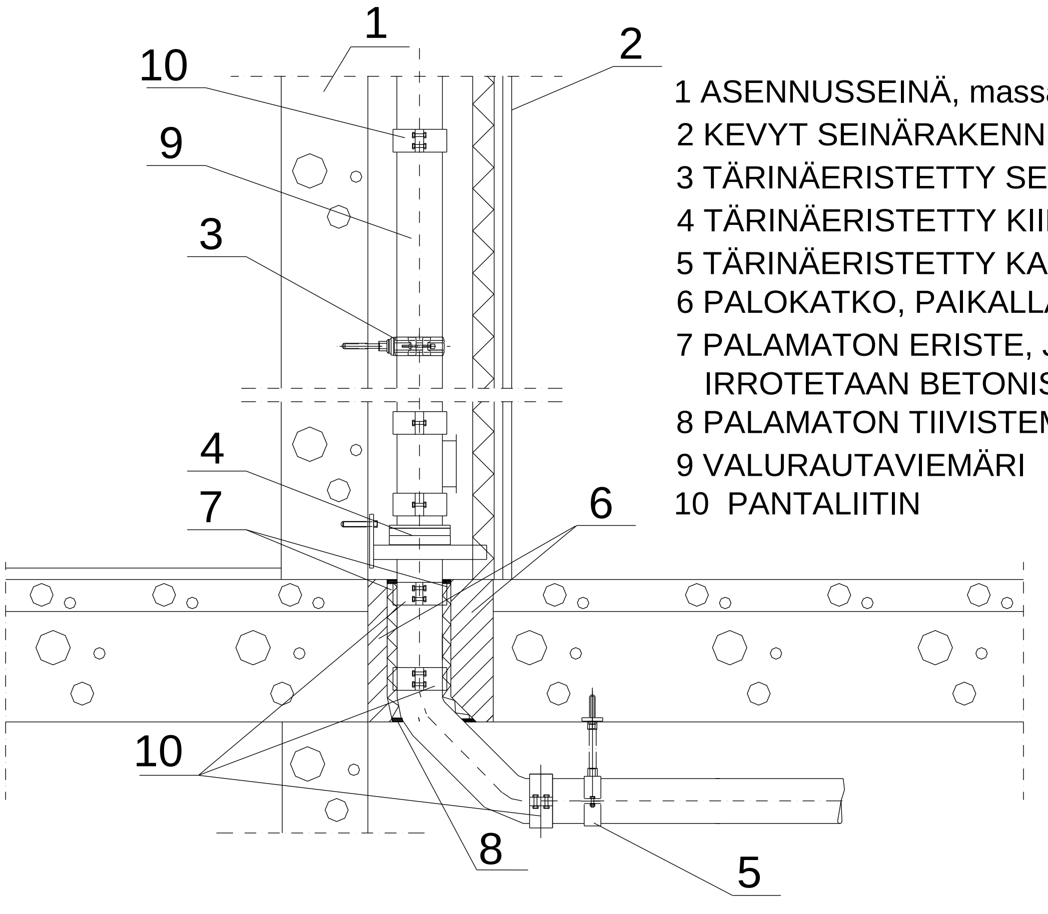
HUOM.

- ALAMUTKA TOTEUTETAAN KOLMELLA 30-ASTEEN KULMAYHTEELLÄ
- VAAKAVIEMÄRIN SIVUILLA OLTAVA VÄHINTÄÄN 100 mm BETONIA

IRTI system
(German installation method)

- Pipes must be supported from massive concrete structures only
  - > 180 mm concrete
  - Or from steel support frames attached to floor
- All attachments are done using vibration isolated supports (vertical and horisontal forces need to be considered separately)
- Sewer pipes must be alinged as straight line from top to bottom floor (no horisontal transitions in dwellings)
- In penetrations through the floor structure the pipes are disconnected from the structure using fire resistant mineral wool and elastic mastic
- Mineral wool is inserted in the flues (hormit)
- At the bottom of the sewer line there is a pipe connector at a 45 degree angle with which the flow is diverted to horisontal direction

IRTI system
Bottom angle



1 ASENNUSSEINÄ, massa > 220 kg/m2
2 KEVYT SEINÄRAKENNE
3 TÄRINÄERISTETTY SEINÄKANNAKE
4 TÄRINÄERISTETTY KIINTOKANNAKE
5 TÄRINÄERISTETTY KANNAKE
6 PALOKATKO, PAIKALLAVALUBETONI
7 PALAMATON ERISTE, JOLLA VIEMÄRI
IRROTETAAN BETONISTA
8 PALAMATON TIIVISTEMASSA
9 VALURAUTAVIEMÄRI
10 PANTALIITIN

IRTI system
Bottom angle

Heating systems

Heating systems

- Noise caused by pipes and equipment in the heating system must fulfill the regulations in RakMK C1-1998
- Pipes and installation belonging to the heating system must not deteriorate the sound insulation between spaces
  - Typical problem: poorly sealed penetrations of radiator pipes in walls and floor structures
  - Airborne and structure borne noise transmitted via radiator pipes

Electrical systems

Electrical systems

- Electrical installations between spaces transmit sound
  - Pipes
  - Cable channels
  - etc.
- Inadequately sealed electrical installations and penetrations is a typical reason for poor sound insulation betwen spaces!
- Electrical equipment cause noise and vibration
  - Transformers (muuntajat) without appropriate vibration isolation
  - Electrical conctrol rooms
  - Lighting
- Problems can also be cause by, e.g.
  - Elevators
  - Motorised doors

Vibration isolation

”Joka tapauksessa hyvän eristyksen edellytys on se, että resonanssijaksoluku saadaan tarpeeksi alas. Sen ei tulisi olla missään tapauksessa korkeammalla kuin kolmasosan vaikuttavasta jaksoluvusta.”

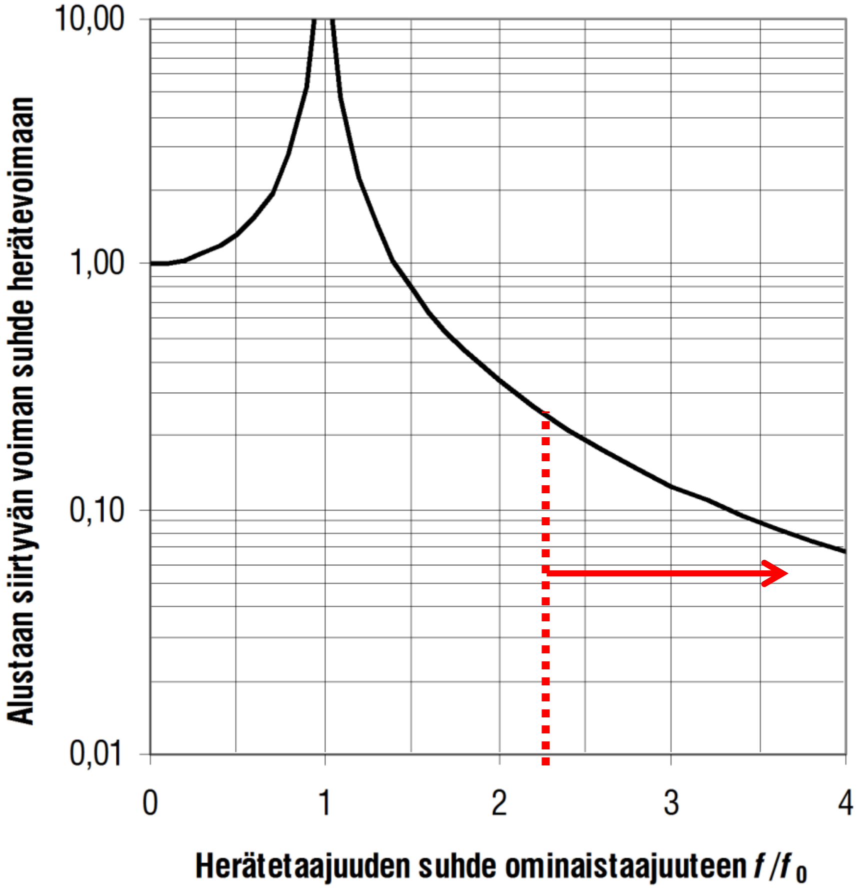
Yli-insinööri Paavo Arni 1949

Structure borne sound

- The vibration of equipment attached to the building frame causes structure borne sound
- Structure borne sound traverses along structures and excites air molecules into vibration causing airborne sound in the receiving room
- Air condition machinery usually have built-in vibration isolators, while other types of equipment have not:
  - Frequency regulators (taajuusmuuntajat)
  - Pumps
  - Compressors

  → Vibration isolation needs to specifically designed

Vibration isolation design



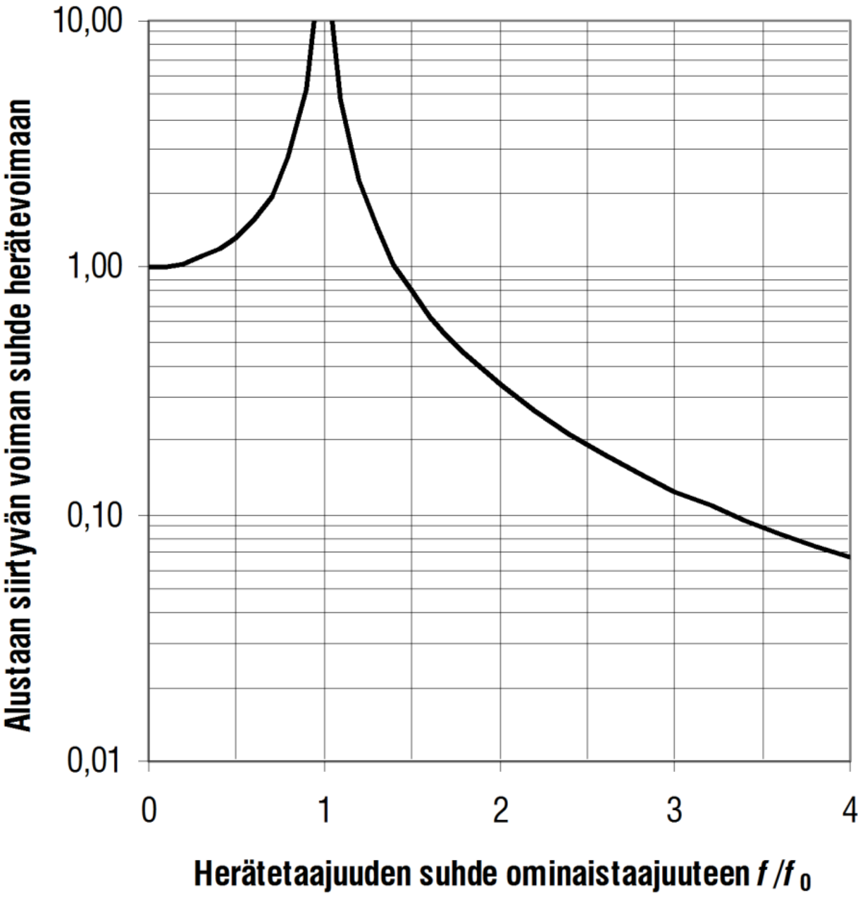
f = lowest characteristic frequency of the equipment to be isolated (e.g. the rotation speed/frequency of fan)

f_0 = resonance frequency of the mass-spring system consisting of the mass of the equipment and the vibration isolators (= spring)

Force transmitted to surface is highest when the rotation frequency of the machine equals the resonance frequency of the mass-spring system, $f = f0$

Region where vibration isolation is achieved, minimum requirement $f/f_0 > 2,5$

Vibration isolation design



$$f_0 = \sqrt{\frac{1}{4\delta}}$$

$$\delta = \frac{1}{4 f_0^2}$$

In order to achieve vibration isolation in the whole hearing frequency range, the compression of the isolators caused by the mass of the equipment must be ≥ 4 mm (f_0 < 8 Hz)

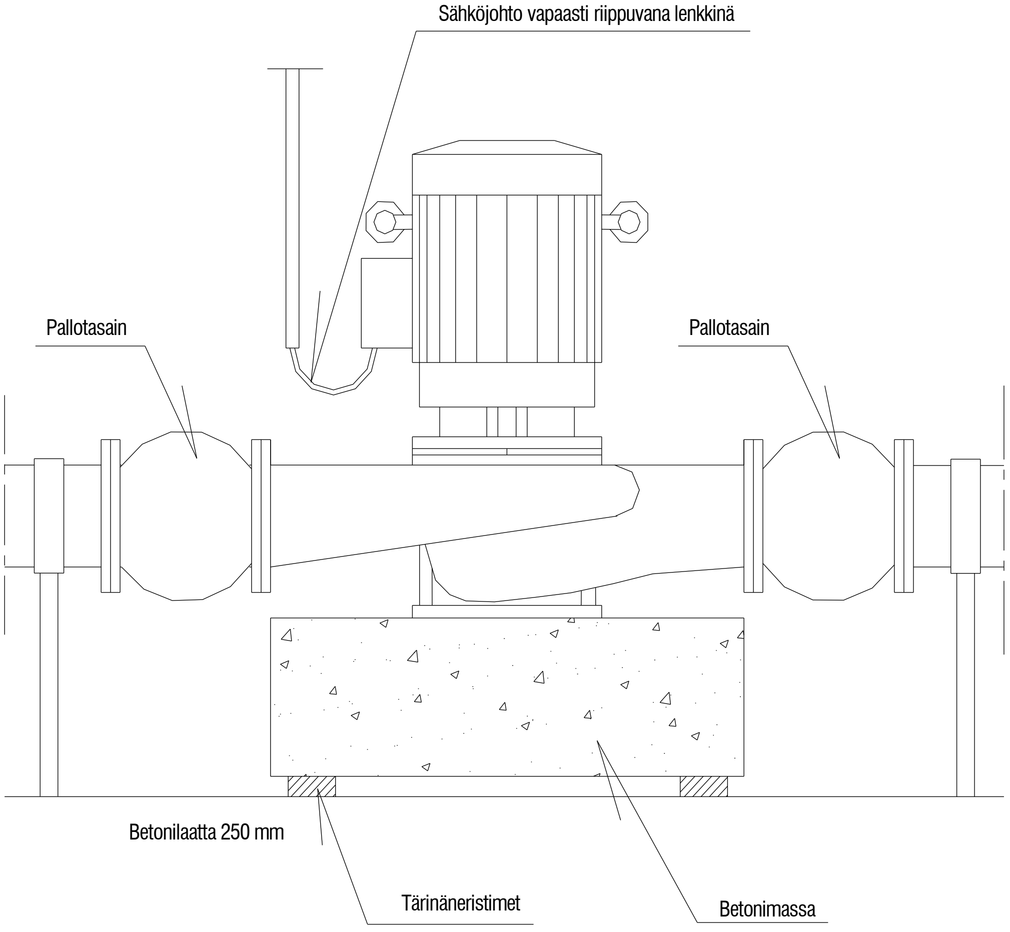
Vibration isolation design

1. Find out the lowest rotation speed of the equipment (r/min), e.g. 600 r/min
2. Convert the rotation speed to frequency (Hz), e.g. 600/60 = 10 Hz
3. Resonance frequency of the vibration isolators f_0 must be 2,5-3 times lower, e.g. 10 Hz/3 = 3,3 Hz
4. Calculate the minimum compression with equation on the previous slide, δ (in this example 23 mm)
5. If the mass of the equipment is too small, an additional mass is needed to achieve the required compression

Vibration isolation design

- Design of vibration isolation of a simple mass-spring system assumes that the surface is rigid and massive with no resonances of its own
- Floating floor must not be done below vibration isolated equipment because the resonance frequency of the floating floor may coincide with that of the vibration isolators in which case no vibration isolation is achieved
  - E.g. air condiotion machinery rooms: no floating floor because the air condition equipment have built-in vibration isolators

Vibration isolation of pump



Vibration isolation – incorrect design

Vibration isolation – correct design

Vibration systems with two masses

Mass-spring systems with two or masses need to be avoided, because the second mass causes an additional resonance around which vibration isolation decreases!