Technology Collaboration Programme on Heat Pumping Technologies (HPT TCP)

Annex 51

Acoustic Signatures of Heat Pumps

Final Report – Part 9

3 Overview on Heat Pump Component Noise and Noise Control Techniques

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Preface

This project was carried out within the Technology Collaboration Programme on Heat Pumping Technologies (HPT TCP), which is a Technology Collaboration Programme within the International Energy Agency, IEA.

The IEA

The IEA was established in 1974 within the framework of the Organization for Economic Cooperation and Development (OECD) to implement an International Energy Programme. A basic aim of the IEA is to foster cooperation among the IEA participating countries to increase energy security through energy conservation, development of alternative energy sources, new energy technology and research and development (R&D). This is achieved, in part, through a programme of energy technology and R&D collaboration, currently within the framework of nearly 40 Technology Collaboration Programmes.

The Technology Collaboration Programme on Heat Pumping Technologies (HPT TCP)

The Technology Collaboration Programme on Heat Pumping Technologies (HPT TCP) forms the legal basis for the implementing agreement for a programme of research, development, demonstration, and promotion of heat pumping technologies. Signatories of the TCP are either governments or organizations designated by their respective governments to conduct programmes in the field of energy conservation.

Under the TCP, collaborative tasks, or “Annexes”, in the field of heat pumps are undertaken. These tasks are conducted on a cost-sharing and/or task-sharing basis by the participating countries. An Annex is in general coordinated by one country which acts as the Operating Agent (manager). Annexes have specific topics and work plans and operate for a specified period, usually several years. The objectives vary from information exchange to the development and implementation of technology. This report presents the results of one Annex.

The Programme is governed by an Executive Committee, which monitors existing projects and identifies new areas where collaborative effort may be beneficial.

Disclaimer

The HPT TCP is part of a network of autonomous collaborative partnerships focused on a wide range of energy technologies known as Technology Collaboration Programmes or TCPs. The TCPs are organized under the auspices of the International Energy Agency (IEA), but the TCPs are functionally and legally autonomous. Views, findings and publications of the HPT TCP do not necessarily represent the views or policies of the IEA Secretariat or its individual member countries.

The Heat Pump Centre

A central role within the HPT TCP is played by the Heat Pump Centre (HPC).

Consistent with the overall objective of the HPT TCP, the HPC seeks to accelerate the implementation of heat pump technologies and thereby optimize the use of energy resources for the benefit of the environment. This is achieved by offering a worldwide information service to support all those who can play a part in the implementation of heat pumping technology including researchers, engineers, manufacturers, installers, equipment users, and energy policy makers in utilities, government offices and other organizations. Activities of the HPC include the production of a Magazine with an additional newsletter 3 times per year, the HPT TCP webpage, the organization of workshops, an inquiry service and a promotion programme. The HPC also publishes selected results from other Annexes, and this publication is one result of this activity.

For further information about the Technology Collaboration Programme on Heat Pumping Technologies (HPT TCP) and for inquiries on heat pump issues in general contact the Heat Pump Centre at the following address:

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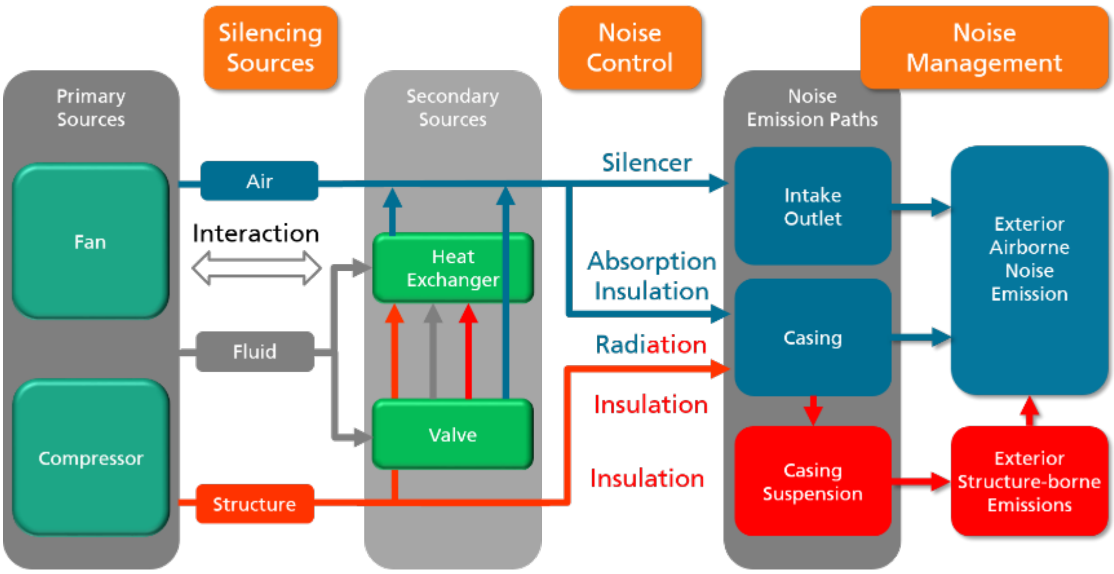
Acoustic Signatures of Heat Pumps

IEA HPT Annex 51

3: Overview on Heat Pump Component Noise and Noise Control Techniques



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1. Introduction

The global climate is changing faster than ever, and the European public is more and more informed and aware of it. The anthropogenic climate change due to the wasteful usage of fossil fuels is one of the central challenges in the 21st century. In order to assume responsibility for future generations, it is important to meet this challenge with appropriate measures and urgency. Energy used for heating and cooling accounts for almost 50 percent of the EU's total primary energy demand [1]. Hence, moving towards efficient and environment-friendly heating systems can significantly reduce primary energy consumption. In this context, heat pumps have been identified as a key technology, especially heat pumps using outside air as a heat source. By using green electricity in connection with heat pump technology, almost zero $C0_2$ emissions can be reached [2]. On the other side, when used in large numbers in urban areas, heat pumps emit a considerable amount of noise. To protect residents against noise pollution and avoid annoyance, noise emission standards need to be obeyed. In the Annex 51 Task 1 report, the national laws and regulations regarding noise protection of some European member states are summarised [3]. Besides general laws and regulations on noise protection, the Task 1 report also covers regulations which specifically address noise emissions from heat pumps.

Noise emissions from split heat pumps are usually generated by the outdoor units. For packaged heat pumps, the main noise emission comes from the air intakes and outlets. The generation and emission of noise from heat pumps is a complex matter. The block diagram in Figure 1-1 shows how the noise sources can be divided into two groups: the primary or main noise sources, such as the fan and compressor, and the secondary sources, such as valves or heat exchangers, which occur due to the self-noise (EU standard term: inherent noise) of the refrigerant and the air flowing through the system or the interaction between the compressor and the pipework and between the heat exchanger and the fan.

To describe the transmission from the sources through the casing to the environment, noise transmission and emission paths need to be considered. The noise emission paths are: direct sound radiation from the components, the airborne sound transmission through the heat pump casing, the inlet and outlet (ducts), and the vibro-acoustic transmission paths.

This report gives an overview on heat pump component noise and the fundamentals of noise control techniques that can be applied. The aim is to provide information on the characteristics of the different airborne and structure-borne noise sources and the noise control measures needed to design quiet heat pumps with low noise emissions.

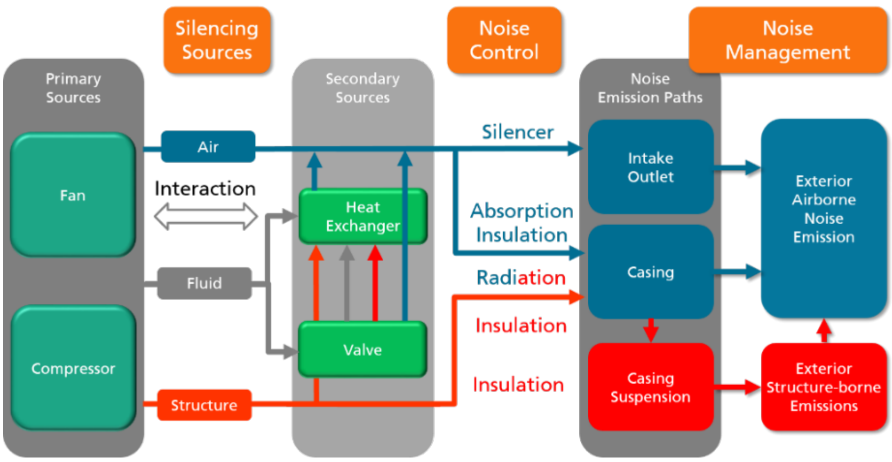


Figure 1-1: Primary and secondary noise sources in a heat pump and main airborne and structure-borne transfer paths to the exterior.

1.1. Fundamentals of noise Control

Noise can affect physiological and psychological health by generating stress [3]. Hence, direct consumers and the wider public have an interest in quiet heat pumps. This puts pressure on heat pump manufacturers to produce heat pumps that are energy-efficient, quiet, and economical at the same time.

The most efficient method to reduce noise emissions is to suppress the noise generation at the source. In heat pumps, this can be achieved by using more quiet compressors or fans. The next best way to reduce emissions is to decouple the main noise source from the rest of the heat pump. For example, using elastic mounts to fasten the compressor to the heat pump frame can greatly reduce the vibrational power transmitted into the frame [3]. Any source may transmit airborne or structure-borne noise via a number of different parallel paths. Hence, the further away from the sources, the more difficult it is to control noise transmissions efficiently. If noise control measures at the heat pump are not efficient enough, it may be necessary to protect the surrounding with additional expensive retrofit solutions such as noise barriers or acoustic housings (Section 5.2.2).

Classical noise and vibration control measures are usually passive, such as rubber mounts or mufflers based on porous material. In recent years there have also been developments in the area of active noise and vibration control, such as active engine mounts. To date, the application of active noise and vibration control measures is still restricted to a small number of special applications in the automotive and aerospace industry. Although active solutions are becoming increasingly accessible, they are currently not used in heat pumps due to their complexity and relatively high costs.

1.2. Description of a heat pump

This section presents several types of heat pumps. The individual steps of the heating process are briefly discussed. The main task of a heat pump is the transfer of thermal energy (heat) from an energy source, e.g. ambient air, groundwater, or geothermal resources, to a second medium (air, water), which is then used in the heating system of a building. There are two kinds of heat pumps; on the one hand, the packaged system which has only one unit, installed in the house (Figure 1-2 b); on the other hand, the split heat pump systems with two units – an indoor and an outdoor unit (Figure 1-2 a).

a) b)

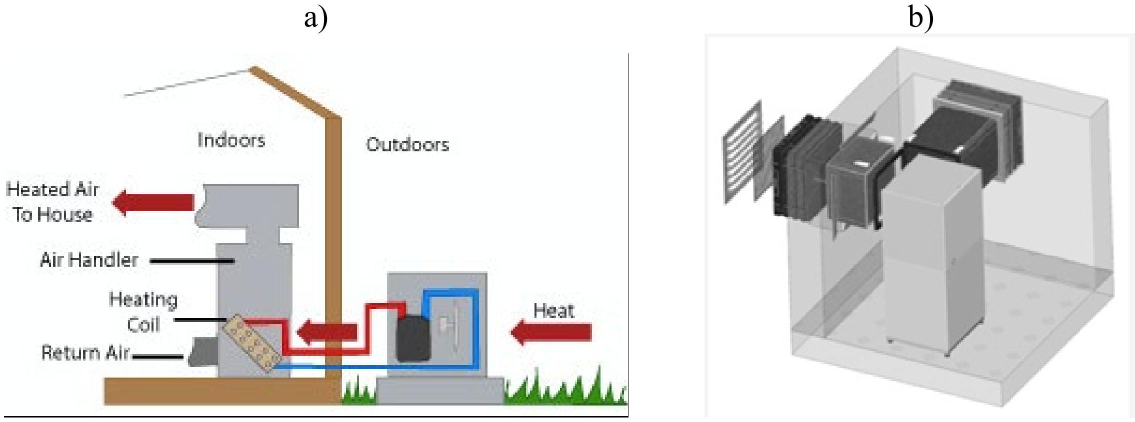


Figure 1-2: Sketches of common heat pump systems; a) a split heat pump with indoor and outdoor unit [4] and b) a packaged unit system [5].

A packaged heat pump allows to have all components (fan, heat exchangers and compressor) in one cabinet. Air supply and return ducts lead the air through the building's exterior wall to the packaged unit, which is often placed in the basement. Such heat pumps have both the heat transfers to the refrigerant and the heat transfer to the medium of the heating system in the same unit. As the ducts have a higher flow resistance than the outdoor units, mainly radial fans are installed in these systems.

Split heat pumps have an indoor and an outdoor unit which are connected through an isolated conduit. These systems split the heat transfer process into two sections. The first heat transfer from the air to the refrigerant occurs in the outdoor unit, the second heat transfer from the refrigerant to the medium of the existing building heating system occurs in the indoor unit.

The heating process starts with warming up the refrigerant. The fan (axial or centrifugal) draws in ambient air through a heat exchanger, which absorbs the energy from the ambient air and transfers it to the refrigerant circuit via convection. This process causes a phase transition of the refrigerant from fluid to gas. The refrigerant has the property to evaporate even at temperatures below 0 °C in order to guarantee the function of the heating system even during periods with outdoor air temperatures around or below 0 °C.

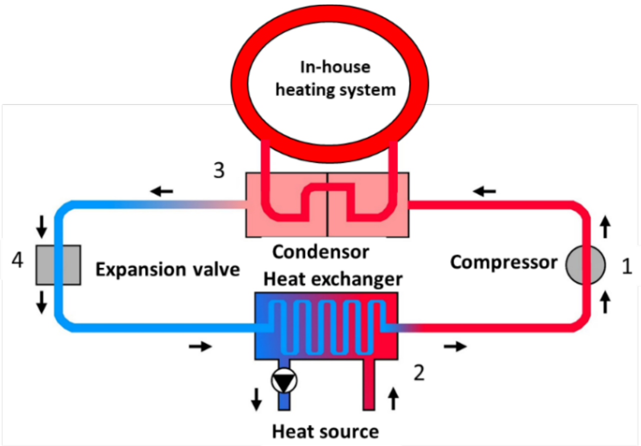


Figure 1-3: Schematic process of the heat transfer in a heat pump.

The gaseous refrigerant is compressed in the compressor, which causes a rise in temperature and evaporation. Usually, scroll compressors are used in heat pumps (Figure 2-13 and Figure 2-16). The liquid phase is precipitated in the compressor and fed back to the refrigerant circle, near the place where the refrigerant reaches the heat exchanger. In parallel, the gaseous part of the refrigerant reaches the condenser. At this stage the energy of the meanwhile hot refrigerant gas is transferred to the medium of the central in-house heating circuit. Finally, the refrigerant is liquefied and expanded by an expansion valve to restore its initial state.

In the following sections, the components are discussed in more detail both from an acoustic point of view and with regard to the application of noise control techniques.

1.3. Component noise

Any "active" mechanical component may be described as three different sources but also as transfer functions:

- Airborne source
- Structure-borne source
- Fluid-borne source

The characterisation of airborne sources is well-known and can be achieved on a standalone mode. To characterise the contribution of structure-borne and fluid-borne noises, the integration parameters have to be known (characterisation of receiving structure or circuit).

Transfer functions are then to be evaluated according to the vibro-acoustic scheme (Figure 1-4).

Like heat exchangers, some components can be considered as transfer function and noise sources: isolation of the noise of other components placed behind them and source due to circulating fluids inside and outside.

The objective of the noise approach has also to be considered : the characterisation of components will be different if the objective is to reduce stationary or transient noise, or if noise level or annoyance is addressed.

Considering all the sources and transfer paths presents an enormous task. The vibro-acoustic scheme is intended to support the hierarchisation of the contributions. It is the expertise of acousticians which will help to make the relevant hypothesis.

1.4. Noise analysis approach: acoustic synthesis

The noise reduction of heat pumps is complex and far-reaching. Each component can be considered as a "black box" that emits acoustic and vibrational energy in three forms:

- Airborne noise: acoustic radiation from the component casing (source) and the nearby structures, propagating in the surrounding air; it is measured using microphones or sound intensity probes.
- Hydraulic or fluid-borne noise generated in the piping systems and radiated: it originates in the pressure pulses emitted by the source in the liquid columns of the system (at both the suction and the discharge of the compressor, for example) and which are broadly responsible for the noise emission by the piping. It is generally evaluated using flush-membrane pressure sensors.
- Structure-borne noise: Components generate dynamic forces that are transmitted to the other components or the heat pump. This structure-borne noise is generally evaluated by using force sensors, accelerometers, or both.

Each component can include several noise generation mechanisms, and each source can generate airborne, structure-borne, or fluid-borne noise transmitted in the components, which have different levels of response to each of these noises. It is evident that, with a view to an overall reduction of the noise of such a circuit, the main issue is to manage the interaction of these sources with the components to which they are connected; these components must be taken into account and identified with respect to the source in order to obtain a "silent system". Thus, an elemental vibration and acoustic analysis of such a system is essential, as an overall approach cannot lead to an optimum solution. However, in general, the supplier of any of the (active) components may not necessarily know the other components of a system, and is therefore not in a position to adapt its source to the actual needs, if necessary. It is common for a given component, even if it is relatively quiet, to be installed in two different circuits to generate distinct and potentially damaging vibration and acoustic behaviour. Information sharing is necessary, particularly regarding source/structure interfacing, and the common work of suppliers and integrators is consequently essential for the joint execution of an action, for the improvement of the product, using an acoustic synthesis.

An acoustic synthesis is a design or diagnostic approach consisting of breaking down a system into noisy or vibrating components and transmission paths, in order to control its vibration and acoustic behaviour to a required reception point (Figure 1-4). A performance criterion to be optimized is generally assigned to this reception point (e.g. sound power level of a machine or component, psychoacoustic descriptor); optimisation generally requires numerical modelling to view and quantify the "acoustic situation" of a machine or product to control its behaviour.

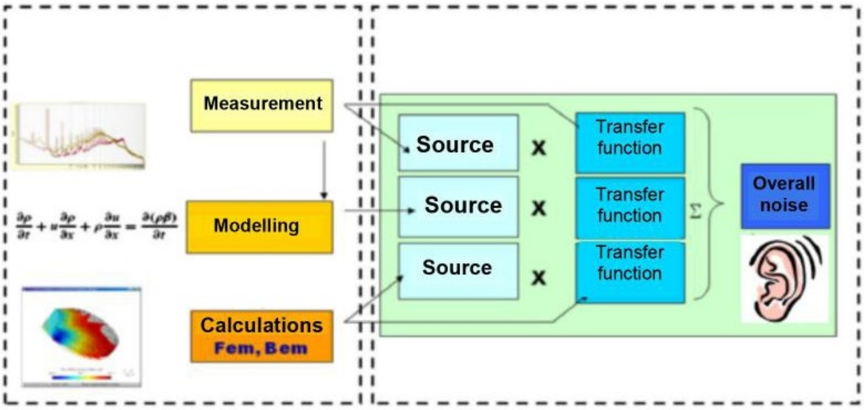


Figure 1-4: Vibro-acoustic scheme for acoustic synthesis approach.

Each component can be fully described by experimental or numerical data or both, as well as by analytical or empirical formulations. The approach accepts the sub-model concept, enabling the development of models with components not available at present, which will eventually be incorporated into the complete system. Modelling of the active components, treated as sources in the acoustic synthesis approach, can be done by using several different approaches:

- on the basis of a morphological representation of the component, using FE (Finite Element) and CFD (computational fluid dynamics) software;
- on the basis of experimentation and a reverse method, characterizing the source according to operating parameters (determination of the terms of the impedance or admittance matrix).

Passive components are treated as transfer functions in the acoustic synthesis approach but necessitates a non-trivial experimental validation approach. According to their place in the circuit, some components have to be treated as both sources and transfer functions. To enable substructuring, each part of the assembly and each substructure (active parts – the sources – passive parts – the transfer paths) are fully described at these interfaces, so that compatibility between the substructures is maintained while taking into account any interaction between the active and passive substructures.

2. Overview of Components Noise

The following parts of this document will describe the acoustic behaviour of the main components of heat pumps.
The main noise sources are the fan and the compressors. They are described in the following sections. Other sources such as the heat exchanger contribute only insignificantly to the overall noise.

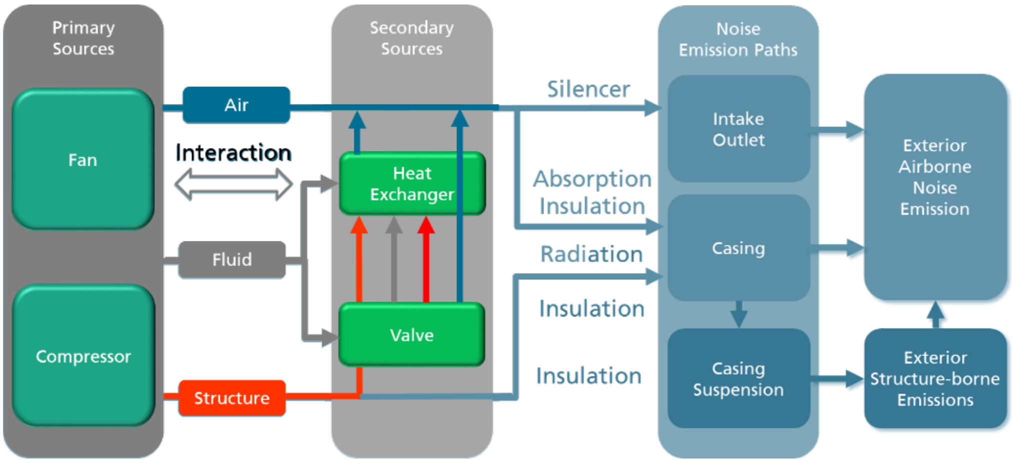


Figure 2-1: Block diagram indicating the primary and secondary noise sources of the heat pump.

2.1. The fan

In order to correctly describe the acoustic behaviour of a fan, some terms have to be defined:

Blade:

- Leading edge: part located upstream of the blade in relation to the flow direction .
- Trailing edge: unlike the previous edge, this represents the downstream end of the blade.
- Chord: direct length between the trailing edge and the leading edge.
- Axial chord: projection of the chord on the rotation axis of the fan.
- Camber: angle formed by virtual lines tangential to the leading edge and the trailing edge.
- Pressure side/suction side: convex and concave surfaces of the blade.

Reaction/action/radial: in centrifugal fans, the direction of the blades is used to differentiate three types of blade arrangement (Figure 2-2). If the blade is curved in the direction of rotation, the fan is a “forward-curved blade fan”. Conversely, if the blade is directed in the opposite direction of rotation, the fan is a “backward-curved blade fan”. If the blade tip is radial, the fan is called a “radial blade fan”.

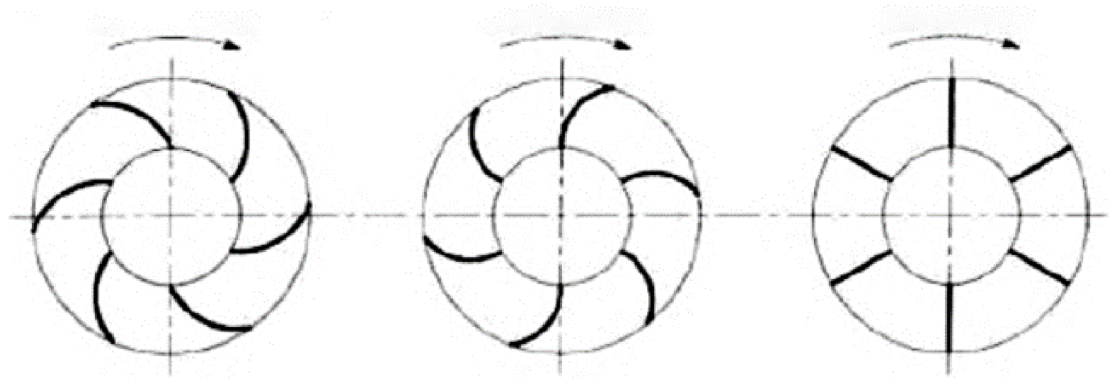


Figure 2-2: Blades in reaction, in action and radial.

- Shroud (or housing): fixed part around an axial fan.
- Stator: fixed part of the fan that affects the aerodynamics, such as the volute of a centrifugal or cross-flow fan, or the inlet or outlet guide vanes of an axial fan.
- Impeller: moving part of the fan with a standard number of blades in an equidistant (or non-equidistant) distribution.
- Hub to tip (H/T) ratio: on centrifugal fans: ratio of the diameter at the leading edge to the diameter at the trailing edge. On axial fans: ratio of the hub diameter to the tip diameter. On high pressure backward-curved centrifugal fans the H/T ratio is usually small. Conversely, on axial fans: the higher the H/T ratio, the higher the pressure (e.g. vane axial fans).
- Aspect ratio: ratio of the blade chord to the blade span.

In operation, the fan presents the aerodynamic and acoustic characteristics listed below:

- Pressures and power:
  - Outlet dynamic pressure: pressure equal to the square of the average flow velocity at the fan outlet multiplied by half of the air density.
  - Fan total pressure: difference between the total pressures at the fan outlet and the fan inlet.
  - Fan static pressure: difference between the fan total pressure and the outlet dynamic pressure.
  - Fan air power: product of flow rate multiplied by static or total pressure at any operating point.
  - Impeller power: mechanical power on the axis of the fan, measured or deduced from the electric power, if the motor efficiency is known.
- Efficiency: ratio of the air power to the impeller power. Static efficiency uses static pressure whereas efficiency requires the use of the total pressure of the fan.
- Velocity triangle: this triangle indicates the amplitude and direction of the flow velocity through the impeller. A triangle is formed of the absolute velocity v (flow velocity in the fixed coordinate system), blade velocity u (velocity of the blade at radius considered) and relative velocity w (flow velocity in the coordinate system rotating with the blade).
- Blade passage frequency (BPF): first frequency of the tonal noise of a fan with equidistant blades, this frequency is equal to the number of blades times the rotation speed in rps or Hz (number of rounds per second).

Aerodynamic approach

The fan responds to a flow resistance imposed by the user's application. The fan must therefore overcome the pressure drop of the distribution ductwork while providing the flow rate required by the user.

Concept of pressure loss

The total pressure of a small volume of air in movement is equal to its dynamic pressure plus its static pressure. The fan provides the necessary energy to compensate for the total air pressure difference between intake and exhaust of the system in which it is installed in order to make air move in the system by compensating for air friction losses. This total pressure difference is called "pressure loss", Δp. This pressure loss corresponds to the total pressure loss associated with the resistance of the system at a given air flow rate.

Friction, the cause of pressure loss, increases with the flow velocity. Laminar or turbulent energy-dissipating phenomena are therefore a function of the flow rate in the system. Therefore, for a given ductwork (thus representing the environment into which a fan is installed), a characteristic graph is plotted representing the variation in pressure (or pressure loss) in relation to the flow rate (Figure 2-3). This graph relates to specific features of the ductwork (presence of obstacles upstream or downstream, length and shape of the ducts, change of direction, …).

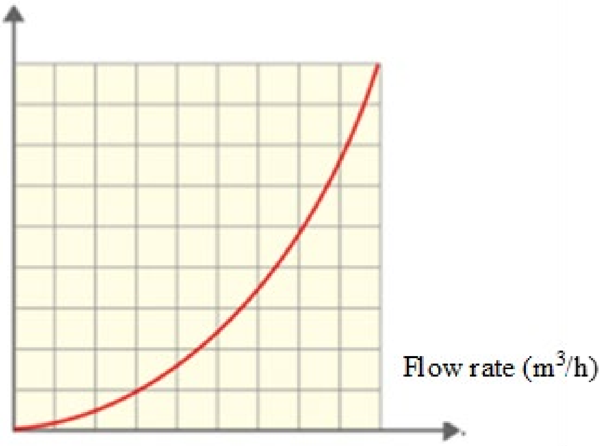


Figure 2-3: Characteristic graph of any distribution network.

<u>Performance of a fan</u>

The air performance curve of a fan is the graph representing the variation in fan pressure in relation to the flow rate for a given fan size and speed of rotation. This air performance curve represents the way to characterize the performance of a fan in addition to its geometry, cost, and potential noise constraints.

Based on the fan performance curve and the system resistance curve, an operating point corresponds to a flow rate-pressure pairing for which the pressure supplied by the fan is equal to the pressure loss of the circuit. The operating point defines the unique pressure-flow rate point of the fan at a given speed (Figure 2-4).

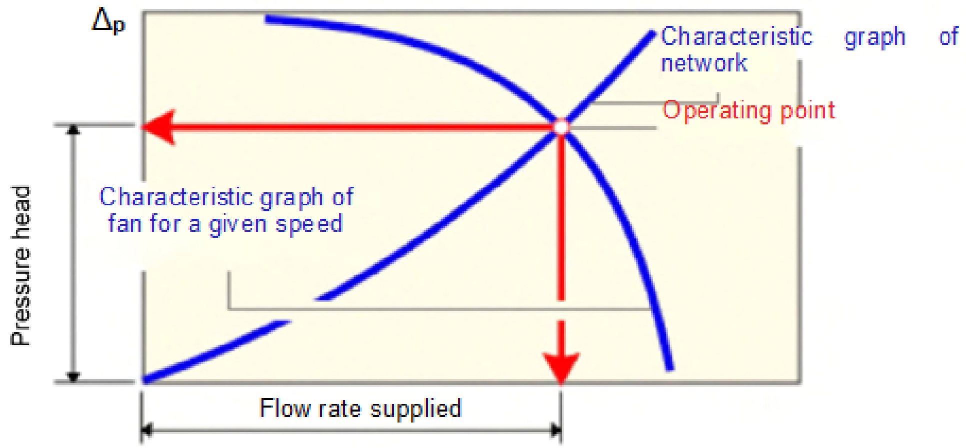


Figure 2-4: Determination of the operating point of a fan within its environment.

The concept of a reduced performance curve is used to determine a single performance curve of a fan whatever the rotation speed. The air performance of the fan is no longer dependent on pressure and flow rate but on two non-dimensional coefficients μ and δ, known as the pressure coefficient and the flow coefficient, respectively:

$$\mu = \frac{\Delta p}{\rho_0 \cdot U^2} \text{and } \delta = \frac{Q}{U \cdot S} \qquad \text{Eq. 1}$$

Where (SI units):

- Δp: fan static pressure at operating point
- Q : fan flow rate at operating point
- S : outlet area of the impeller
- U : peripheral speed of the impeller (U = πDN)
- ρ0 : air density at operating temperature

These coefficients are used to give the operating points of the fan in a system for any speed of rotation (see Figure 2-5 as an example). The operating points (μ-δ) of the fan will be the points where the reduced curve of the blower crosses the reduced system resistance curves for various ventilation configurations .

Note: A single system resistance curve with reduced coordinates is obtained for variable speeds if the pressure losses of the system are proportional to the square of air flow rate, which may not be valid for some components (air filters, for instance).

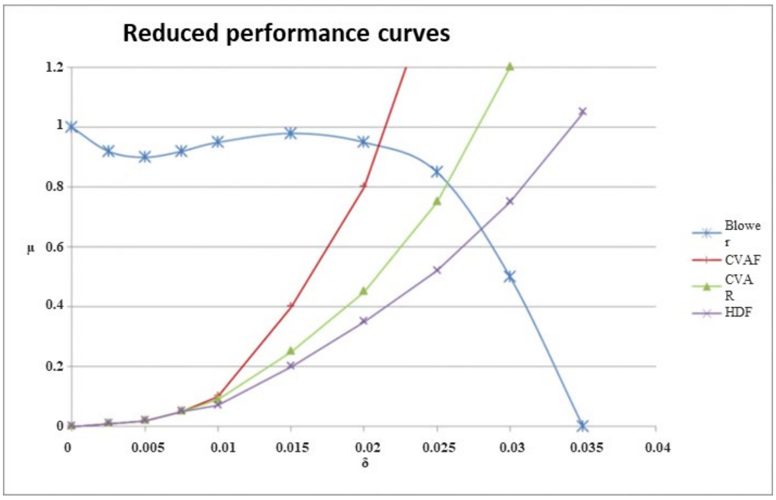


Figure 2-5: Example of reduced graphs of HDF, HFF and CVAF (pressure drop) ducts and the blower.

Air performance and characteristics of fans

Like compressors, there are multiple types of fans which meet a range of operating requirements (high flow rate/low pressure, medium flow rate/medium pressure, low flow rate/low or high pressure, …). Since a fan provides a limited range of flow rate-pressure couple, there are four categories of fans (Figure 2-6) that provide solutions to the majority of industrial problems:

- Axial fans: the air flow remains parallel to the axis of rotation of the impeller. Axial fans, especially propeller fans, are useful if a large flow rate at low pressure is required.
- Centrifugal fans: the air enters the impeller parallel to its axis whereas the outlet flow is radial. The centrifugal effect of the rotation allows an additional pressure to the pressure generated by the blades. Therefore, these fans, or more precisely the backward-curved centrifugal fans, are used to generate a moderate flow rate at high pressure.
- Mixed flow fans: entering axially, the disrupted air leaves the fan at an angle determined by the blade geometry. The centrifugal force combined with the forces exerted by the blades allows an increase in pressure. The performances are intermediate between those of axial and centrifugal fans.
- Cross-flow fans: even though the impeller is similar in shape to that of the forward-curved centrifugal fans, the path of the fluid is perpendicular to the axis of rotation. This path triggers the appearance of a vortex, the location of which depends on the operating point. In cases where it is necessary to obtain a high flow rate at low pressure, this type of fan is appropriate.

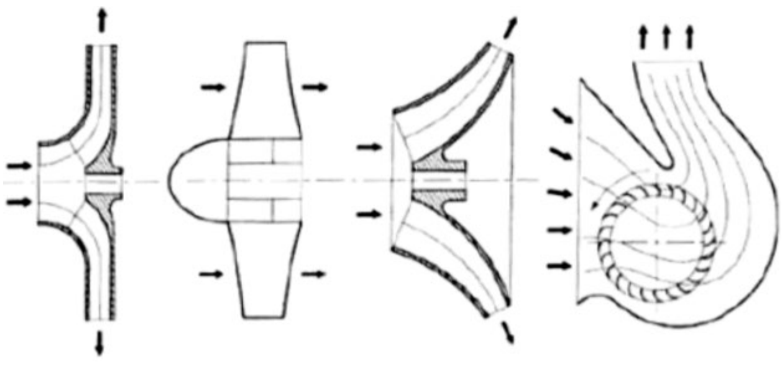


Figure 2-6: Centrifugal, axial, mixed-flow and cross-flow fans.

In this section, fans of all types (axial, centrifugal, mixed, and cross-flow) will be investigated to show the geometrical characteristics and air flow performances as well as their efficiency.

Axial fans

There are three types of axial fans:

- Tubeaxial fans: these fans are composed of a wheel with a medium number of blades (between 3 and 10) and without outlet guide vanes. The impeller is integrated into a cylindrical housing with a diameter slightly larger than the impeller. The fan generates a pressure of no more than 1 kPa and produces a flow rate with optimum efficiency (up to about 80 %) at three quarters of the maximum flow rate.
- Vaneaxial fans: these fans, consisting of airfoil blades with a hub to tip ratio greater than 50 %, have outlet guide vanes to make the flow at the fan exit axial in order to recover some static pressure and increase efficiency compared to a tubeaxial fan with the same impeller. The fan pressure may reach 3 kPa or even more.
- Propellers: with 2 to 6 blades, these fans are useful where it is necessary to deliver a flow for which the static pressure does not exceed about 200 Pascal. Therefore, these fans (some of them look like "Mickey Mouse ears" (see Figure 2-7) are particularly useful in condensing and cooling unit applications. With a total efficiency reaching 60 %, this type of fan is also useful because it can provide satisfactory results at low or high flow rates (40 to 100 % of the maximum deliverable flow rate).

Figure 2-7: Propeller fan with "Mickey Mouse ears" shaped blades.

Centrifugal fans

There are three types of centrifugal impellers: backward-curved (BC) blades, forward-curved (FC) blades (or squirrel cage), and radial blades. The BC centrifugal fans have between 8 and 16 blades curved or inclined in the opposite direction to the rotation. The operating point of the fan is determined by its aspect ratio (the higher the aspect ratio, the more the flow rate increases with a reduction of pressure). Where a fan is enclosed by a volute housing, its efficiency is higher (about 90 % for very well designed impeller) than without housing (plug fan or in-line centrifugal fan). Centrifugal fans with radial blades present an aspect ratio similar to BC fans, but their efficiency is much lower. Radial centrifugal fans are generally useful for industrial problems in dealing with flows loaded with solid particles. FC centrifugal fans, or squirrel cage fans, are very useful when the application requires low-pressure ventilation (1000 to 1500 Pa maximum). These fans have a large number of blades (24 to 60 according to the diameter) and a small blade chord, which results in a high hub to tip ratio. Efficiency is generally not more than 70 %.

Mixed-flow fans

Mixed-flow fans are much less common than axial, centrifugal, or even cross-flow fans. Their behaviour is close to that of axial fans. It is simply necessary to take account of the centrifugal component of the flow to approach the real aerodynamic performance and, therefore, as will be discussed later, the acoustic performance. It is useful to note that for the same air performance, the noise of the mixed-flow fans is generally lower than that of the axial fans. This may be a useful feature if the industrial issue requires a low noise level for an operating point with a higher pressure. The shroud of mixed-flow fans often has the same geometry as that of an axial fan.

Cross-flow-fans

This fan presents strong advantages when there is a need for an air-conditioning system requiring a relatively high flow rate at low pressure with a moderate noise footprint. The impeller of the cross-flow fan is very similar to that of a FC centrifugal fan, despite a much higher length, but the passage of the flow through the impeller is very different, as the flow is essentially 2D and remains in a section normal to the axis (Figure 2-8). The operating point depends on the position of a vortex located across the blades near the cross. At a low flow rate, the vortex, which approaches the centre of the impeller, triggers instabilities in the fan operation. The air performance depends on the shape of the wheel and the volute. The closer the volute and the cross are to the wheel, the higher the air performance, but the higher the noise level.

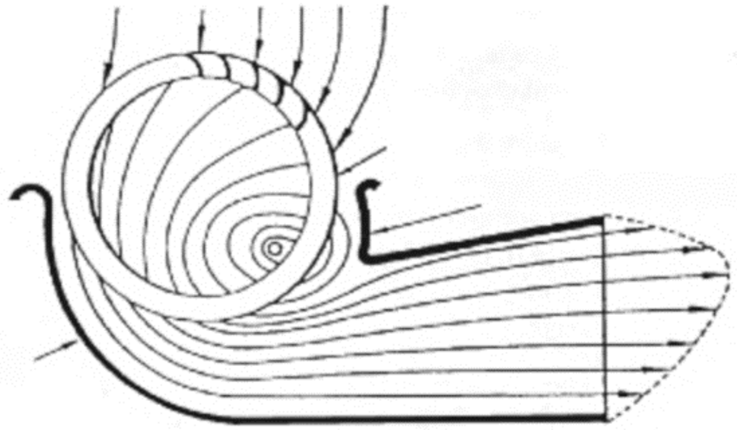


Figure 2-8: Flow path in a cross-flow fan (and appearance of a vortex).

Sources of noise

The fans produce three categories of noises of different origins: aerodynamic, electromagnetic and mechanical.

The electromagnetic noise is associated with the electric motor in operation. The main cause of the electromagnetic noise is associated with the interaction between the rotor (moving part of the motor) and the stator (static part of the motor) which varies the magnetic field and therefore the forces generated. These forces cause a solid-borne noise by the stator and the fan support structure. A noise of relatively high frequency (inverse of motor size) is also emitted by direct radiation. The magnetic noise spectrum is composed of tones, the frequencies of which are multiples of the rotation speed. The highest frequency of these tones is equal to the product of the number of rotor notches by the rotation speed. The motor in operation may also produce tones that are multiples of the network frequency (50 Hz). This noise, which is minor compared to the aerodynamic noise and of the same order of magnitude as the mechanical noise, appears in the spectrum of the total noise when the fan's rotation speed is low (i.e. below about 1000 rpm).

Noticeable and annoying when the fan is kept at a low rotation speed, the mechanical noise has its origin in vibrations induced by various phenomena:

- The occurrence of an imbalance (unbalanced wheel, alignment around an imprecise axis of rotation...) produces vibrations at the rotation frequency.
- The bearings cause noise if they are not well maintained (bearings with worn balls or needles or insufficiently lubricated bearings).
- The links between the motor and the fan or the support structure cause solid-borne noise that entails radiation of the impeller and casing.

The aerodynamic noise is predominant compared to the two other sources during regular fan operation (at or near the best efficiency point). The control of the aerodynamic noise requires the study of the full spectrum. The noise of a fan is composed of tonal noise at the blade passing frequency ($F_{pp} = N_{blades} * V_{rot}$) and its harmonics, and broadband noise. In their work, Lighthill, Curle and then Ffowcs Williams-Hawkings divide the sources of aerodynamic noise into three categories (monopolar, dipolar and quadripolar):

- For fans, the monopole noise generates a noise associated with the fluctuations of flow due to the thickness of the blades crossing the flow. However, for the majority of fans, the monopole noise, or thickness noise, is negligible.
- The quadrupole noise comes from the fluctuating shear stress in the turbulent flow surrounding the blades. It becomes important for Mach number > 0.8, which never happens on fans.
- Ultimately, only the dipolar aerodynamic noise contributes in a dominant way to the tonal and the broadband spectrum of a fan in regular operation.

The main tonal noise generation mechanism is due to the non-uniformity of flow at the fan inlet or outlet due to flow obstacles or non-symmetrical ducts. The variation in absolute flow velocity at the inlet of the impeller triggers a variation of the relative speed on the blades and, therefore, of the forces applied to them. On the other hand, for obstacles located downstream (volute cut-off for example), a noise appears at the blade passing frequency when the wake of each blade interacts with the volute cut-off. The experiments showed that the amplitude of the tones increases as the upstream and downstream obstacles are closer together. Indications of the distance between obstacles and the fan were listed in order to simplify the design of the fan environment (for example in [47]).

The phenomena producing broadband noise are various. Fluctuations in the turbulent or laminar flow trigger random fluctuations in the forces acting on the blades and, therefore, the impeller radiates a broadband noise. The turbulent blade wakes also trigger the emission of broadband noise when they meet obstacles located downstream. Other phenomena contribute to the broadband noise:

- Noise due to ingestion of turbulence: the fluctuations of the turbulent flow velocity at the fan inlet lead to random fluctuations in the angle of attack, resulting in random fluctuations of the aerodynamic forces on the rotating blades and thus radiating a broadband noise;
- Trailing edge noise: this noise occurs when the turbulent boundary layer on the blade side (usually the suction side) is convected past the trailing edge. As a result, the

turbulent energy of the wall pressure fluctuations is converted into acoustic energy that radiates to the far-field;

- Vortex shedding noise: this noise source, like the previous one, is located at the blade trailing edge. It is created by the vortex shedding associated with von Karman vortex streets in the blade wakes when the blade trailing edge is thick as compared to the boundary layer thickness;
- Flow separation associated with rotating stall: this phenomenon may occur when the fan is operated at a very low flow rate.

Numerical approaches

The coupling of CFD calculations and calculations of sound generation and propagation is now offered by several computer programs or software packages. Using CFD calculations based on Navier-Stokes equations to determine the acoustic source terms and propagation of these terms by means of generalised Euler equations, the acoustic performance of fans starts to be predicted, but due to the complex phenomena, a lot of research is still needed to obtain reliable results, especially regarding the broadband noise generation.

Prediction of noise level and rules concerning similarity

Similarity

With the aim of characterising, classifying and extrapolating the noise levels measured at an operating point, the fans required rules regarding similarity to be established. Thus, a performance graph corresponding to one fan running at a given speed can be used to extrapolate graphs corresponding to other speeds or fan size.

Based on measurements taken on a fan of diameter D1 and rotating at frequency V1, the air performance of a geometrically similar fan with the characteristics {D2, V2} are determined based on the following formulas, named “fan laws”, while limiting excessive scaling effects (with Q volume flow rate, Δp pressure, PW fan air power and η efficiency):

$$Q_2 = Q_1 \cdot \frac{V_2}{V_1} \cdot \frac{D_2^3}{D_1^3} \quad \text{Eq. 2}$$

$$\Delta p_2 = \Delta p_1 \cdot \frac{V_2^2}{V_1^2} \cdot \frac{D_2^2}{D_1^2} \cdot \frac{\rho_2}{\rho_1} \quad \text{Eq. 3}$$

$$P_{W_2} = P_{W_1} \cdot \frac{V_2^3}{V_1^3} \cdot \frac{D_2^5}{D_1^5} \cdot \frac{\rho_2}{\rho_1} \quad \text{Eq. 4}$$

$$\eta_2 = \eta_1 \quad \text{Eq. 5}$$

These formulas only remain valid if the compressibility effects are negligible (Δp < 2500 Pa or Mach < 0.3). Next, based on measurements taken on a large panel of fans, it is possible to calculate the sound power level of fan 2 {D2, V2} from the level measured on fan 1 {D1, V1}using the following empirical equation:

$$L_{W_2} = L_{W_1} + 10.\log_{10}\frac{Q_2}{Q_1} + 20.\log_{10}\frac{\Delta p_2}{\Delta p_1} \qquad \text{Eq. 6}$$

Taking into account the aerodynamic similarity rules (Eq. 2 to Eq. 5), Eq. 6 may also be written:

$$L_{W_2} = L_{W_1} + 50.\log_{10}\frac{V_2}{V_1} + 70.\log_{10}\frac{D_2}{D_1} \qquad \text{Eq. 7}$$

However, if the ratio of rotation speeds $\mathbf{v_2/v_1}$ is not close to 1, sound transposition has to be applied not only in amplitude but also in frequency to account for the distortion of the spectrum with speed; in fact, a high speed fan radiates emits a higher frequency noise than a low speed fan. Therefore, AMCA (Air Movement and Control Association) proposes the following equation for applying the conversion from conditions 1 to 2 in octave or one-third octave bands of bandwidth BW.

$$L_{W_2}\left(f_2 = f_1 \cdot \frac{V_2}{V_1}\right) = L_{W_1}(f_1) + 40.\log_{10}\frac{V_2}{V_1} + 70.\log_{10}\frac{BW_2}{BW_1} \qquad \text{Eq. 8}$$

However, it should be noted that this formulation does not take into account sources whose frequency does not depend on the rotation speed, such as an acoustic resonance resulting from the cavity of a stator. More generally, these formulas are valid for fans without installation effects and remain approximations.

Prediction of noise level

Given the large number of types of fans, their dimensions, their ducts upstream and downstream, and their operating points, it appeared essential to assess the noise level of a fan at a given operating point based on reference blowers. The sound performance of these reference fans were obtained from comprehensive series of tests performed on a large number of fans. These references take account of the type of fan (centrifugal or axial of different geometry), the impeller diameter as well as the range of operating pressures. A method is proposed by ASHRAE (American Society of Heating, Refrigerating and Air conditioning Engineers) based on Eq. 6:

$$L_w(f) = K_w(f) + 10.\log_{10}\frac{Q}{Q_{ref}} + 20.\log_{10}\frac{\Delta p_T}{\Delta p_{ref}} \qquad \text{Eq. 9}$$

Where (in SI units):

- Lw(f): fan sound power level in octave band radiated by inlet + outlet + casing (dB)
- Kw(f): specific sound power level in octave band (dB) (see Table below)
- BPF: value in dB to add to Kw in the BPF octave band
- Q: fan flow rate
- Δp_T: fan total pressure
- Δp_{ref} = 1 kPa
- Q_{ref} = 1 m^3/s

Specific Sound Power Levels (K_w) (Graham 1992, 293-300)

Fan Type	Diameter	Octave Band Center Frequency (Hz) 63	125	250	500	1k	2k	4k	8k	BPF
Centrifugal Fans										
Backward-curved blades	> 0.75 m	85	85	84	79	75	68	64	62	3
	< 0.75 m	90	90	88	84	79	73	69	64	3
Radial blades: Δp = 1 - 2.5 kPa	> 1m	101	92	88	84	82	77	74	71	7
	< 1m	112	104	98	88	87	84	79	83	7
Radial blades: Δp = 2.5 - 5 kPa	> 1m	103	99	90	87	83	78	74	71	8
	< 1m	113	108	96	93	91	86	82	79	8
Radial blades: Δp = 5 - 15 kPa	> 1m	106	103	98	93	91	89	86	83	8
	< 1m	116	112	104	99	99	97	94	91	8
Forward-curved blades	All	98	98	88	81	81	76	71	66	2
Axial Fans										
Vaneaxial										
Hub ratio = 0.3 to 0.4	All	94	88	88	93	92	90	83	79	6
Hub ratio = 0.4 to 0.6	All	94	88	91	88	86	81	75	73	6
Hub ratio = 0.6 to 0.8	All	98	97	96	96	94	92	88	85	6
Tubeaxial	> 1m	96	91	92	94	92	91	84	82	7
	< 1m	93	92	94	98	97	96	88	85	7
Propeller	All	93	96	103	101	100	97	91	87	5

Figure 2-9: Reference sound power levels [6]

Other methods

A prediction method based on the concept of acoustic efficiency was proposed:

$$\eta_{acou} = \frac{W_t}{P} = 10^{-4}.\,Re^{\alpha}.\,M^{\gamma-3} \qquad \text{Eq. 10}$$

The values of α and γ depend on the type of fan, the diameter, and the rotation speed. All the coefficient values are deduced from a series of tests. These prediction methods only give an approximate indication of the global noise level. As soon as the frequency resolution is increased, the precision is in fact highly degraded. The best way to estimate and predict the power level remains to maintain the same method of experimental determination. The following section, therefore, looks at standardized methods aiming to measure the noise emitted by a fan.

Measurement of noise

Since it is necessary to determine the sound power level of a fan for different operating conditions, the implementation of standards for determining the spectrum of a fan is essential. These methods, presented in the following paragraphs, are used to assess the noise spectrum with a specific prerequisite. Standard ISO 13347 describes the first three (reverberation chamber, enveloping surface, sound intensity method), whereas the last two have dedicated standards (ISO 5136: in-duct measurement; ISO 10302: mylar plenum). These methods account for the four standardized categories of fan installation A (non-ducted at inlet and outlet), B (ducted at outlet only), C (ducted at inlet only), and D (ducted at inlet and outlet). Of course, ISO 5136 is not relevant for testing a fan according to installation category A, since there is no duct.

Reverberation chamber

Before any measurement, it is necessary to have access to a reverberant room corresponding to the standard defined by AMCA. Determination of the acoustic power of the fan is implemented by comparing the measurement of the blower with that of a reference source. After having measured the one-third octave pressure spectra at different locations in the chamber with the fan, using the reference source in operation, the power level is then determined with the following formula:

$$L_{W_V}(f) = \overline{L_{P_V}}(f) + L_{W_{ref}}(f) - \overline{L_{P_{ref}}}(f) \qquad \text{Eq. 11}$$

Where (SI units):

- $\mathbf{L_{W_V}}$: fan power level
- $\overline{\mathbf{L_{P_K}}}$: averaged sound pressure level in chamber (k for fan or reference source)
- $\mathbf{L_{W_{ref}}}$: power level of reference source obtained outside the chamber

In order to control the operating point of the fan, the flow rate and the fan pressure must be measured in accordance with standard ISO 5801.

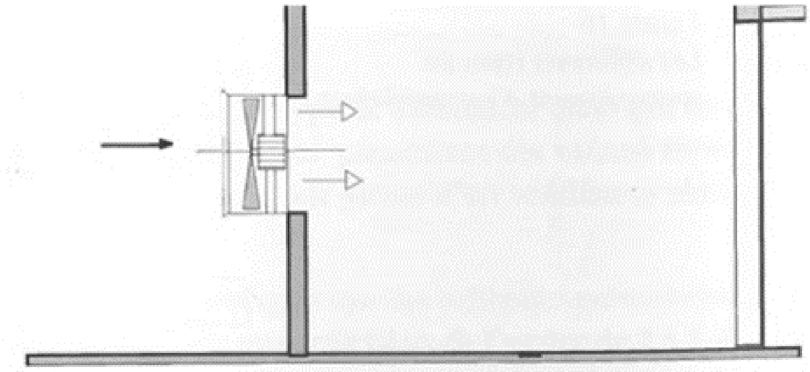


Figure 2-10: Reverberation chamber method.

When measuring the pressure levels emitted by the fan assembly for the same installation conditions as with the reverberation chamber method, including a reflecting floor, the sound power level in octave or one-third octave band can be calculated using the following formula:

$$L_W = \overline{L_P} + 10\log_{10} S - K_1 - K_2 \qquad \text{Eq. 12}$$

Where (SI units):

- $\overline{\mathbf{L_P}}$: mean acoustic pressure measured on the enveloping surface
- S : area of the enveloping surface
- K_1 : background noise correction
- K_2 : environmental correction

By using microphones at locations on the enveloping surface defined in ISO 13347-3, the sound power levels of the fan are determined. However, the determination of the correction coefficients requires a control of the measuring site. Correction K_1 is calculated by performing an acoustic measurement of the fan both in operation and switched off. K_1 represents the difference in pressure levels measured between background noise and source noise. If K_1 is too low (commonly an emergence lower than 6 dB), the background noise must be reduced before continuing the test. Correction K_2 is obtained by using a reference source placed at a location specified in the standard. If the absolute value of K_2 is lower or equal to 2, the measuring site is validated. These corrections K_1 and K_2 therefore require control of the measuring site before a test series can be carried out, as is the case with the reverberation chamber method.

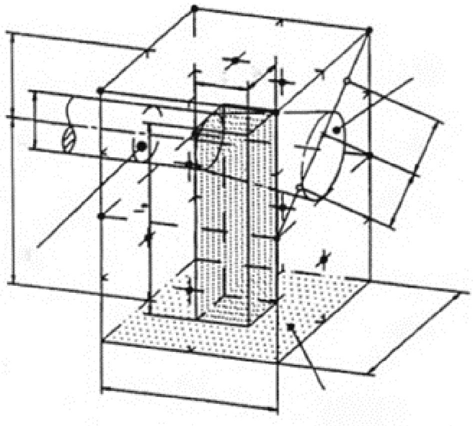


Figure 2-11: Enveloping surface method.

<u>Sound intensity method</u>

The sound intensity method is used to obtain the power radiated by the fan under the same conditions as the two previous methods. Requiring no specific measuring environment and validated by standards, the method must nevertheless be conducted by qualified staff who know how to handle an intensity probe and the appropriate post-processing operations. The measuring surface can be a hemisphere or a parallelepiped on which the measurement is undertaken by scanning or using specific fixed locations (ISO 9614-1 and ISO 9614-2).

<u>In-duct method</u>

The in-duct method is used to determine the acoustic power of the fan based on three pressure measurements (the three microphones are in a same section of the duct specified in ISO 5136). The duct end opposite the fan is fitted with an anechoic termination to avoid stationary waves due to the interference between the direct waves radiated by the fan and the reflected waves at the duct end. With control of the operating point of the fan, the air and acoustic performances are measured at the same time. The formulation of the sound power spectrum based on the sound pressure measurements is obtained according to the following formula:

$$L_W = \overline{L_P} + 10 \log_{10} S + C \qquad \text{Eq. 13}$$

Where (SI units):

- S: area of the duct cross-section
- C : standard corrective term depending on frequency, flow velocity and microphone shield.

Mylar plenum

Perfectly suited to the case of small-diameter fans with reduced performances ($Q \leq 1$ m^3/s-1, $\Delta p_T \leq 750$ Pa), the method consists in connecting the fan to a cubic plenum with walls made of mylar (plastic compound) sheets that are transparent to sound waves. Therefore, the acoustic pressure measurements are used to determine the power level radiated by the fan inlet and outlet. By controlling the operating point of the fan using static pressure taps within the plenum, it is possible to measure the power level radiated at any point of the fan curve.

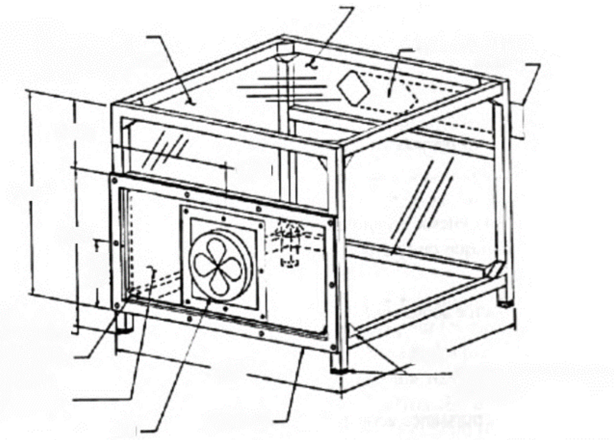


Figure 2-12: Sketch of a mylar plenum according to standard ISO 10302.

2.2. The compressor

The compressor is used to transform pressure (in bars) to a fluid element between an entry point and an exit point. The different parts are:

- Tubes: two tubes interact with the compressor. The first, the intake tube, transmits fluid to the compression system, whereas the exhaust or discharge tube evacuates the compressed fluid to the outside.
- Fluid (generic term), coolant (refrigeration unit): fluid contained in the intake and exhaust tubes and in the dome of the compressor. This fluid undergoes variations in pressure during its passage in the compressor. In refrigeration compressors, the fluid is used to cool the heat exchangers and condensers, providing the function of thermal regulation of the machine.
- Dome: this represents the upper part of the shell of the compressor. It is this zone in which the fluid is projected outside the compression system to the exhaust tube.
- Feet: they are located at the bottom of the compressor and they are used to attach the compressor to a support structure via pads.

There are many compressors for varied wide range of applications. However, there are different classes:

- Reciprocating or piston: in these compressors, each piston makes an alternating movement within a cylinder. Drawn in by one movement of the piston, the fluid is compressed on the return movement of the piston equipped with a check valve. The exhaust valve offering a defined resistance allows the expulsion of the compressed fluid. As the piston descends, it creates negative pressure, opening the check valve and taking in the fluid. Then, the piston goes back and little by little compresses the fluid until the pressure is high enough to open the exhaust valve to release the fluid. Often equipped with several pistons with offset intake and exhaust phases, compressors ,therefore, provide a more constant output of fluid.

- Rotary: compressor with pressure variation provided by a rotating element, as opposed to the translational movement in piston compressors.
- Vanes: a vane compressor is composed of a cylindrical stator within which an eccentric rotor turns. The latter is equipped with radial grooves in which vanes slide, which are pushed against the wall of the stator by centrifugal force, so that the fluid can be sucked in through the intake pipe and transported to the exhaust tube, the area in which the fluid volume decreases.
- Scroll: this type of compressor requires two intercalated spirals like vanes in order to pump and compress the fluid. One of the coils is fixed, whereas the other turns eccentrically around the first in order to pump, hold and expel the fluid in the exhaust tube.
- Lobe: often called a roots compressor, this is a mechanical system comprising two lobes that hold the fluid during rotation. The compression rate is determined on the basis of the volume held and the ratio between the speed of rotation of the lobes and that of the motor.
- Type G: this compressor consists of two fixed spirals and two moving spirals. Driven by the pulley of a camshaft, the mechanism is used to pump the fluid and push it towards the centre of the compressor close to the outlet. Due to the presence of several pairs of spirals, there is no noticeable drop in pressure between each arrival of the fluid at the intake.
- Screw: this type of compressor consists of two counter-rotating synchronised screws that compress the fluid. A reduction in the volume of space available for the fluid leads to an increase pressure. The ambient fluid is drawn in along the axis of the screws, in the area where the screw indentation is at maximum, whereas discharge is performed after an increasingly narrow path between the screws. An oil film, cooled to increase viscosity, ensures the compressor seal. To go through the oil film, the screws cannot get in contact with each other and allow oil-free fluid discharge.

<u>Several parameter influence the vibro-acoustic behaviour of compressors:</u>

- Drive frequency: in the case of a variable speed compressor, the command frequency corresponds to the number of iterations made by the compression element during 1 second (rps). Changing the frequency changes the motor operating speed. The harmonics of this drive frequency will appear in the vibro-acoustic spectrum of the compressor.
- Flow rate: linked to the compression/discharge phenomenon, the flow rate is a type of periodic excitation (corresponding to the speed of rotation in the case of a rotary compressor) which represents the phenomenon of expulsion of fluid towards the exhaust tube. The value of these pressure pulsations is expressed as flow rate or as the normal speed at the compression system outlet section.
- Torque: linked to the rotation of the motor and the friction induced, this is a secondary contribution to the vibration and sound levels produced by the compressor.
- Pressure pulsation: the fluid ejected into the exhaust tube and circulating in the intake tube interferes harmonically (linked to drive frequency) with the tubes that radiate and vibrate as well as the compressor dome by repercussion.

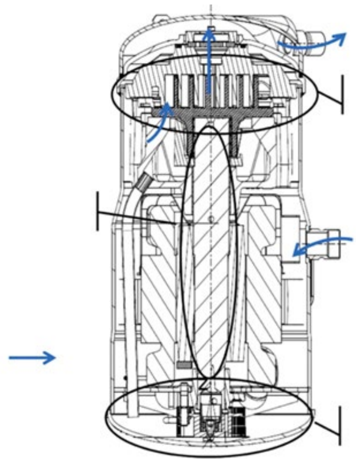


Figure 2-13: View of a scroll compressor.

Compressor as Source of noise

For any type of compressor, the noise consists of a spectrum of lines and a broadband spectrum, as is the case with the majority of machines that have a motor element in regular movement (translation/rotation).

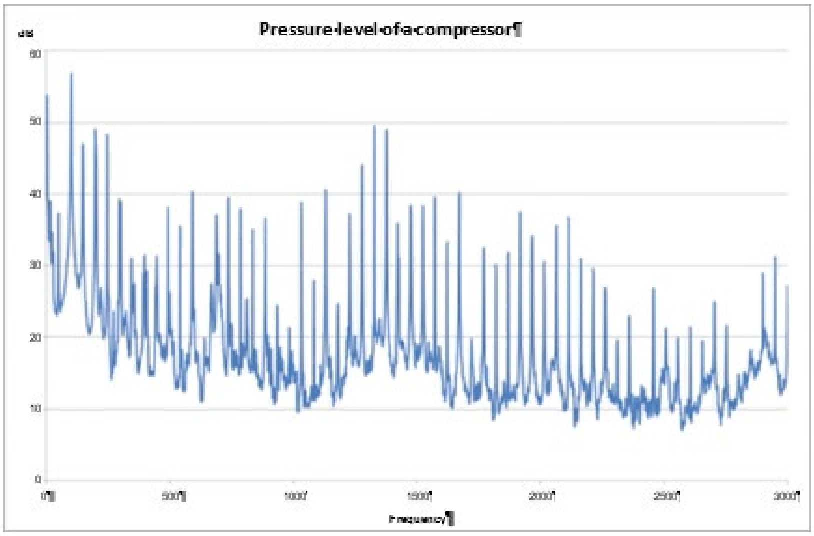


Figure 2-14: Typical spectrum of SCROLL compressor noise.

The vibro-acoustic of refrigeration compressors comprises several phenomena playing major roles, the origin of which is linked either to the dynamic behaviour or internal or external acoustics. These two sources have an impact on the immobile elements connected to the compressor: support structures and cooling pipes.

Dynamic behaviour is the result of internal forces. The main forces causing excitations are linked to inertial movements of the compression elements such as the piston or vane. The vibrations of the motor unit are propagated on the shell of the compressor and through the intake and exhaust pipes. The vibratory behaviour comes from two different aspects of motor behaviour: the first is linked to the movement of the motor itself and can therefore be represented by a torque or a force in translation, according to the operating mode of the compressor. The second phenomenon is linked to the projection of coolant against the upper part of the dome located downstream of the compression chamber. The fluid thrown in this way directly impacts the dome before it is directed into the exhaust tube. This phenomenon is closely linked to the compressor drive frequency and the variations in the flow rate from the compression chamber. Very often, a sliding phenomenon appears on the harmonics of the drive frequency due to the occurrence of resistive phenomena corresponding to the friction of the blades and the internal slowing of the compressor parts.

The internal acoustics are linked directly to the behaviour of the compression mechanism. The fluctuations in coolant pressure are propagated in the piping circuit connected to the compressor. The various organs of propagation of the fluid can weaken or amplify the pressure pulsations. Excitations of this type deform and scratch the cooling pipes. Within the frequency domain, the vibro-acoustic phenomena can be divided into three categories:

- In the low-frequency domain, due to the primary harmonics of the excitations caused by fluid compression, the compressor behaves like a resonator with a dipolar acoustic behaviour.
- In the mid-frequency domain, taking account of higher excitation harmonics, the acoustic radiation is obtained using a modal synthesis.
- In the high-frequency domain, an energy-based approach is used to quantify the acoustic contribution of the circulation of fluid in the cooling pipes.

Vibro-acoustic behaviour

In order to determine the vibration behaviour of the compressor alone, it is essential not to have to consider compressor mounting elements and cooling pipes. An initial observation of acoustic and vibration measurements taken on a single compressor excited with a shock hammer allows for recording the presence of two types of modes:

- feet modes, clearly visible in the vibratory response;
- dome modes, visible in the acoustic measurements

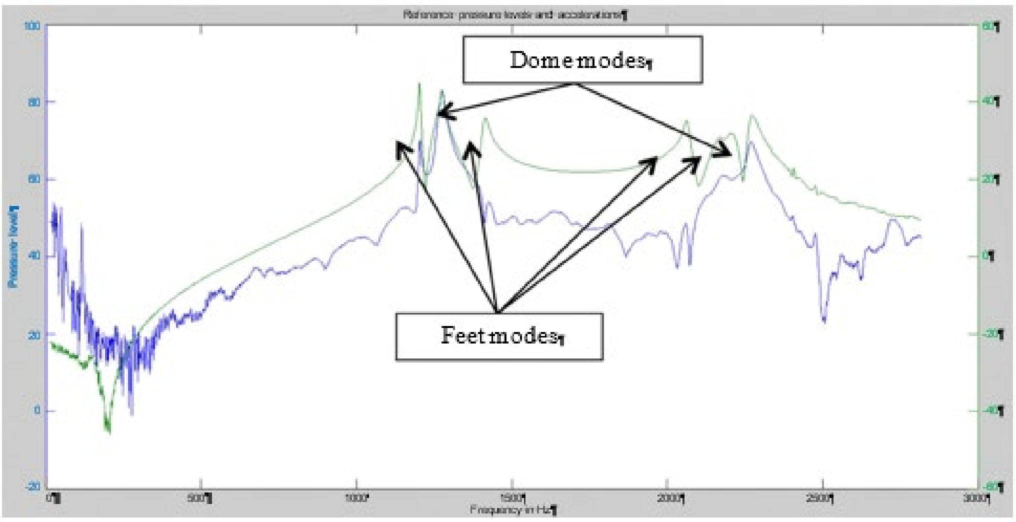


Figure 2-15: Acoustic (blue) and vibration (green) spectra of a compressor.

These two types of modes appear in the medium frequencies and then regularly over the range of frequencies. Often, the first modes are the feet modes (strong vibratory response), which are then joined by the dome modes at quite a high frequency but which are mostly responsible for the pressure level. The compressor is currently modelled according to Figure 2-16 below, considering the compressor dome as a plate that only radiates along the external normal. The internal acoustic cavity is not taken into account, which is justifiable considering the space inside the compressor.

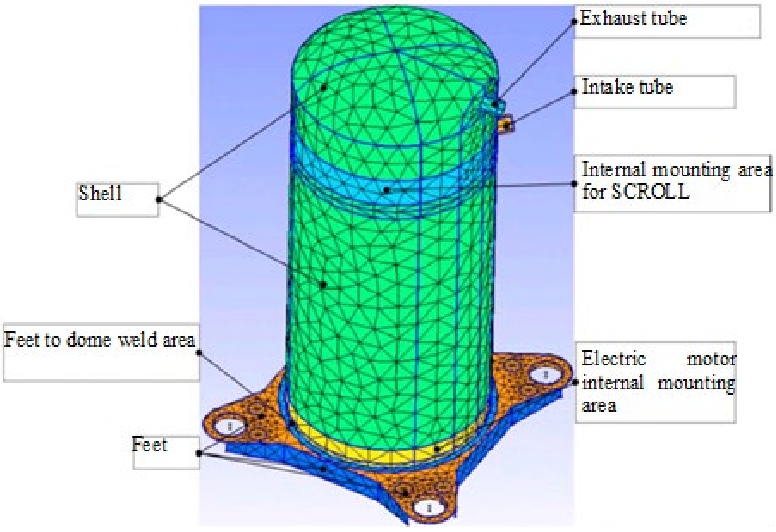


Figure 2-16: Modelling of a scroll compressor.

Inserting the compressor in its surrounding

Firstly, the compressor is attached to the structure by pads that will transform its vibratory behaviour into excitation forces on the structure, taking on vibro-acoustic behaviour. Without considering the process for determining the radiation of the receiving structure, it is essential to take account of the pads as elements of connection between the compressor feet and the support structure. Several mobility methods can be used to determine the force transmitted by the compressor via the pads to the structure. First, they need to characterise the mobilities of the receiving structure, the source, i.e. the compressor and the pads (Figure 2-17).

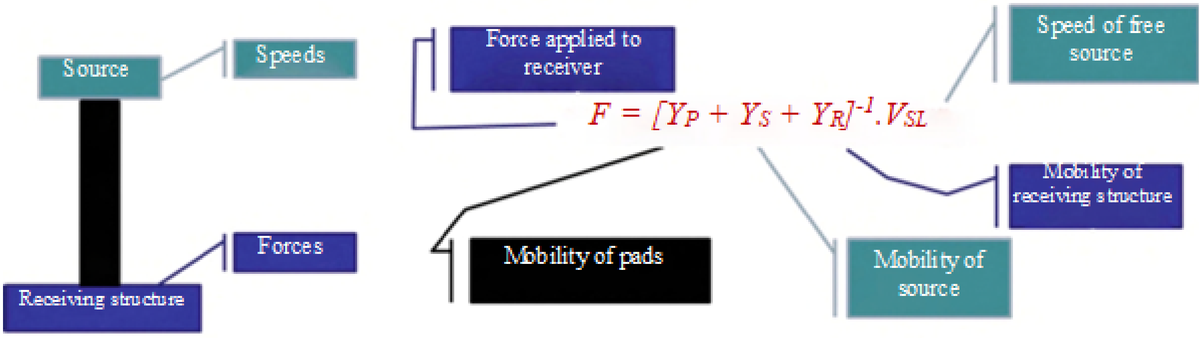


Figure 2-17: Insertion of the compression : Method to obtain the dynamic forces applied on the receiving structure.

The cooling pipes are excited by pressure pulsations of the coolant, radiate and thus contribute to the total noise level. In addition, it is apparent that the geometry of the pipes as well as the mounting conditions affect the result because the pipes behave like guitar strings that will excite the acoustic cavity formed by the compressor dome. The strong link between the pipes and the dome is still difficult to parameterize because it introduces coolant and changes the vibro-acoustic behaviour of the compressor in the intake and exhaust zones.

Numerical prediction

The noise level calculation is carried out in three stages:

- Firstly, the excitations are calculated in relation to the characteristics of the compressor (electrical power supply, drive frequency, dimensions, inertia, and masses of elements in movement,...). For now, two types of excitations are considered: the flow rate of the fluid at the compression chamber outlet and the torque applied at the inertial centre of the element in movement (shaft for a rotation, for example).
- Next, a calculation of the vibro-acoustic response is carried out, taking account of the two unitary excitations. This calculation is used to quantify the acoustic radiation of the compressor dome and cooling pipes, taking the pads into account.
- To finish, the unitary response is multiplied by the excitation spectrum to obtain the acoustic behaviour of the compressor.

As the compressor dome has a controlled vibro-acoustic behaviour, the main obstacle to this modelling remains the quantification of the excitations and the determination of their spectra.

Measurement of noise emissions

The measurement of airborne noise radiated by a compressor can be carried out using conventional methods: reverberation chamber, enveloping surface, and acoustic intensity probe (see fan noise measurement).

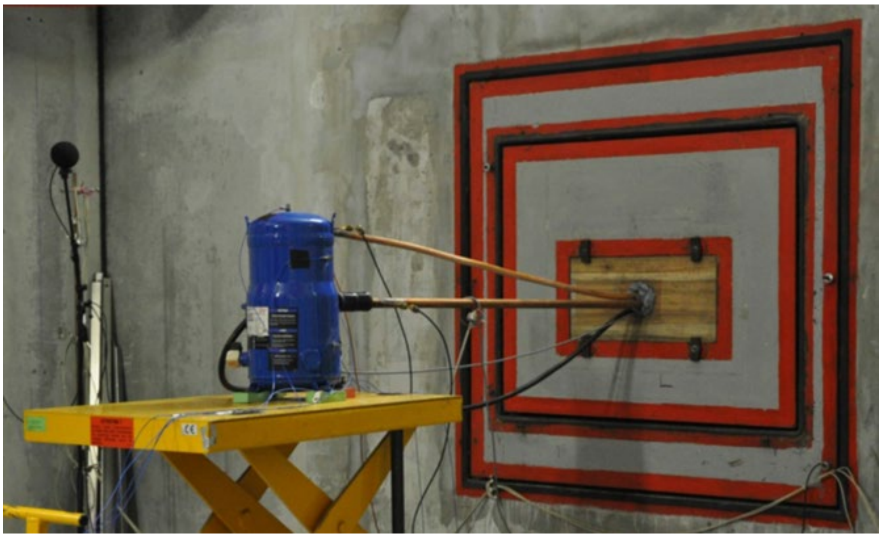


Figure 2-18: Typical set-up for measurement of airborne radiation of a compressor.

Another approach has been developed to calculate radiated noise based on vibration measurements, which presents several benefits:

- Reduced cost of determination: a few accelerometers are sufficient (6 or 10) (Figure 2-19);
- an operating procedure that takes account of the vibratory influence of the pipes on the compressor;

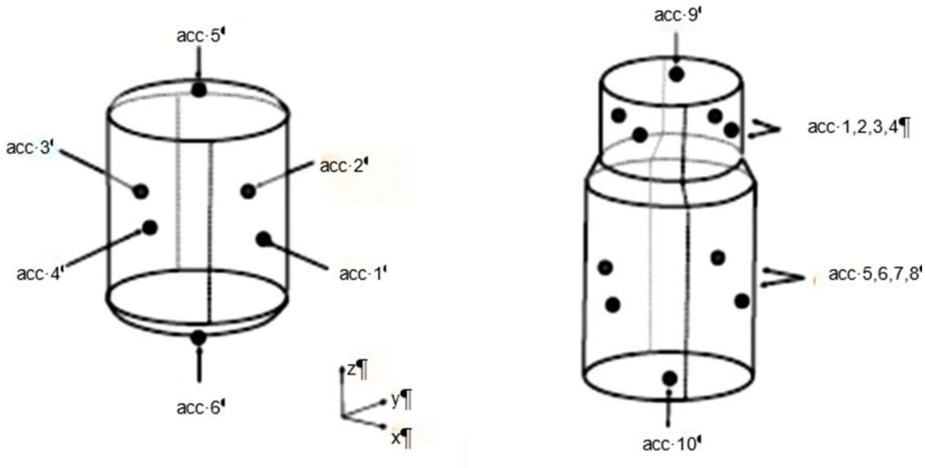


Figure 2-19: Arrangement of six sensors for a compressor and ten sensors for a compact compressor to evaluate airborne radiation from acceleration measurements.

The acoustic determination based on vibration measurements is undertaken analytically by considering two methods:

Panel methods considering the sum of inconsistent acoustic sources:

- In the low-frequency domain, the total acoustic power is calculated by adding the acoustic power radiated by each panel corresponding to a rigid part of the compressor dome.
- In the high-frequency domain, the calculation of the radiated acoustic power is based on the use of Maidanik's formula for the radiation coefficient of the panels considered flexible.
- The experimental data shows that the width of the transitional area is about one octave. The transitional area is defined empirically by connecting the two areas linearly in a decibel scale.

The source superposition method:

- Acoustic power is calculated by viewing the sound field as similar to that produced by four individual acoustic sources, one of which is monopolar and three dipolar oriented on each of its axes. A dome expansion movement (monopolar) and a global balancing movement of the rigid body type (one dipole per direction of movement) are translated by these individual sources).
- Initially, the amplitude of the sources is defined by recalibrating the field of acoustic accelerations produced by the sources for the acceleration values measured on the walls of the machine. Then the acoustic power radiated by the individual sources is calculated.
- The acoustic sources are considered incoherent: the total acoustic power is calculated by adding together the acoustic powers radiated by each source.

2.3. Secondary sources

The evaporator in an air-source heat pump is the component where heat is exchanged from the outside air to a refrigerant in a pipe system. The most common exchanger type for modern residential heat pumps is configured with a round tube and continuous fins. The fins are connected to the tubing to increase the heat transfer area and can be configured as plain or wavy fins, with wavy fins further increasing the heat transfer area (Figure 2-20).

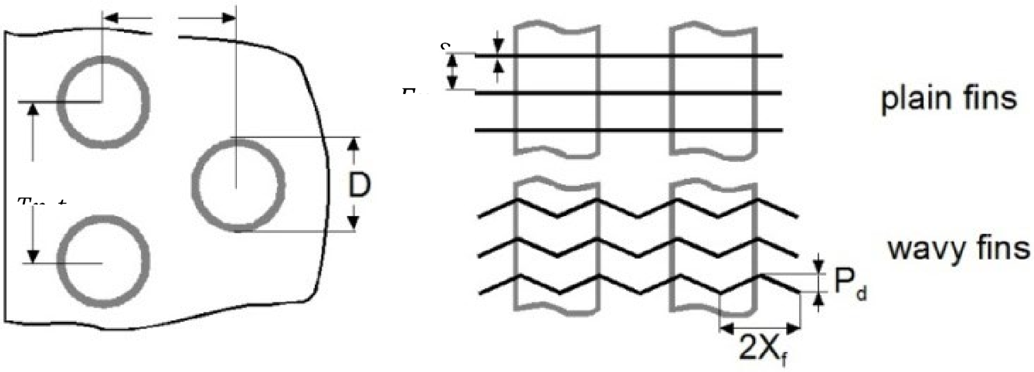


Figure 2-20: Sketch of a round tube heat exchanger with plain and wavy fins.

Other configurations are, of course, possible, and one example from other applications is to use flat tubes. In a study by Gustafsson et al., a round tube heat exchanger was compared with two types of flat tube heat exchangers. In Figure 2-21 the round tube heat exchanger is denoted B5, and the flat tube heat exchangers are denoted FFC and MPET. The sound power level of the flat tube heat exchangers is lower for the tested configurations. The "direct" effect of the heat exchanger sound radiation on the overall sound power level is very small or negligible. In the normal air velocity range, the sound radiation originating from the heat exchangers is more than 15 dB below the sound power level of a commercially available heat pump.

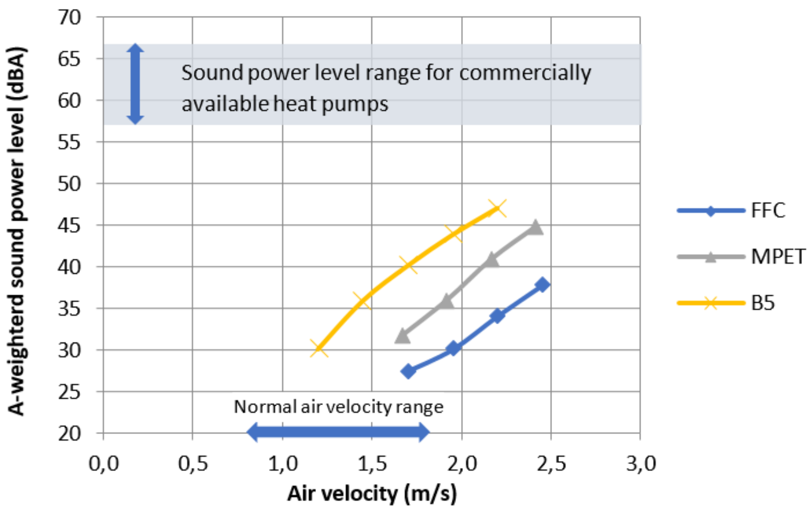


Figure 2-21: Measured sound power level of three heat exchangers compared to commercially available heat pumps. B5 – round tube, FFC and MPET – flat tube.

<u>The results from these measurements</u>

The choice of heat exchanger type and design will have a bigger influence on the sound radiated by the fan. Since the fan is a dominant noise source in an air source heat pump and the noise level is dependent on its state of operation, the heat exchanger will have an "indirect" effect on the overall sound power level of a heat pump. A possible way to reduce the overall sound power level of a heat pump is to optimize the heat transfer ability for a lower air flow and hence reduce the fan sound radiation. Estimation of the radiated sound power level of the fan according to a method proposed by ASHRAE show a dependence on these factors, Eq. 14:

$$L_W = K_W + 10 \log \dot{V} + 20 \log P \qquad \text{Eq. 14}$$

From empirical correlations the air flow rate $\dot{\mathbf{V}}$ and pressure **P** can be deduced to evaluate the possible effect of variations in design parameters such as frontal area and depth (number of tube rows). As eq. 1 indicates, the fan sound power level can be reduced, if a reduction of the airflow rate and pressure drop can be achieved. However, it is necessary to maintain a high heat transfer ability to achieve a high a heat transfer rate. An increased frontal area results in more heat transfer surface and less forced convection (lower airflow rate). Adding more tube rows would also give more heat transfer area, but will increase the air-side pressure drop. The total heat exchanger volume is, of course, important, as this is constrained by the space limitation in the heat pump. In Figure 2-22, the air-side heat transfer coefficient (α_{air}) is shown as a function of air velocity. The alpha value is normalized with the total volume of each heat exchanger from Figure 2-21. Two measured values of the FFC and MPET are shown as points, which showed an over- and underestimation of the correlations of approximately 10%. The correlation of the B5 heat exchanger is performed with two different methods, which also show quite a large uncertainty. However, the flat tube heat exchanger MPET seems to have a lower capability for heat transfer in a heat pump application.

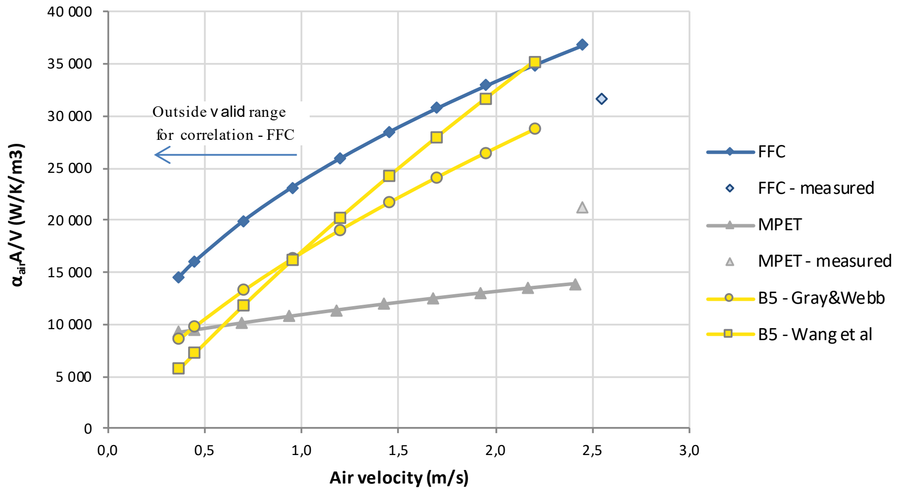


Figure 2-22: Air-side heat transfer coefficient versus air velocity. The alpha value is normalized with the total volume of the heat exchanger. Two measured values of the FFC and MPET heat exchanger are shown as points.

2.4. Heat exchangers in interaction with fan

As discussed in the previous sections, both the heat exchanger and fans contribute to the total acoustic behaviour of a heat pump. Hence, a relevant topic for heat pump manufacturers is the interaction between the fan and the heat exchanger and how they influence the acoustic behaviour of each other. Several parameters, such as the distance between the components or the angle of the heat exchanger, can have an influence on the fluid flow and, further, on the aeroacoustic behaviour in the heat pump case. When the air enters the outdoor unit, the air immediately flows through the heat exchanger. The fins in combination with the refrigerant pipes or channels cause a turbulent structure in the flow field behind the heat exchanger (see Figure 2-23). These eddies produce size-dependent pressure fluctuations which generate noise. As already mentioned in the previous chapter, the non-uniformity of the flow at the fan inlet due to turbulence is one of the main noise generation mechanisms in the flow domain. One reason for this is the impingement of the random eddies on the moving fan blades which generates pressure pulses.

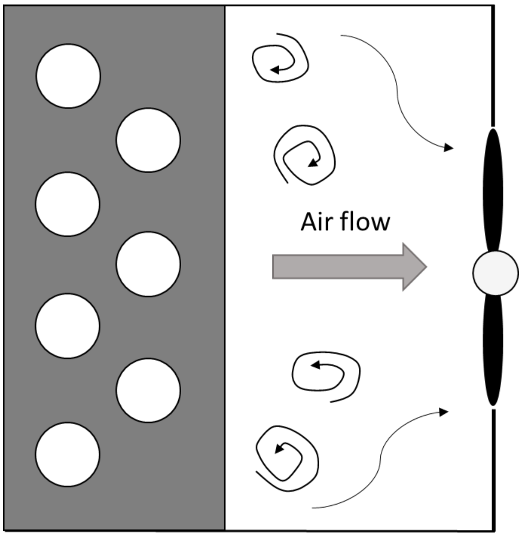


Figure 2-23: Simplified flow field in a flow domain between the heat exchanger and the fan.

Due to the pressure loss through the HEX, fans often do not operate at their most efficient point. Hence, the real operating conditions do not force the fan to operate at an acoustically optimised operating point. In general, in today’s domestic or outdoor units, multiport extruded tubes with serpentine flat fins are installed. Latest investigations deal with the application of new heat exchanger models, such as microchannel heat exchangers (MCHE, Figure 2-24). The MCHE are still not widely used for mass production in heat pumps because the defrosting issue is not easy to handle [7]. Introducing new heat exchanger geometries leads to a different turbulent flow field in the trailing area. That is why the acoustic field might also be changed.

a) b)

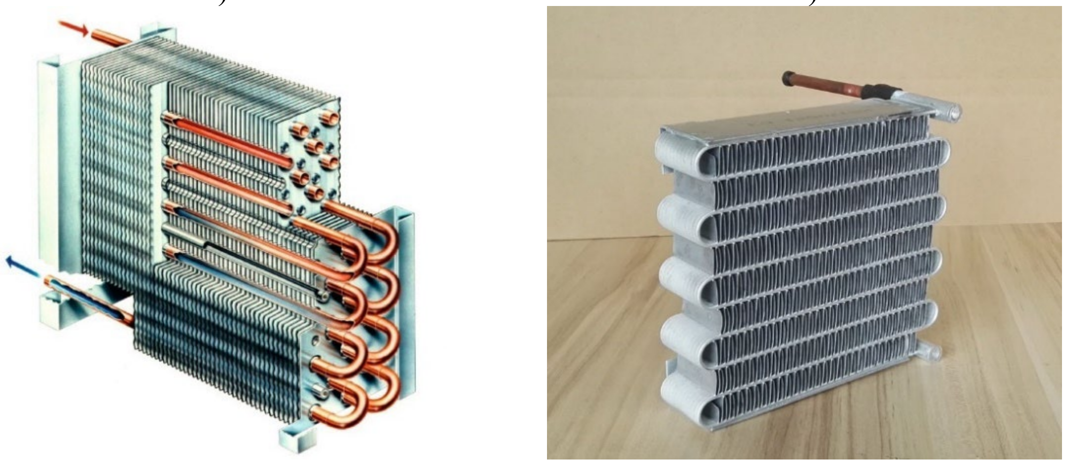


Figure 2-24: a) Multiport extruded tubes with serpentine flat fins; b) Microchannel heat exchanger.

The MCHE are still not used for mass production in heat pumps widely because the defrosting issue is not easy to handle [7]. As already mentioned, the non-uniformity of the flow at the fan inlet is one of the main noise generation mechanisms. In many heat pumps, electronic devices, cooling components, etc., disturb the flow field (Figure 2-25). Also, parts of the case structure elements or the fan motor case influence the flow field.

Figure 2-25: Flow field in a heat pump outdoor unit.

Hence, it is all the more important to investigate if their positions in the flow domain negatively influence the acoustic behaviour and further optimize the flow domain to improve the flow field in front of the fan inlet.

3. Fundamentals of Noise Control Techniques

3.1. Airborne noise

The term “airborn” noise refers to the sound that is transmitted through the air. A significant part of heat pump noise is emitted to the outside via openings, e.g., the air inlet on the suction side. In this section, various passive control techniques are presented to reduce the transmission and emission of airborne noise. At the end of the section, general aspects of active noise control techniques are addressed. In practice, there are three main noise control techniques to reduce sound emission: sound absorption, sound insulation and sound attenuation.

Figure 3-1 illustrates the principles of these three techniques, showing where and how they may be applied in machine acoustics. For the reduction of the overall sound emission, all relevant sound transmission paths need to be taken into account. Weak points in the sound insulation or inlet-outlet paths result in sound transmission paths which can dominate the overall sound emission.

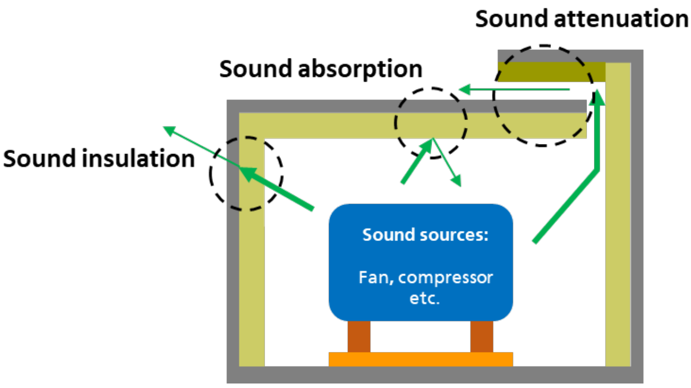


Figure 3-1: Methods to influence the sound emission of a noise source, e.g. heat pump or motor.

Sound absorption

Absorbers, either in the form of porous material or so-called Helmholtz resonators, withdraw acoustic energy from the incoming sound and reflect sound waves with less energy. When they are placed inside a machine casing, they reduce the sound pressure levels inside the casing. If the sound pressure level inside the casing is reduced, also less sound is transmitted to the outside.

Sound insulation

Sound insulation deals with the sound transmission from the inside of a casing to the outside via the closed casing surfaces. In contrast to sound absorbers, where the air molecules themselves can propagate into the absorber, acoustically sealed surfaces prevent the propagation of the air molecules. However, on the source side, the air molecules transfer a part of their momentum to the structure, which starts to vibrate. The vibration amplitudes mainly depend on the mass of the casing walls and other aspects of their structural dynamics, like bending stiffness and internal structural damping. On the receiving side, the vibrating surfaces of the casing radiate the sound to the outside. The efficiency with which the sound is radiated from an vibrating surface is called acoustic radiation efficiency. The acoustic radiation efficiency is frequency-dependent and mainly depends on how well the bending wavelength of the structure matches the acoustic wavelength. In particular, light and rigid casing structures exhibit a relatively high radiation efficiency.

Sound attenuation

Sound attenuation happens mostly in inlet or exhaust channels. The working principle is close to the one in sound absorbers. Sound attenuation also withdraws energy from the sound, but in this case the sound passes the absorbers laterally.

3.1.1. Sound Absorption

The main task of sound absorbers is the dissipation of acoustic energy of propagating sound waves. Porous absorbers made from fibre material or open cell foams are commonly used to reduce reverberation times and sound pressure levels in rooms and noise insulating casings. Porous absorbers combine good acoustic absorption properties with low weight and low costs. Figure 3-2 shows typical porous absorber materials used in noise control applications.

a) b) c)

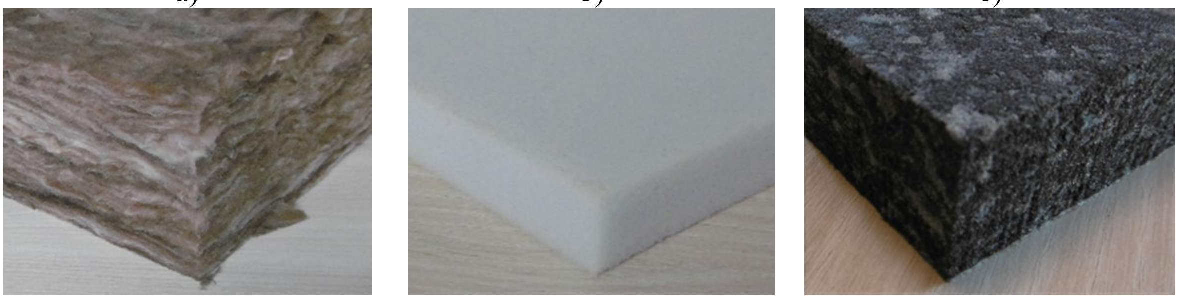


Figure 3-2: Typical absorber materials: a) mineral wool; b) melamine foam; c) polyurethane foam.

Figure 3-3 shows the transmission paths of the power of an incident sound wave through an absorber.

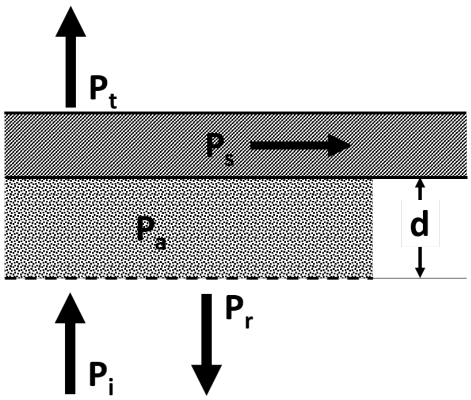


Figure 3-3: Transmission paths of the power of an incident sound wave through an absorbing obstacle.

The incident soundwave with the sound power (P_i) is partially reflected (P_r), absorbed (P_a), converted into structure-borne noise (P_s) or transmitted (P_t). It follows

$$P_i = P_r + P_t + P_s + P_a \,. \qquad \text{Eq. 15}$$

The reflection coefficient **ϱ** is given by

$$\varrho = \frac{P_r}{P_i} = \frac{p_r^2}{p_i^2}, \qquad \text{Eq. 16}$$

the ratio of the incident and reflected sound power or sound pressure amplitudes of the incident and reflected sound waves.

The absorption coefficient **α** is given by

$$\alpha = 1 - \varrho = \frac{P_a + P_t + P_s}{P_i}, \qquad \text{Eq. 17}$$

and characterises the efficiency of the absorber.

The sound absorption of materials can be experimentally characterised in an acoustic impedance tube (Figure 3-4). On one end of the tube with ridged walls, a loudspeaker produces an acoustic wave which travels through the tube. At the other end, an absorption material test sample is mounted in front of a rigid termination. The wave enters the absorption material and is reflected from the rigid termination behind it. Without the test sample, nearly 100 % of the acoustic wave would be reflected by the rigid termination. Hence, the incident and the reflected wave would have the same amplitude. When the test sample absorbs sound energy, the incident and the reflected waves have different amplitudes. This difference in the pressure amplitudes is measured using microphone probes inside the tube and is used to calculate the reflection and absorption coefficients [8].

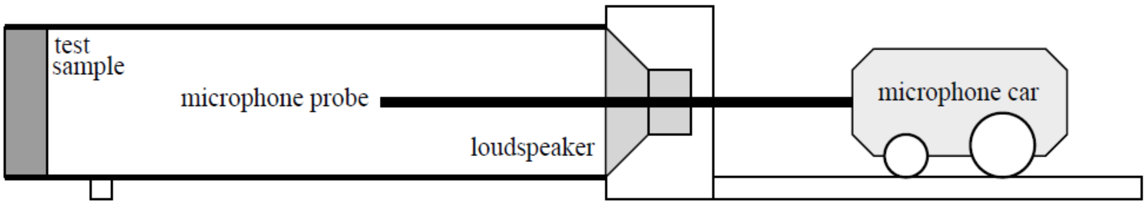


Figure 3-4: Standing wave apparatus [8].

The absorption coefficient depends on the porosity, the size of the pores, and the flow resistance, which need to be matched. The thickness of the absorption layer also significantly influences the absorption performance of a sample. Figure 3-5 shows representative sound absorption coefficients of a typical absorber material, depending on the thickness of the layer. The graphs show that the absorption coefficients generally increase with higher frequency. However, in order to achieve a specific absorption coefficient at a certain target frequency, a minimum layer thickness is required. For example, in order to achieve an absorption of about 80 % at 1000 Hz, a layer thickness of 40 mm is required. To achieve the same performance at 500 Hz, a layer thickness of 80 mm required. Depending on the application and target frequencies, very thick absorption layers may be required, which may conflict with constraints regarding installation space.

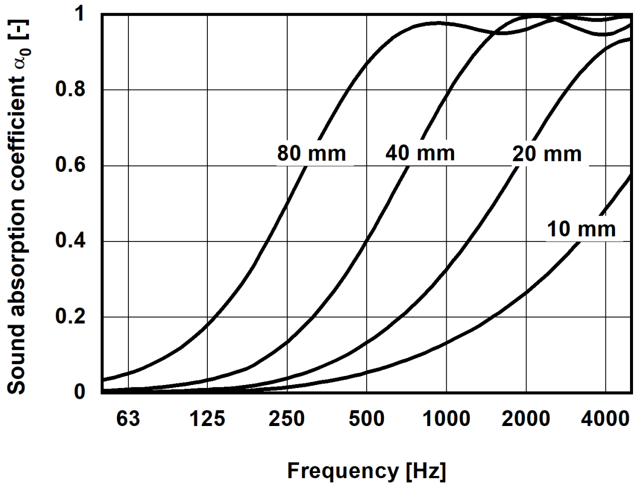


Figure 3-5: Sound absorption depending on the thickness of the layer.

An alternative way to achieve acoustic absorption are acoustic resonators, in particular vibrating plate resonators or Helmholtz resonators. Resonators are mainly used for sound absorption at lower frequencies.

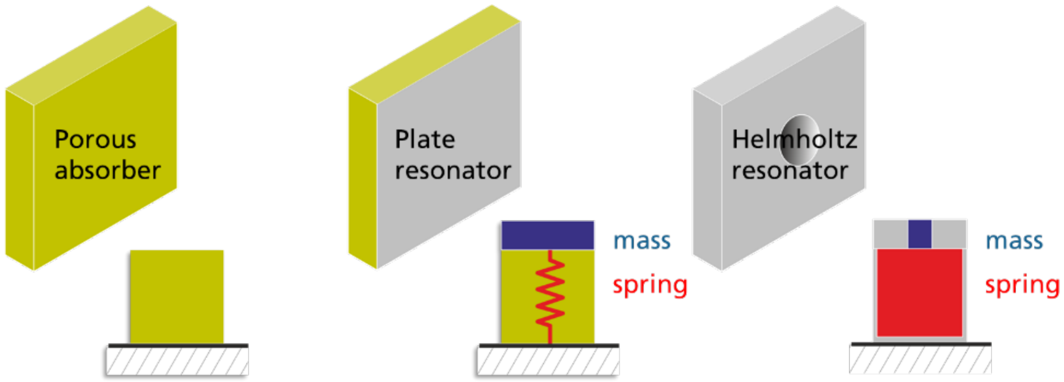


Figure 3-6: From porous absorber to plate resonator (porous absorber with sheet metal cover) and the scheme of a Helmholtz resonator (casing with opening).

If you cover a porous absorber by a plate of sheet metal, as shown in (Figure 3-6), you get a mass-spring resonator. The mass is determined by the plate and the spring by the absorber. Another frequently used resonator is the so-called Helmholtz resonator, where the mass is determined by the mass of air in the neck and the spring by the enclosed air volume in the casing. With resonators, the maximum absorption is achieved at the resonance frequency and the bandwidth is determined by the friction of the resonator system. Thus, a high absorption is associated with low bandwidth. By combining various resonators in parallel, a wider absorption spectra can be realised [3, 9].

For the application of noise control in casings, sound absorption measures can be used to reduce the sound pressure levels inside the casing. Under idealised diffuse field conditions, the resulting sound pressure level L_p is given by

$$L_p = L_W - 10\lg\left(\frac{A}{4}\right) \qquad \text{Eq. 18}$$

where L_w denotes the sound power level of the source and **A** defines the effective absorption surface. This is given by the product of the absorption coefficient and absorber surface. According to Eq. 18, the absorption surface has to be doubled to achieve a reduction of 3 dB of the sound pressure level [3]. For practical applications, this means that in cases where there is no acoustic absorption, considerable reductions may be achieved by adding a relatively small amount of acoustic damping. However, in cases where there is already some acoustic absorption, a significant increase of the acoustic damping material may be required to achieve further perceptible improvement.

3.1.2. Silencers

In almost all applications, machine casings need openings for ventilation, e.g., for cooling, the intake of fresh and the outlet of exhaust air, or as an energy-carrying medium like in heat pumps. Adding silencers is a common way to mitigate noise emissions from air inlets and outlets. This section explains the structure of silencers and the basic principles of their operation and design. Silences are designed to reduce noise which passes through ducts or ventilation slits. Figure 3-7 shows schematic drawings of typical silencer configurations with porous absorber lining material.

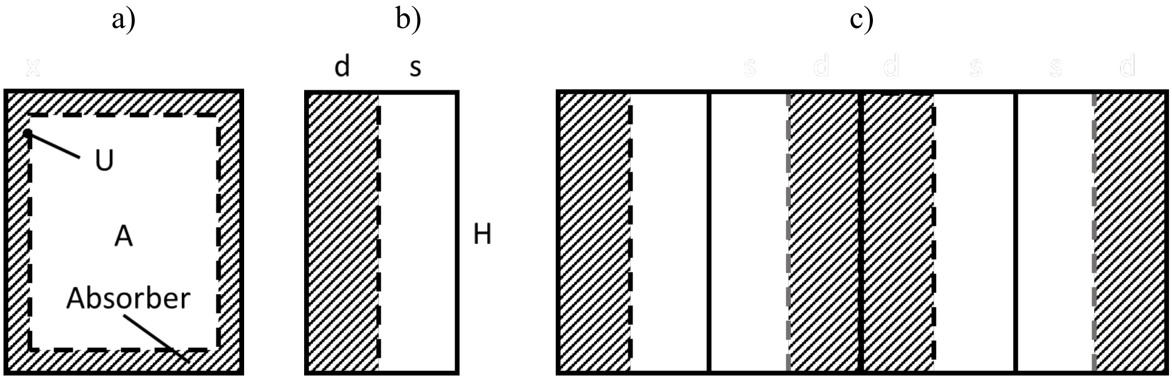


Figure 3-7: a) cross-section of an absorber-lined channel. b) basic element of a splitter type silencer; c) combination of basic elements of a splitter type silencer.

For a rough estimation of the sound attenuation, especially for silencers with linings made from porous acoustic absorption material, the empirically derived equation from Piening [10] can be applied to estimate the resulting sound attenuation:

$$D = 1.5\,\alpha\,\frac{U}{A}\,L\ . \qquad \text{Eq. 19}$$

The sound attenuation D results from the absorber-lined circumference U, the free channel/duct area A, the length of the silencer L and the absorption coefficient α (Figure 3-7, a). Using the basic element of a splitter type silencer (Figure 3-7 b) with an absorber-lined circumference H and the air gap width s, according to Eq. 19, the sound attenuation of splitter type silencers is given by

$$D = 1.5\,\alpha\,\frac{1}{s}\,L\ , \qquad \text{Eq. 20}$$

where the sound absorption α depends on the layer thickness d and the absorber material. The construction of a splitter type silencer and the typical installation is shown in Figure 3-8.

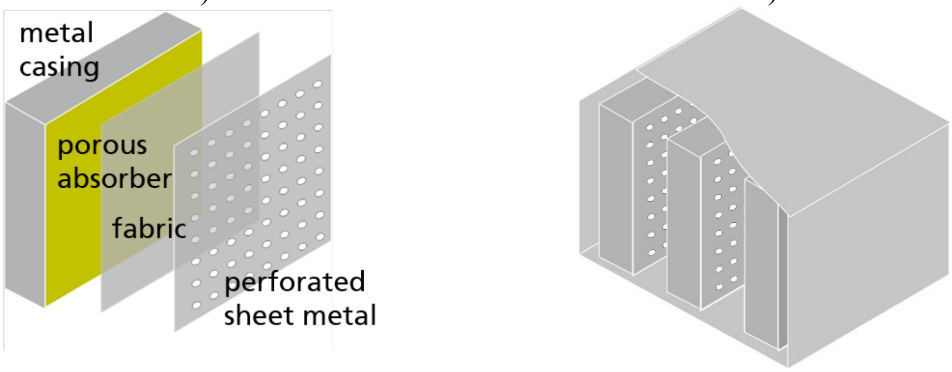


Figure 3-8: a) Components of a splitter silencer b) application of splitter silencers in a duct.

An acoustic baffle consists of a porous absorber, mounted in a metal frame for stabilization and covered by a protective layer, e.g., a fleece in combination with a perforated sheet metal. The most widespread absorber materials are glass, metal wool or wood wool [3]. The performance of the silencer is measured in terms of sound attenuation, which is characterized by the so-called insertion loss. Insertion loss is the difference between the sound power levels in the air duct with and without installed silencers. The insertion loss spectra are usually measured in one-third octave bands, according to DIN EN ISO 7235 [11]. Figure 3-9 shows the typical insertion loss spectra of a conventional splitter silencer.

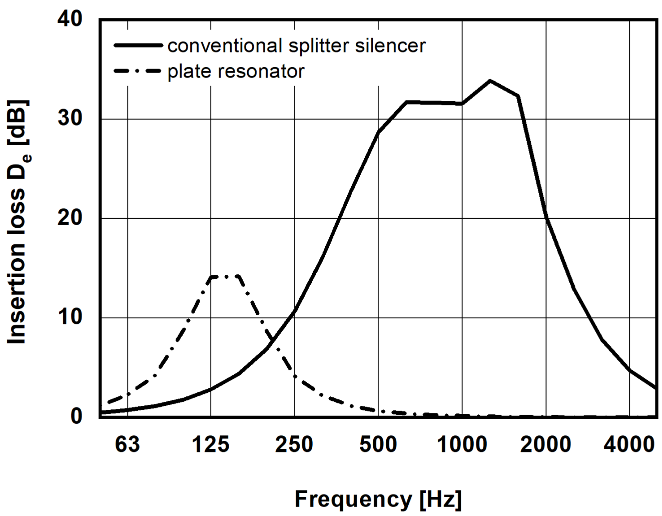


Figure 3-9: Insertion loss spectra in one-third octave bands of a conventional splitter silencer and a splitter silencer with plate resonator.

As shown, absorption silencers have a broadband attenuation spectrum, with limited performance towards lower frequencies. To achieve attenuation at low frequencies, resonators, e.g., a plate resonator, can be used. To obtain the plate resonator, the absorber is covered by a sheet metal plate (see Section 3.1.1). In Figure 3-9 also the typical damping of a plate resonator is shown. Other important parameters for applying silencers are the flow noise and, from an energy efficiency point of view, the total pressure loss.

3.1.3. Sound transmission and insulation

As discussed in Section 3.1, sound transmission describes the transport of sound through a solid partition. In our case, it describes the transportation of sound from the inside of the heat pump to the outside via the walls of the casing. The airborne sound transmission depends on the structural dynamics of the partition, i.e. how easily the partition is excited by acoustic waves on the source side, the resulting vibration levels and the efficiency with which the vibrating partition is radiating sound to the receiving side.

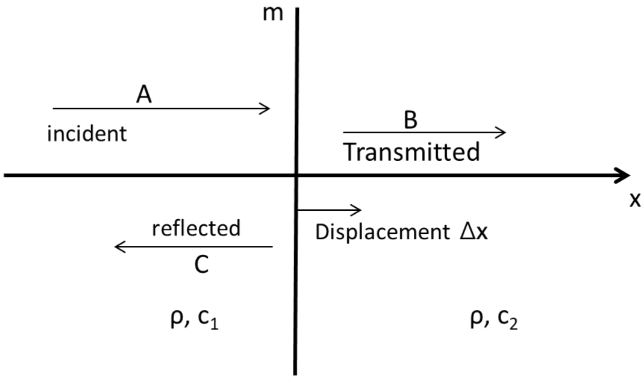


Figure 3-10: Idealized model of sound transmission through a single leaf partition.

A simplified model of the sound transmission process is shown in Figure 3-10 [12]. The considered partition is thin, uniform, plane, and infinite in extend. The partition is characterized by a mass per unit area $\boldsymbol{m}$. In general, the fluids on either side of the partition may be different. They are specified by the terms of their specific acoustic impedances $\rho_1 c_1$ und $\rho_2 c_2$. Their ratio is defined as $r = \rho_1 c_1 / \rho_2 c_2$. The transmission coefficient τ is defined as the ratio between transmitted to incident sound power. In the special case of normal incidence, it is equal to the sound intensity transmission coefficient, which is given by

$$\tau = \frac{4r}{(1+r)^2 + \left(\omega\, m/\rho_2\, c_2\right)^2}, \qquad \text{Eq. 21}$$

where the non-dimensional parameter (ω m/ρ_2 c_2) is a measure of the ratio of the mechanical impedance per unit area of the partition to the acoustic impedance of the fluid on the receiving of the partition [12]. The logarithmic form of the sound power transmission coefficient is given as the "transmission loss", which is defined as

$$\mathrm{TL} = 10 \log_{10}\left(1/\tau\right). \qquad \text{Eq. 22}$$

In case of air-to-air sound transmission, which is the case in many practical applications, the ratio $\boldsymbol{r}$ becomes one. In general, the fluid loading parameter (ω m/ρ_0 c_0) is much greater than unity, so that Eq. 22 may be approximated by

$$\tau = \left(2\rho_0 c_0/\omega\, m\right)^2. \qquad \text{Eq. 23}$$

Hence, the sound transmission is controlled by the product of frequency and the mass per unit area [12]. With the assumed standard values for air (ρ_0,c_0), the transmission loss at frequencies above the critical frequency is defined by:

$$\mathrm{TL}(0) = 20\log_{10}(\mathrm{fm}) - 42\ [\mathrm{dB}]. \qquad \text{Eq. 24}$$

Where (0) indicates the normal incidence and f the frequency [Hz] at which the sound reduction index is evaluated. This Equation expresses the normal incident mass law for sound transmission [3]. Curves of TL(0), TL_d and TL_f for subcritical frequencies are compared in Figure 3-11.

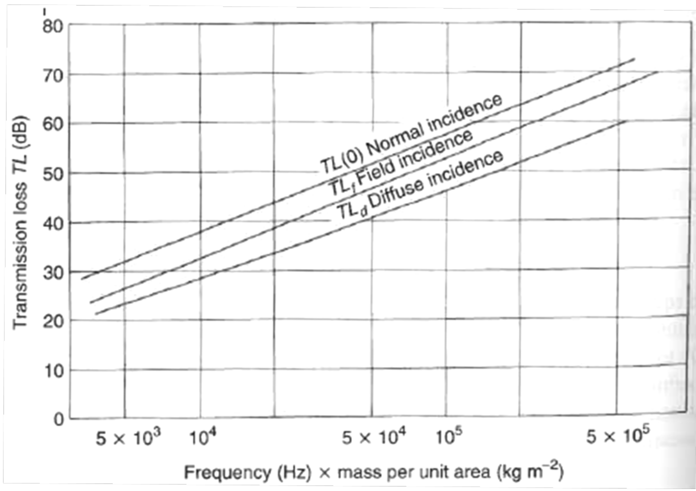


Figure 3-11: Sound reduction indices for infinite partitions at subcritical frequencies [3].

The other two lines show the frequency dependency of the sound reduction indices for 'Field Incidence' and random diffuse field incidence. Experimental characterisation of sound transmission is usually carried out under the assumption of random diffuse acoustic fields on the source and receiving side.

Besides the mass as a dominating factor, the sound transmission through an infinite partition also depends on the boundary conditions, the effective bending stiffness, and the internal structural damping. Figure 3-12 illustrates the tree typical sound transmission regions for the sound transmission between two diffuse sound fields via finite partitions.

At 'low' frequencies, the sound transmission is dominated by well separated low order structural modes of the partition, which are easily excited by the acoustic wave. The natural frequencies of these structural modes depend on the dimension of the partition, the bending stiffness in combination with the mass per unit area, and the boundary conditions along the edges of the partition. The vibration amplitudes depend on the internal structural damping. Not all structural modes are equally efficiently excited by an acoustic excitation. Even order modes, for example, have a zero net displacement and are therefore less efficiently excited than odd order modes. In many cases, the one-one order mode dominates the sound transmission through a partition in that frequency region.

In the 'mid' frequency range, the structural modes start to overlap, and spatial correlation effects between the diffuse acoustical fields on the source and receiving side and the structural dynamics of the partition result in a sound transmission characteristic which largely depends on the mass per unit area and follows the mass law. In this frequency region, neither the bending stiffness nor the structural damping play a significant role for air-to-air diffuse field transmission.

At 'higher' frequencies, the so-called acoustic coincidence effect causes a dip in the sound transmission loss. This effect occurs when the acoustic wavelength and the bending wavelength of the partition start to match. If the wavelengths match, the bending waves of the partition are easily excited by acoustic waves on the source side. Also sound is efficiently radiated on the receiving side. For different angles of attack of the acoustic waves, acoustic coincidence occurs at different frequencies. For acoustic diffuse field excitation, this results in an relatively broad acoustic coincidence frequency region. The frequencies at which acoustic coincidence effects occur, largely depends on the bending wavelength. The bending wavelength depends on the mass per unit area and the bending stiffness of the partition. For thin steal sheets, acoustic coincidence effects occur at the upper end of the audio frequency range and may therefore be unproblematic. However, for rigid lightweight structures, like sandwich panels or structures made from EPS particle foams, the acoustic coincidence effects may already occur in the middle audio frequency range, which can lead to problems for the noise insulation properties of heat pump casings. In this acoustic coincidence frequency range, though, the sound transmission loss is sensitive to structural damping. Adding structural damping can reduce the depth of the dip in sound transmission loss in the coincidence frequency range.

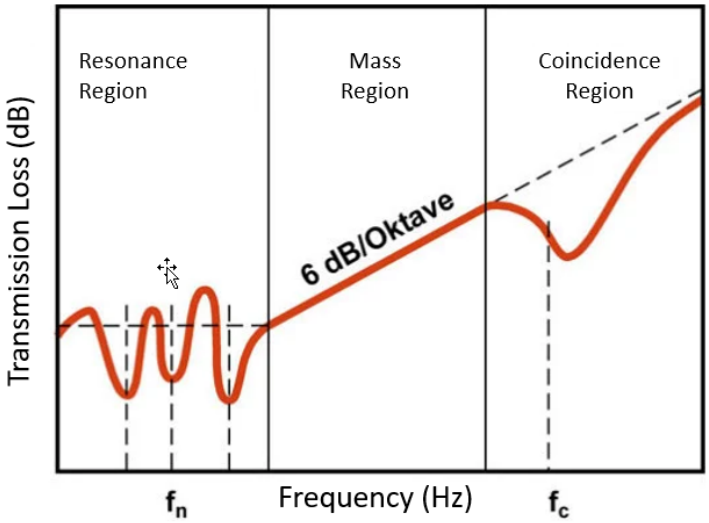


Figure 3-12: Typical sound transmission ranges of a plate-like partition of finite size [13].

3.1.4. Active noise control techniques

Towards low frequencies, the effectiveness of passive acoustic treatments is limited. According to the mass law mentioned in the section above, the sound transmission loss of partitions decreases with decreasing frequency. To be effective, porous acoustic absorbers should also have a thickness of about one quarter of the acoustic wavelength [14]. Acoustic resonators e.g. Helmholtz resonators or λ/4 resonators, have a band-limited absorption characteristic, which depends on the tuning frequency and inner damping [9]. The tuning frequency and efficiency of acoustic resonators largely depends on the resonator back volume. Hence, due to the long acoustic wavelength at low frequencies, conventional airborne noise control concepts are heavy and require a large installation space.

For applications with weight restrictions and where the installation space is limited, active noise control (ANC) approaches can be an alternative. There are many different ANC concepts for various applications. In general, these approaches can be divided into two main classes, feedforward ANC and feedback ANC. An overview of the principle ANC concepts is given in references [15] and [16].

The feedforward ANC requires an error and reference sensor (physical or virtual), a control transducer, and a controller. In most practical applications, an adaptive control algorithm is implemented on a digital signal processor (DSP); an example for a feedforward ANC module for ventilation ducts is described in [17]. The main benefit of feedforward ANC systems is that they allow to control both broadband and tonal noise components, depending on the control algorithms and Feedforward ANC system set-up used. An overview of different control algorithms and control system configurations is given in references [18] and [19]. The main drawback of adaptive feedforward ANC systems is that they are relatively complex and relatively expensive. Currently, feedforward ANC is mainly used in the automotive industry to control tonal combustion engine noise components and, more recently, also broadband road noise. Feedforward ANC is also used in specialised applications in the aerospace industry. Otherwise, feed forward ANC technology has not yet had its breakthrough into wider commercial applications. However, due to the fact that more and more machines like heat pumps are fitted with electronic controllers, feedforward ANC may become more feasible for a wider range of applications in the future. Feedback ANC systems usually comprise a sensor-actuator pair coupled via a feedback control loop. This feedback control loop may be implemented on a DSP or as an analogue electronic circuit. In theory, feedback loops with perfectly tuned and mutually dual sensor-actuator pairs are unconditionally stable. However, practical feedback loops are only conditionally stable to a maximum stable feedback gain and may produce unwanted control spill-over effects. Hence, the gain of a feedback ANC system needs to be carefully tuned. Nevertheless, implementing a feedback controller using an analogue circuit has the benefit that the hardware costs and system complexity are relatively low. One example for a successful implementation are active resonator silencer cassettes (ASCs), which have been developed at the Fraunhofer IBP for more than two decades [20]. One drawback of feedback ANC systems is that they cannot tackle tonal noise components, but usually only act in a band-limited low frequency range.

3.2. Structure-borne noise

The meaning of sound radiation by vibrating structures is of great practical importance [12]. On the one hand, the product designers have to think about the appearance of the casing, on the other hand, they have to consider the vibration properties of the material that is used. In the casing, vibration is generated through vibrating components, such as the compressor and, more or less, the fan. As already mentioned in the introduction, transmission paths, such as fluid pipes or structure parts, are the reason for noise transportation to the heat pump casing (Figure 3-13). The vibration is transported through the heat pump pillars to the acoustic housing. The oscillations make the acoustic housing emit noise to the environment. Structure-born noise is also transported via additional transmission paths to the ground. As Figure 3-13 shows, the ground also emits noise to the environment, which originally comes from the heat pump.

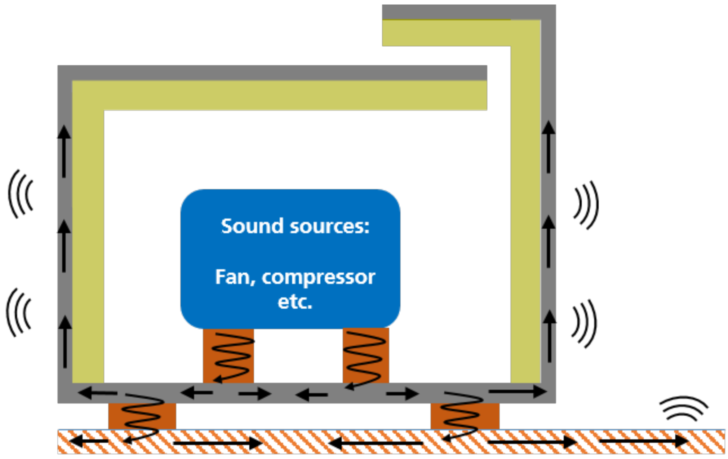


Figure 3-13: Schematic drawing of the transmission of structure-borne noise from a vibrating heat pump to the environment.

Structures which radiate sound through vibration are extremely diverse in their geometric forms, material properties and construction forms. In practice, the process of theoretical estimation of the detailed spectrum and directivity of the noise is very difficult [12]. In the following sections, various methods which can be applied against structure born noise are explained.

3.2.1. Vibration isolation

Vibration isolation is the dynamic decoupling of two connected systems and is usually achieved by placing a resilient element (isolator) in the transmission path [21]. In practice, two common situations occur:

- Isolation of a vibrating machine from its surroundings;
- Isolation of a delicate piece of equipment from a vibration host structure.

At relatively low frequencies, when the flexural wavelength in the source and receiving structure is long compared to its dimensions, vibration isolation can be considered using simple lumped parameter models. This elementary analysis has some merits, but it also has various deficiencies when vibration isolation at 'high frequencies' needs to be considered. In this context the term 'high frequency' is used when the vibrating source, the isolator, or the receiving structure exhibits a resonant behaviour. Alternative mobility modelling approaches enable to consider also the high frequency dynamics in the system.

Assuming a vibrating source on a simple isolator suspended from an ideally rigid base, the system can be modelled as a Single Degree Of Freedom (SDOF) system. In this simple case, the performance of the isolator can be expressed in terms of the ratio of the force transmitted to the base and the force produced by the source. This ratio is known as the force transmissibility, which should be as low as possible for good isolation;

$$T = \left| \frac{\textit{Force transmitted to the base}}{\textit{Force exherted by the source}} \right|. \qquad \text{Eq. 25}$$

One issue in defining vibration isolation requirements is that the source may be supported on a wide range of different types of base structures, ranging from solid concrete foundation or floors with high thickness to lightweight sub-floor systems, steel frames, and steel grids. The structural dynamics of a lightweight flexible base structure will have a significant impact on the effectiveness of the isolator system and the resulting vibration levels of the base structure.

Therefore, in cases where either the source or the receiving structure can no longer be considered rigid, a simple transmissibility model will not give an accurate prediction of the effectiveness of the isolator. An alternative mobility approach is required [21]. Within the mobility approach, the point and transfer responses of the system component are expressed in terms of complex point and transfer mobilities. The performance of the isolation system can be described in terms of the isolator effectiveness, which should be as high as possible for good isolation;

$$E = \left| \frac{\textit{Velocity of receiver when it is rigidly connected to the source}}{\textit{Velocity of rceiver when it is connected via the isolator}} \right|. \quad \text{Eq. 26}$$

Note that transmissibility and effectiveness as defined above are not strictly comparable; however, comparing the two quantities gives a useful reference for the impact that resonant behaviour of the receiving structure has on the performance of the isolation system.

For the evaluation of the resulting vibration level produced on the receiving structure, it is important to know the excitation characteristics of the source (spectrum of the complex forces produced by the source) and the structural response of the receiving structure (complex point mobility, i.e., velocity response / excitation force).

Consider two structures coupled together at their interface by rigid and elastic connections (Figure 3-14), the structure of the active component or source structure, and the structure of the passive component or receiver.

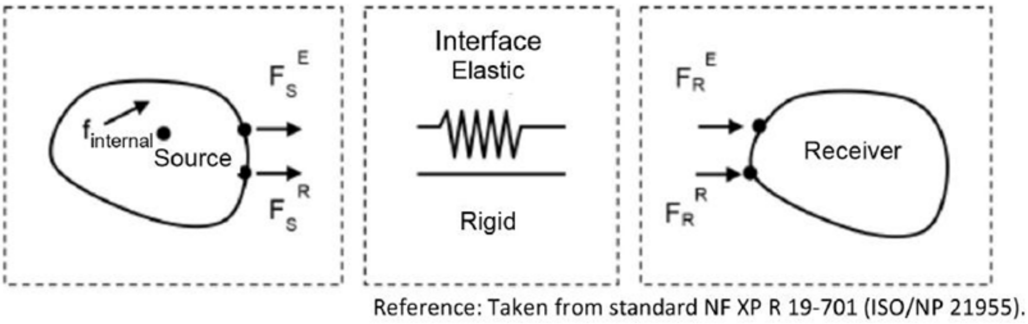


Figure 3-14: Coupling by connections between components Source (active) – Receiver (passive).

The vibrations of a mechanical source, such as a compressor, can change substantially if the source is coupled with the supporting structure by means of the interface connections (Figure 3-14), which may be single or multiple (several connection points); depending on the case, the coupling forces at a connection point may be propagated to the other points and thereby generate interference. The vibratory power injected at each connection point is the product (expressed in watts) of the excitation force and the induced vibration velocity (or the torque and the angular velocity) for each of the six degrees of freedom of the compressor (three translations and three rotations). As stated above, the vibratory power injected at each point is highly dependent on the source-receiver coupling, i.e., the ratio of the admittances (or impedances) in contact. Therefore, in principle, an admittance matrix (6 x 6) should be determined at each connection point. For N contact points, this gives a (6N x 6N) matrix, which is very cumbersome to obtain in practice (measuring the transfer functions between all vibration forces and all vibration velocities). **A** complete and totally accurate characterization inevitably entails a large number of measurements, which in turn creates problems of uncertainty, dispersion and accuracy of the results. For this reason, some characterization methods apply simplified coupling modes (e.g. source on resilient mount to approximate ISO 9611, or use of an equivalent forces for the method developed in the standard ISO 21955 and described below). Nevertheless, such simplifications must be judged and/or estimated in order to consider whether it is really necessary to take into account all the connection points or whether a more general approach is feasible. An optimum balance must be achieved between simplicity (linked with industrial application), method robustness, and fidelity of the results. The theoretical approaches demonstrate that a component can be characterized intrinsically by measuring two independent quantities:

- a dynamic quantity, which characterizes its capacity to generate an excitation; such as free velocities or blocked forces at the coupling points of the source on the structure;
- static quantities for the component and the receiving structure, such as impedances or mobilities.

These quantities are used in the model at the interface between component and receiving structure. The techniques of identification by measuring the source characteristics and the vibration transfer paths can be divided into two categories:

- "Direct" ways, where the required variable is measured or calculated;
- "Indirect" methods, where the required variable is obtained by inversion, after measuring other quantities.

As an example, following, an analysis of the structure-borne noise of a compressor, is investigated. The "Block-Sensor Method" is used to characterize a "scroll" compressor at different operational points. The compressor is supported by its 4 feet, either rigidly, by bolts, or softly, via resilient rubber mounts. It is an indirect method which uses a solid steel block as receiving structure to determine intrinsic quantities to characterise the source. A solid steel block was used as a sensor-receiver (Figure 3-15).

Figure 3-15: Experimental set-up to determine a characterisation of a compressor for structure-borne sound.

This method allows to determine the impedance of the component and the "blocked forces" (Figure 3-16a) and b)) that are generated by the compressor (the blocked forces are the forces that would be applied by the component on a perfectly rigid structure).

a) b)

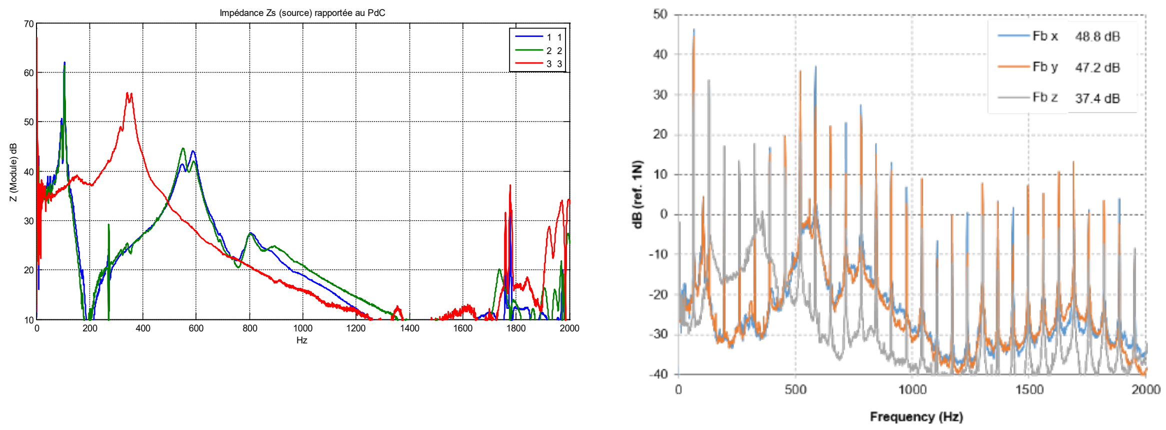


Figure 3-16: a) Impedance of the component for 3 directions; b) Blocked force.

This method can be used to measure the influence of soft mountings (Figure 3-17).

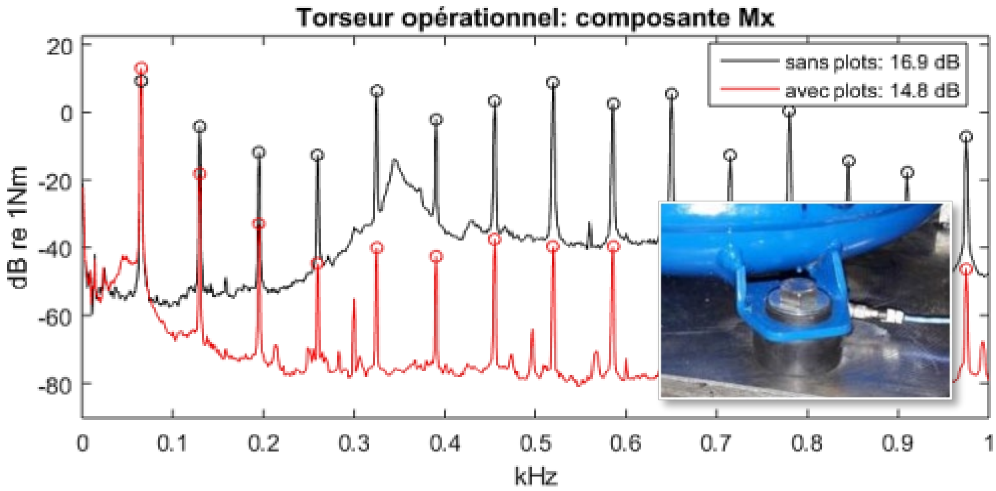


Figure 3-17: Blocked forces Fb with/without isolators.

3.2.2. Fluid-borne noise

The methodology to characterize the fluid-borne noise is identical to that used for the structure-borne noise. The characterization of a source of pressure pulsations (fluid-borne noise) is done by identifying the pulsed flow rate and the hydraulic impedance. A major difference is that the hydraulic impedance component can depend of the static pressure and the flow rate. The sensors that are used are dynamic pressure sensors or PVDF wires.

3.2.3. Modelling and design of passive isolation systems

The Systematic sketches of four different vibration isolation models are shown in Figure 3-18. Model (a) with the source supported by an ideally rigid surface via an idealised spring-damper system, (b) with the source supported by an ideally rigid surface via a two spring-damper system with the heat pump chassis as embedded mass, (c) with the source supported by a flexible plate structure via an idealised spring, and (d) with the source supported by a flexible plate structure via a two spring-damper system with the heat pump casing as embedded mass.

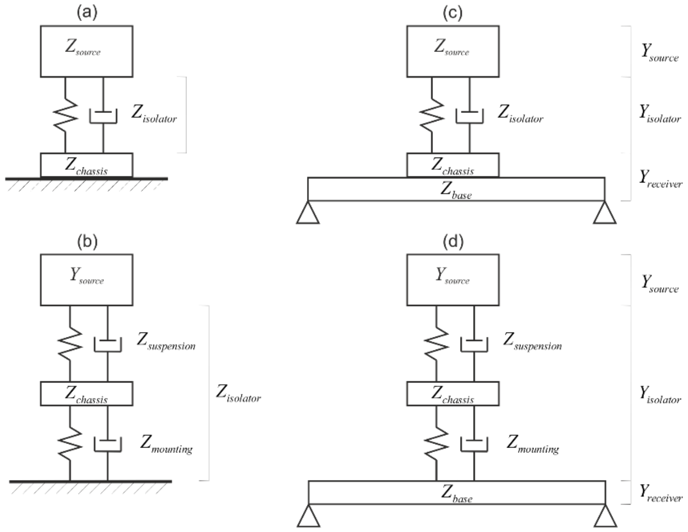


Figure 3-18: Sketches of the four idealised vibration isolation mounting systems.

In case (a), the performance of the isolator can be described in terms of the transmissibility [21] which is given by

$$T = \frac{1 + j\eta}{1 - \Omega^2 + j\eta}, \qquad \text{Eq. 27}$$

where $\Omega = \omega / \omega_n$ is the forcing frequency normalised by the undamped natural frequency $\omega_n = \sqrt{k/m}$ of the source on the isolator, and η is the damping loss factor of the isolator.

In case (b) it is assumed that the casing itself may not be rigidly mounted to the base, which results in an additional system resonance. In the case of such a two-stage isolator system, the transmissibility is given by

$$T = \frac{(1 + j\eta)^2}{s(1-s)\mu\Omega^4 - (1 + j\eta)\left(1 + (1-s)\mu\right)\Omega^2 + (1 + j\eta)^2}, \qquad \text{Eq. 28}$$

where $s = 1/(1 + k_s / k_m)$ which is the combined stiffness of the suspension k_s and the mounting k_m in series divided by $\boldsymbol{\mu = m_c/m_s}$, and k_s , the ratio between the mass of the chassis and that of the pump. As in model (a), Ω is the forcing frequency normalised by the resonant frequency of the system (with $\boldsymbol{m_c}$ set to zero).

Note that all models assume hysteretic damping, such that the impedance of the spring-damper system is given by $Z = k(1 + j\eta)/(j\omega)$, i.e., the impedance is expressed in terms of a complex stiffness. Note that model (b) in Eq. 28 is simplified, such that it assumes equal damping loss factors in both spring-damper systems.

Figure 3-19 shows the exemplary transmissibility of an a) a single and b) two-stage elastic vibration isolation system over a frequency range which is normalised to the natural frequency of the principle Single Degree of Freedom System (SDOF), of the source mass on the elastic

$\sqrt{2}$ times the normalised frequency, the magnitude of the transmissibility drops below unity, and generally increases continuously with increasing frequency. This means that the vibration isolation is reducing the transmission from the source to the base. Therefore the frequency range above $\sqrt{2}$ times the normalised frequency is also referred to as the isolation frequency range of the isolation system. The amount of amplification in the frequency range around the natural frequency of the principle SDOF can be reduced by additional damping of the elastic isolator. However, by adding viscous damping to the isolator , the slope of the transmissibility in the isolation frequency range is changed, whereas increasing the viscous damping reduces the efficiency of the isolation.

The two-stage isolation system in model (b) shown in Figure 3-18 b creates a Two-DOF System, which exhibits two principle resonance peaks in the transmissibility. The benefit of such a system is that above the second natural frequency of the Two-DOF system, the transmissibility decreases twice the rate as the transmissibility of a single-stage isolation. In practice, two-stage isolation systems are often implemented by mounting a machine with isolators on an elastically supported concrete plate or heavy and rigid steel frame. The stiffnesses below the source mass and the intermediate mass need to be chosen with care. In order to avoid excessive vibration amplitudes of the source mass, the natural frequency of the lower SDOF (intermediate foundation) is usually chosen to be lower than that of the upper SDOF (source mass).

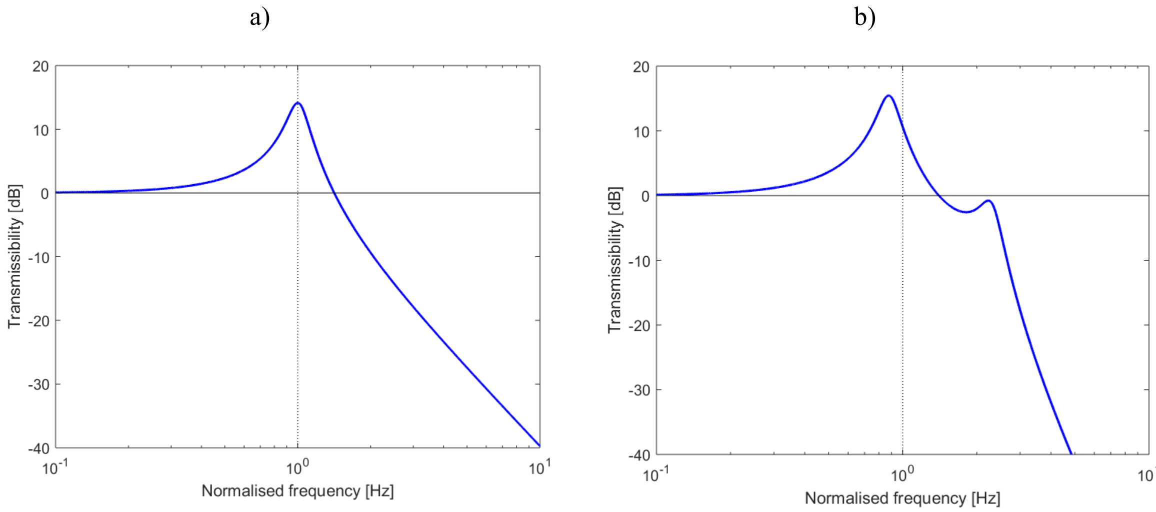


Figure 3-19: Exemplary transmissibility of a) a simple single-stage isolation system and b) a double-stage isolation system.

Typical elastic isolators are made from rubber blocks or coil springs. Above a certain frequency, these isolation elements will exhibit internal structural resonances, which can mitigate the effectiveness of the isolation system at higher frequencies. The effect of internal modes of the isolation is not covered in the simplified models in Eq. 27 and Eq. 28. Often it is not sufficient to consider the source vibration and the vibration isolation behaviour in the vertical axis only. Complex machines may produce vibration forces with different phase relationships in different directions, which can result in moment excitations and rocking motions of the machine. For a more detailed prediction of the transmissibility theses effects may need to be taken into account.

Another crucial simplification made in the model (a) and (b) is that it assumes that the isolator reacts from a rigid base with infinite impedance. For practical applications, this may not be the case. Resonances of the base structure can significantly reduce the effectiveness of the isolations system. Especially if machines like heat pumps are mounted on lightweight, relatively flexible steel frames, the dynamics of the flexible base structure may need to be taken into account in the prediction model for the transmissibility of the isolation system.

To account for the input mobility of the receiving floor structure, it is necessary to use a mobility approach [9, 14]. In this approach the dynamics of the source, the isolator and the receiving structure are described in terms of point and transfer mobilities. The effectiveness of the isolator system for the models (c) and (d) is then given by

$$E_I = \left| 1 + \frac{Y_I}{Y_S + Y_R} \right|, \qquad \text{Eq. 29}$$

Where $\boldsymbol{Y_I}$ is the transfer mobility of the isolator $\boldsymbol{Y_s}$ is the mobility of the source and Y_R is the mobility of the receiving base structure. Note that the inverse of the effectiveness is to a certain extend (not strictly) comparable to the results of the transmissibility models (a) and (b).

The mobilities of the source and the receiver and the transfer mobility of the suspension system $\boldsymbol{Y_I}$ need to be chosen such that they represent the actual system. Considering the chassis to be a lumped mass, the transfer mobility of the isolator in model (c) is given by

$$Y_{I(c)} = Z_I^{-1} = \left[k_I \left(1 + j\eta_I\right) / \left(j\omega\right) \right]^{-1}, \qquad \text{Eq. 30}$$

where $\boldsymbol{k_i}$ and $\boldsymbol{\eta_i}$ are the stiffness and the loss factor of the suspension. In model (d), the transfer mobility is given by

$$Y_{I(d)} = \left[\frac{Z_s Z_m}{Z_c + Z_s + Z_m} \right]^{-1}, \qquad \text{Eq. 31}$$

where the subscripts *s* and *m* denote the suspension (isolator) and the chassis mounting. The impedances Z_s and Z_m can be found analogue to Eq. 30. Other measures in the suspension system, like passive vibration absorbers and neutralizers, can also readily be implemented in terms of as a modified isolator transfer mobility formulation.

Figure 3-20 shows the inverse of the vibration isolation effectiveness of a single-stage isolation system on a flexible base structure. The example shows that the structural resonances of a flexible base can significantly affect the performance of the vibration isolation system, compared to a system reacting to an infinite base.

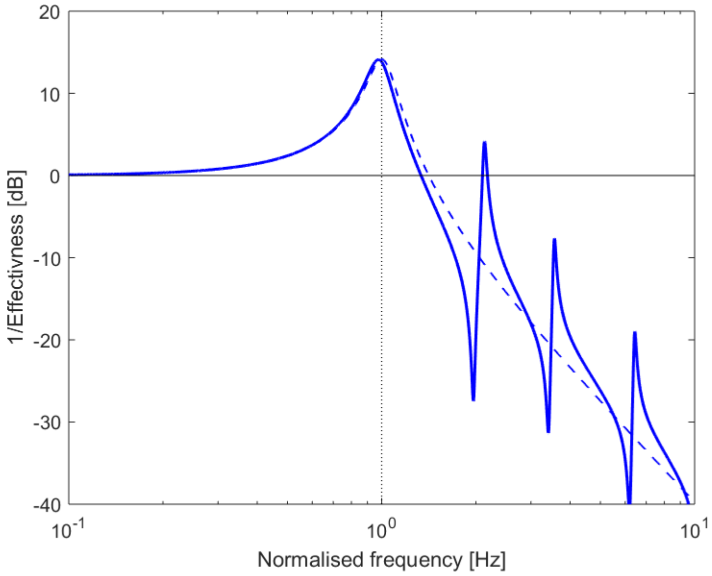


Figure 3-20: Exemplary inverse of the isolation effectiveness of a simple single-stage isolation system on a flexible base structure (solid line), and the transmissibility of a corresponding single-stage vibration isolation system reacting to an infinite impedance (dashed line).

3.2.4. Active vibration control techniques

Active vibration control solutions have the potential to increase the vibration isolation performance as compared to solely passive systems. A multitude of active control strategies and concepts can be found in standard textbooks [22, 23]. With regard to active vibration isolation, the concepts can be divided into two general cases for the source characteristic:

- Stochastic broadband excitation source spectrum
- Deterministic (periodic) excitation spectrum

If the source produces a broadband stochastic disturbance, no reference signal is available. In these cases, active control systems rely on feedback control strategies. In a feedback approach, a control actuator with collocated (closely located) sensor is employed to produce a control force on a structure to reduce/modify the system response at the control point.

In contrast, for deterministic tonal excitation, there usually is a reference signal available, e.g., rotational speed of the machinery, which can be used in a feedforward control strategy. For feedforward control, the reference signal is used together with an error sensor signal to implement a control force which is cancelled out with the primary disturbance at the error sensor position. Feedforward control strategies usually use adaptive algorithms that allow tracking changes in the source characteristics. Often it also possible to combine feedforward and feedback approaches in order to enhance performance.

The core of an active isolation system is an electronic control unit on which a control algorithm can be implemented. Such an active control systems can have the following characteristics:

- Compensation of stochastic and deterministic (periodic) excitation
- Effectiveness across a wide frequency range
- Automatic tuning function

- Possibility of controlling multiple actuators for the suppression of the overall vibration induced by all mounting points.
- Possibility of addressing multiple critical forcing frequencies and critical response resonance frequencies with a single actuator unit (controlled with a complex control signal with multiple poles and zeros).
- Possibility of implementing multifunctionality, e.g., condition monitoring

The main benefit of an active isolation system is that it can adapt variations in the forcing frequency, i.e., changes in the pumps' operational conditions. With an appropriate system, this adaptation could also track the forcing frequency during run-up phase, which may be problematic with a purely passive or passive/adaptive system. In addition, actuator units are able to provide higher actuation force at a lower total system weight than a purely passive solution.

There are multiple concepts for adaptive/active vibration isolation solutions. As an example, an actively tuneable bending beam vibration neutraliser was investigated in Ref. [24]. In practice, the selection of an optimal control concept requires detailed simulation studies with more detailed information of the actual source and receiver characteristics. While up to date there are quite a few active mounting systems available for automotive applications, there seems to be no off the-shelf active vibration isolation systems available for applications in heat pumps.

4. Concepts for Component Noise Control

In this section specific noise control techniques for specific heat pump components are addressed.

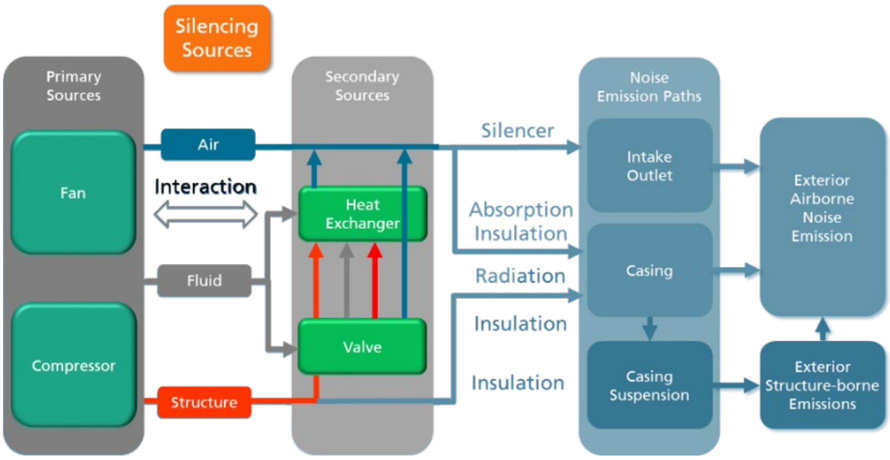


Figure 4-1: Block diagram indicating the primary and secondary noise sources, their interactions and main transmission paths.

4.1. Compressor

The noise emission spectrum of the compressors is dominated by tonal components. The noise can be emitted either via direct acoustic radiation from the compressor casing or via structure-borne transmission to the heat pump chassis, from where it is transmitted to the heat pump casing which radiates the noise to the outside.

4.1.1. Passive treatments

Some investigations were carried out on the airborne noise of a small reciprocating compressor, and the passive treatment to reduce its radiated sound power level.

The compressor, 11.5 cm³ of R407C, 11.5 kg, delivers 1600 W for 50 Hz according to EN 12900 (Cond. T° +50°C / Evap. T° +5°C / Superheat 10 K / Subcooling. 0 K (dew point). Its cylindrical shape dimension is approximately 200 mm in length and 120 mm in diameter. There is no standard soundproof jacket for this compressor reference.

The goal of the study was to determine some acoustic performances of coverings, using different strategies of mixing absorbing material and heavy mass external materials. The reference is an external wood box lying on the floor, totally disconnected from the compressor (or the ducts). It represents a kind of "best performance", even if it is not a realistic configuration.

Figure 4-2:. Reciprocating compressor used as source.

The covers are made either with absorbing material or with a complex consisting of an external mass layer (EPDM or equivalent for 5 or 10 kg/m²) and an internal acoustic layer with viscoelastic foam or felt, PU foam, FireSeal, polyester down.

The compressor is installed on the floor of a reverberant room, in which its sound power level is determined according to ISO 3741. The performance of the jacket is calculated as an insertion loss by comparing the sound power level in the reverberant chamber with and without the soundproof jacket. The overall values are calculated using two references: the A-weighted pink noise and the compressor noise.

The operating conditions are maintained constant using a thermodynamic loop.

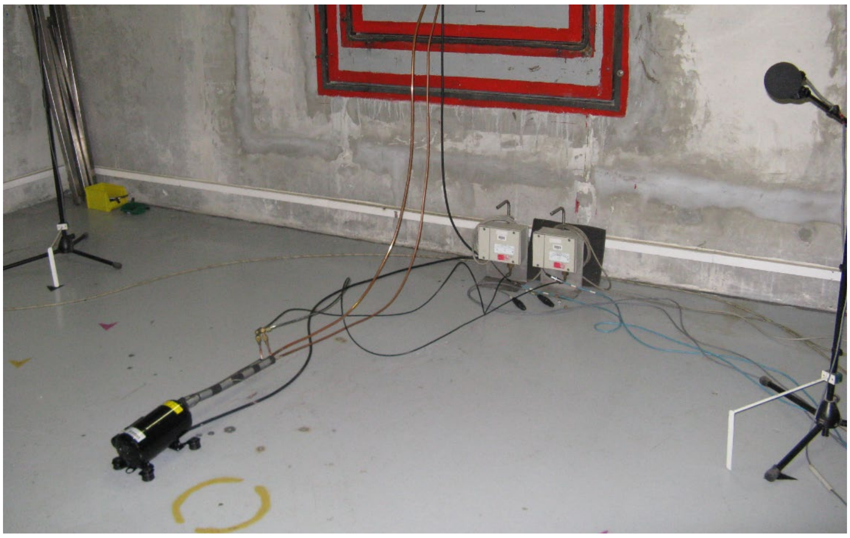


Figure 4-3: Installation of the compressor in the reverberant room.

The wood box is placed on the floor with a peripheral soft seal and has no contact with the discharge or inlet tubes. The inner sides of the box are covered by foam and EPDM 10 kg/m².

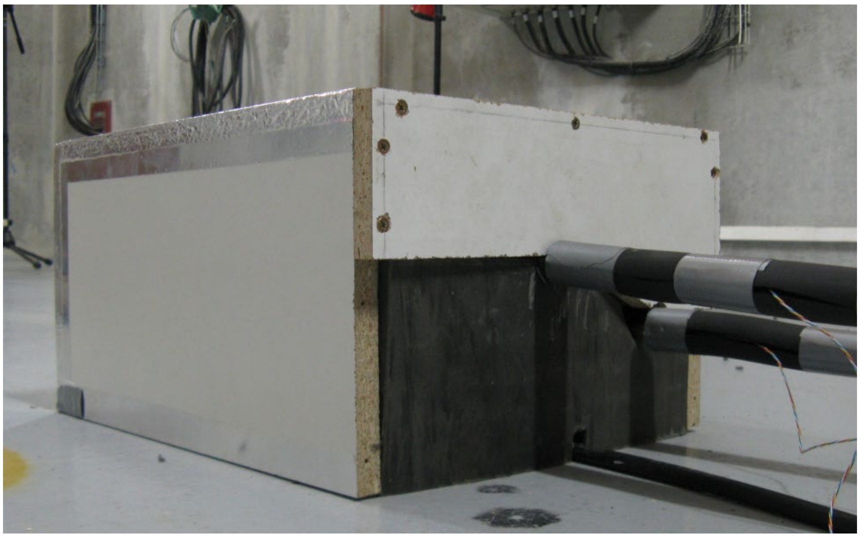


Figure 4-4: Uncoupled wood box around the compressor.

The result shows that without contact, the sound power level can be reduced by 12 dB and 15 dB(A). It should be noted that, with the box, the sound power level at 125 Hz increases due to air coupling. The unwanted contact increases the low frequency amplification.

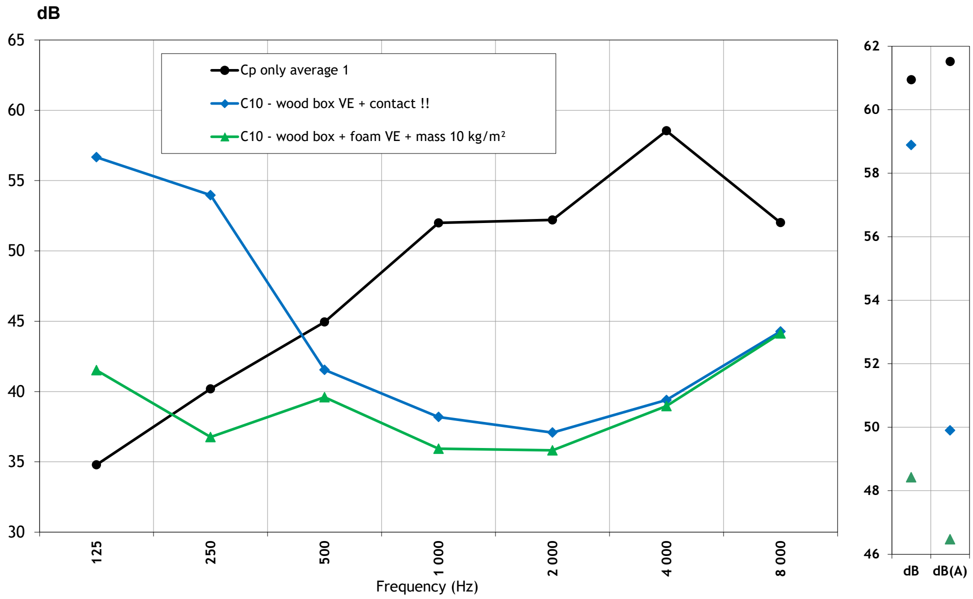


Figure 4-5: Sound power level of the compressor with wood box and without the wood box.

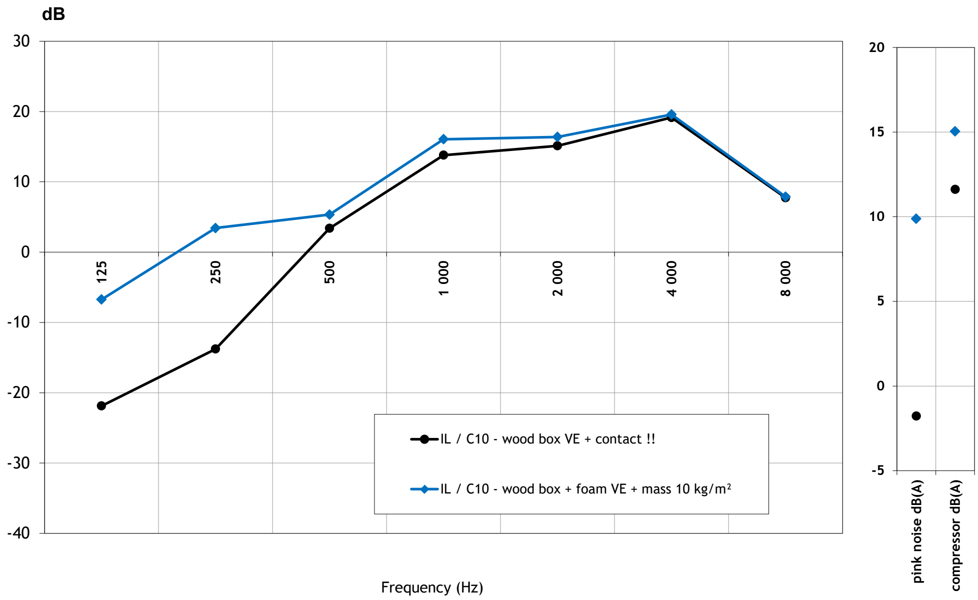


Figure 4-6: Insertion loss for wood box solutions.

The other configurations use several types of internal foam and 5 or 10 kg/m² of external damping layer. The results are usually close and it should be noted that the implementation is a sensitive parameter.

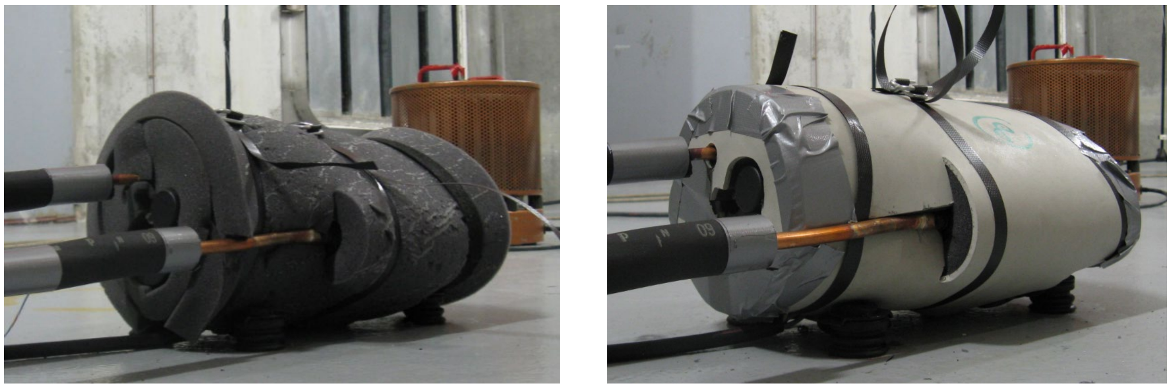


Figure 4-7: Foam (left) and foam + damping layer (right).

<u>Complexity of absorbing material and external damping</u>

The use of a single layer of absorbing material versus a complex assembly of absorbing and external damping layer leads to very different results. The external layer reduces the radiation from 1000 Hz, but amplifies the sound at 125 and 250 Hz. The overall gain is better with the complex assembly, but is highly dependent on the spectrum shape to be attenuated.

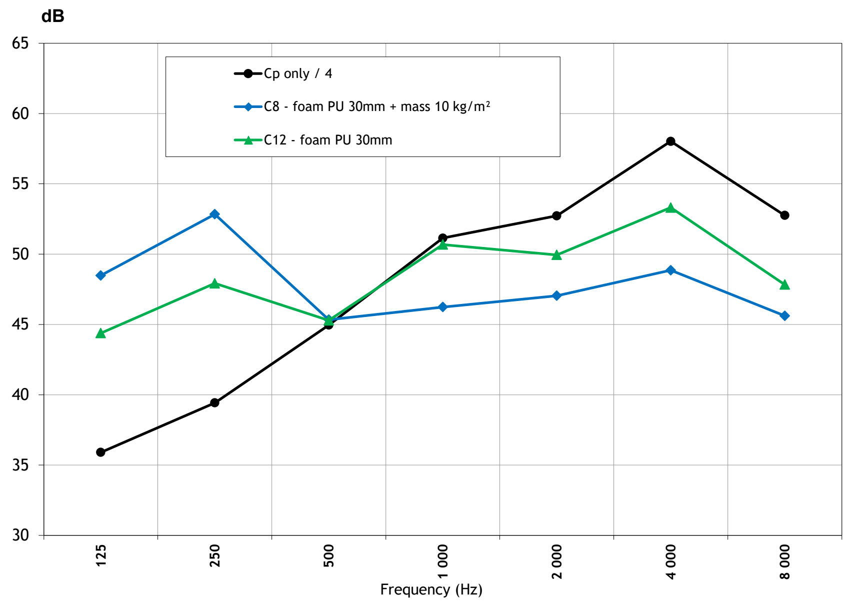


Figure 4-8: Sound power level of compressor without foam, with foam and foam + damping layer.

The results of all the different configurations are not listed in detail here, because the results have results are of the same type, ± 3 dB on the individual frequencies, and ± 2 dB on the overall results. Probably due to the implementation, it is difficult to conclude from the results that thicker absorbing material or heavier mass systematically leads to better results of insertion loss. On average, the insertion losses calculated from the bare compressor noise give results between 5 to 8 dB(A) for "standard" solutions. Only solutions based on an uncoupled box allow reaching an insertion loss of 13 to 15 dB(A). The insertion losses calculated from A-weighted pink noise as reference give drastically lower values.

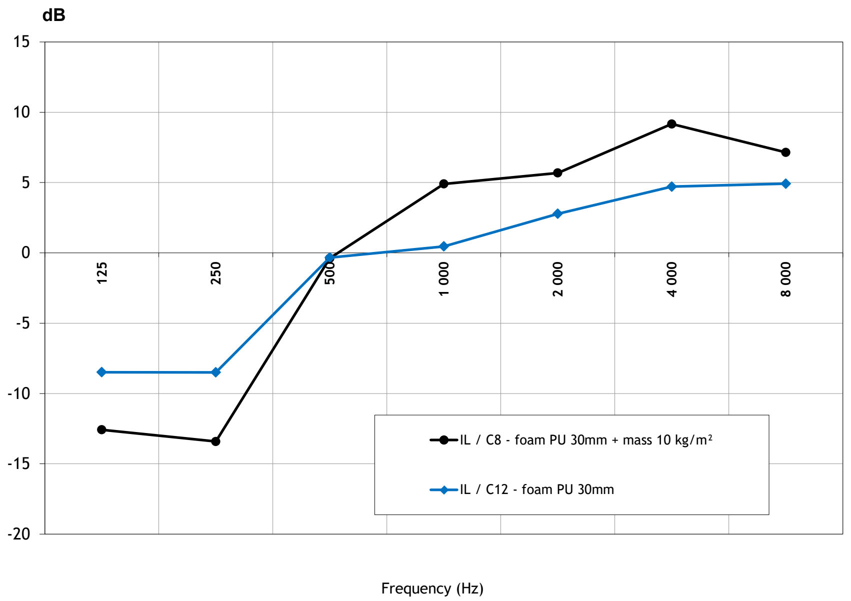


Figure 4-9: Insertion loss for foam or foam + damping layer solutions.

4.1.2. Active vibration control

The excitation spectrum of a compressor is dominated by discrete tonal components, which requires feedforward AVC. The application that comes closest to the problem faced with the heat pumps and their compressors is that of active engine mounts in automotive applications. Active engine mounts are not yet standard off the-shelf systems, but are usually specifically designed. They are usually only used for the main engine and not for auxiliary units. The difficulty in adopting concepts from existing active engine mounts from an automotive application is the difference in weight and dimensions between a combustion engine and a compressor for the most heat pump applications. The principles of active vibration isolation are well documented in academic literature. However, for most applications, it is not yet state-of-the-art technology. There are viewing systems, which are commercially available for automotive applications, but not for vibration isolation of small auxiliary units and components like heat pump compressors.

4.2. Fan

Depending on the type of the fan, the spectrum of the noise emissions can have broadband and/or dominating tonal components. The noise is mainly emitted via direct acoustic radiation from the air outlet of the heat pump. Structure-borne noise transmissions from the fan to the chassis are possible, but are probably of secondary concern in most cases.

4.2.1. Fan noise

Tonal noise reduction, which is particularly relevant for high-speed fans, requires methods derived from the description of the mechanisms in section 2.1.1. Since the discrete noise generation is due to an interaction between the incoming, non-uniform flow field and the impeller, it is advisable to keep the inlet flow field as homogeneous as possible by keeping a distance of at least 3 to 4 duct diameters between the fan and inlet obstacles such as struts, bends, dampers, and quick duct expansion or contraction. Avoiding inlet guide vanes and large obstructions just before the impeller is strongly recommended.

In the same way, fixed obstacles near the outlet of axial fans, which radiate noise when the rotating blade wakes act on them, should be moved away from the impeller. A practical guideline is to keep a distance of at least one rotor blade chord between the impeller and downstream obstacles, such as outlet guide vanes (OGV) or motor struts. The number of vanes or struts is also an important parameter for noise control, especially if the distance between the impeller and the downstream obstacles is not large enough.

Tonal noise of centrifugal fans is mainly due to the interaction between the blade wakes and the volute cut-off (or volute tongue, see Figure 4-10). Different means have been successfully used to reduce this interaction, for instance by increasing the cut-off clearance. The minimum clearance recommended for noise control of centrifugal fans is 10 to 12% of the impeller diameter. Such clearance does not significantly reduce fan performance.

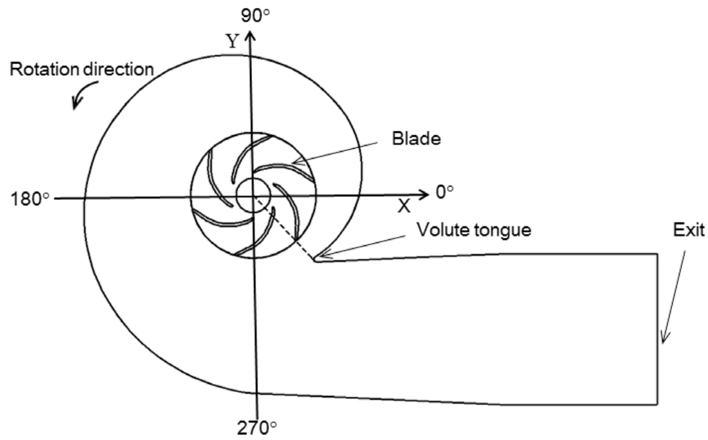


Figure 4-10:. Outline of the impeller and volute of a centrifugal fan.

The control of broadband noise of axial and centrifugal fans is more complex than that of tonal noise, because the hierarchy of the noise sources for a given fan is sometimes difficult to define. Returning to the main phenomena which contribute to the broadband noise mentioned in section 2.1.1:

- Noise due to intake of turbulence,
- Blade trailing edge noise,
- Vortex shedding noise,
- Flow separation associated with rotating stall.

Noise due to turbulence, also called blade leading edge noise, is predominant when the flow entering the impeller is highly turbulent, e.g., in the cooling fan of a car or truck with the radiator close to the fan inlet. It is recommended to reduce the turbulence by using the means already mentioned to reduce tonal noise, i.e. moving the obstacle away from the fan, either upstream or preferably downstream.

Blade trailing edge noise may be reduced for instance by a serrated blade trailing edge (Figure 4-11), but this type of noise control is still the subject of laboratory research as the mechanisms associated with the serrations are not yet fully understood, which may lead to disappointment.

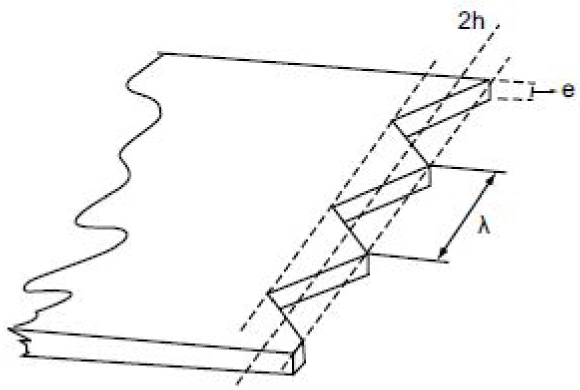


Figure 4-11:. Serrated blade trailing edge.

Vortex shedding noise may be reduced by thinning the trailing edge or using the serrated trailing edge as described above.

Flow separation on the blades may be due, among other reasons, to a stall when the operating point of the fan is far from its best efficiency point on the left side of the performance curve. The pressure loss of the system must be decreased to avoid stall and thus reduce noise.

There are many other ways to control fan noise exist, that may be found for instance in the book "Fan Acoustics – Noise Generation and Control Methods" by Alain Guedel (Published by Air Movement and Control Association International, 2007).

4.2.2. Passive treatments

In [25], Carolus mentions a few passive methods to influence fan acoustics. In Section 6, he describes that the rotation speed of the fan blade has a strong influence on the sound power. On the one hand, the message of T. Carolus is to reduce the rotational speed. In practice, in order to work at maximum efficiency, fans need to reach their aerodynamic operation point $\dot{\boldsymbol{V}}/\Delta \boldsymbol{p_t}$ for which they are dimensioned. Hence, if the rotational speed is reduced, the dimensionless parameters φ and ψ should have high levels. The latter leads to a higher stress level for the wheel (e.g. high number of blades or strongly skewed blades) and can cause noise mechanisms which are called "self-noise" [25]. Many investigations have already been carried out on the influence of the blade contour, e.g. from F. Czwielong et al. [26]. He investigated the interaction of different blade shapes (Figure 4-12) in combination with a heat exchanger. In his investigations he found out that the backward-skewed fan in combination with the heat exchanger has the largest change in the spectra compared to others.

a) b) c)

Figure 4-12: Fan with unskewed blades (a) Fan with forward-skewed blades c) Fan with backward-skewed blades [26].

Carolus [25] also mentions that in case of a disturbed three-dimensional flow, not only the blade geometry is responsible for the acoustic behaviour. More important is the phase mismatch between the blade contour and the disturbance in the flow on the suction side. Fan manufacturers also use passive treatments which are not based on a change in fan geometry. Figure 4-13 shows an axial fan in combination with a diffusor. Using a diffusor minimises the exit losses, improves the fan efficiency and finally reduces the noise.

Figure 4-13: Axial fan with a diffusor on the outlet side [27].

As shown in Figure 4-14, ebm-papst constructed an air inlet grille for noise reduction [28]. The "FlowGrid" is used to reduce the turbulence and to further reduce disturbances in the fan inflow.

Figure 4-14:. Air-inlet grille for axial fans (a) and radial fans (b) [28].

The "FlowGrid" reduces the noise which is generated by the disturbances in the fan inflow. As shown in Figure 4-15, the air inlet grille reduces not only the tonal components, but also the sound pressure level across the whole spectrum.

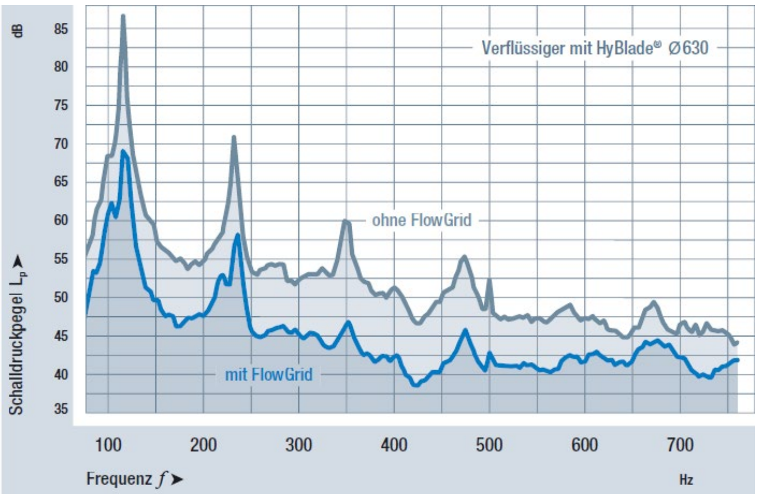


Figure 4-15: Spectra of the sound pressure level L_p, using a fan in combination with the FlowGrid (blue graph) and without the FlowGrid (grey graph) [28].

4.2.3. Active noise control

In heat pump applications, the fans are mounted in a casing wall and blow freely to the outside. Due to the often large size of the fan used and the therefore complex free-field sound radiation characteristics, simple Single Input Single Output (SISO) ANC systems, as they can be implemented in ducts (see Section 5.1.2), are not sufficient. For this type of application a multi-channel Multi Input Multi Output (MIMO) ANC system with multiple loudspeakers and microphone sensors would be required. The complexity and cost of such a system is therefore relatively high. The complexity of the sound radiation characteristics of the ventilator may be reduced by installing diffusors or baffle plates.

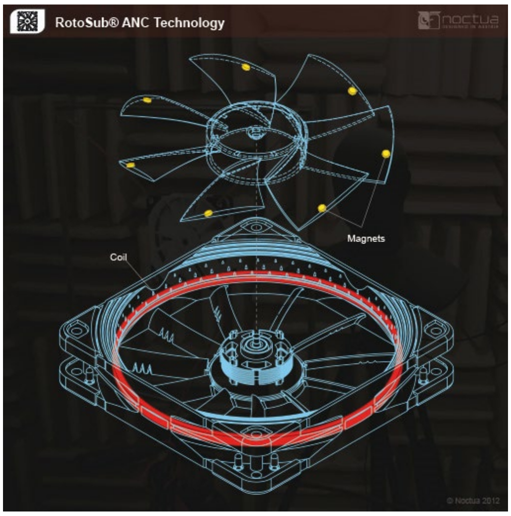
An additional problem is that in outdoor applications, the acoustic behaviour of the system, which changes over time, i.e. due to the temperature differences between winter and summer, must be taken into account. Most adaptive algorithms are capable of compensating for smaller variations. However, for the application of ANC to ventilators of heat pumps, control algorithms with adaptive secondary path model may be required.

Different fan manufacturers have investigated concepts for compact feed forward ANC systems and also developed functional demonstrators. Figure 4-16 shows a demonstrator of ebm-papst with a compact ANC system which acts on both sides of the ventilator [29]. An S-Cube Development Kit of the Silentium Ltd. was used as controller hardware [30]**.**

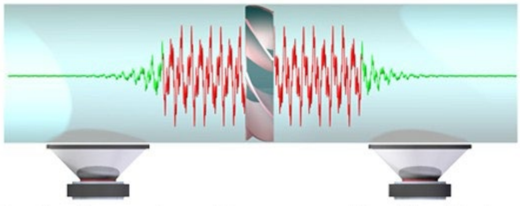
Figure 4-16: Function demonstrator of a compact ANC system which acts on both sides of the ventilator.

Another approach is the patented system by Rotosub AB [31] Rotosub® R-ANC-System (US20070230720, EP1752016B1) [32–34]. In this integrated ANC concept, no additional ANC loudspeakers are required. Instead, the blades of the ventilator themselves are used as an anti-sound source. An actuation system with magnets in the blade tips and an electric coil in the ventilator casing allows dynamically varying the angle of attack of the ventilator plates. The aim of this system is to reduce tonal components of the ventilator noise spectrum. Figure 4-17 shows the set-up of the Rotosub® system and its functionality. The technology is available for licensing, but no products featuring this technology appear to be available on the market. It also seems unclear if this technology can be scaled up to the type and size of fans typically used in heat pumps.

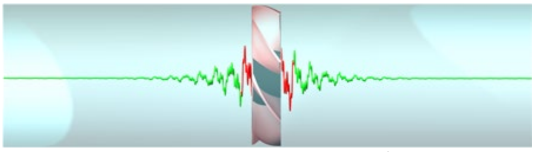
a)



b)



Two-sided ANC using a classical ANC system with two control loudspeakers.



Two-sided ANC using the Rotosub® system with activated fan blades.

Figure 4-17: a) Rotosub® system set-up, source Noctua Webpage [35] and b) illustration of the RotoSub® system functionality , source Rotosub AB Webpage [31] .

The patent documents of the Asia Vital Components Co., Ltd. [36] (DE2014003524U1) [36], i.e., US Patent Application (US20150296295A1) [37] describe a system in which a voice coil actuator on the fan axle is used to produce anti-sound. However, whether such an ANC system is commercially available or has ever been implemented in a functional demonstrator is neither evident from the patent documents nor from the company's webpage.

In summary, ANC for fans remains a challenging topic without commercial solutions that would be readily available to heat pump manufacturers. ANC systems integrated into fans remain an R&D challenge for ventilator manufactures. The most promising approach for future developments seems to be a combination/integration of classical ANC techniques together with passive treatments.

4.3. Secondary sources

Apart from the compressor and the fan, individual heat pumps may contain secondary sources like pumps or electric converters, which need to be acoustically treated, especially if the main acoustic sources have already been treated efficiently.

5. Noise Emissions Paths and Concepts for Noise Control

Figure 5-1 shows the three main ways how the sound emission from heat pumps can take place: direct acoustic emissions through the air intake and outlet, sound radiation from the casing, and structure-borne excitations via suspension of the heat pump casing. The latter is transmitted to the base structure (e.g., a roof or wall structure) and may cause sound radiation at locations remote from the actual heat pump. In the previous sections, various methods are presented to reduce the noise emission directly at the source or by interrupting the air- and structure-borne transmission paths close to the source locations inside the heat pump. In this section, methods close to the emission points around the heat pump casing are addressed.

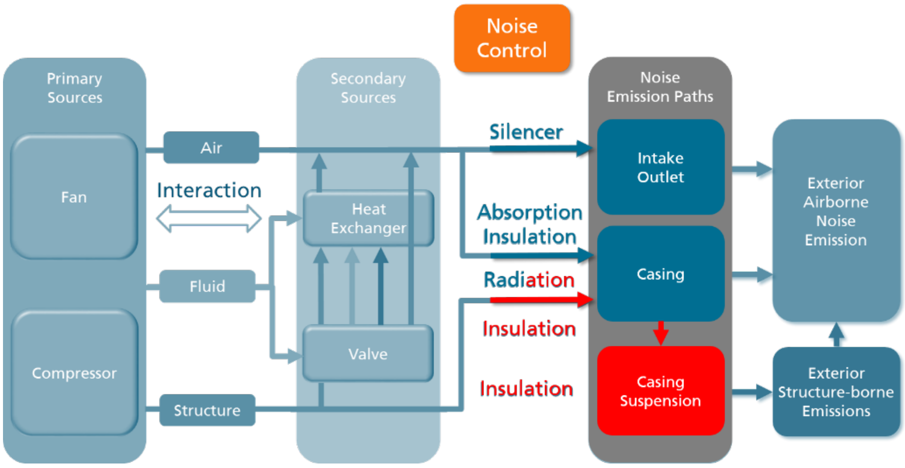


Figure 5-1: Block diagram indicating the main airborne and structure-borne emission paths of the heat pump to the exterior.

5.1. Air intake and outlets

Especially the noise from split heat pumps can be transported to the environment through the inlet or outlet duct. In Section 3.1, various methods are described how to control the sound propagation in the ducts caused by the heat pump.

5.1.1. Passive treatments

In order to reduce the emitted sound at the inlets and outlets of the ducts, several passive treatments can be applied. Typical measures are inlet and outlet acoustic silencers. The sound radiation of noise from the duct surfaces can be reduced by an additional covering of absorption material on the outer surface (see Section 3.1.1) or by using double-walled ducts. Heat pumps of the latest generation, which are state of the art, already feature various passive treatments. The heat pump manufacturer Bosch, for example, uses noise diffusors and baffle plates to reduce acoustic emissions at the outlets of the heat pump outdoor units [38].

Figure 5-2: Bosch Compress 7400i AW with noise diffusor at the outlet [38],[39].

5.1.2. Active noise control

Most exhaust and ventilation systems comprise specifically designed silencers and absorbers to reduce noise emission. As discussed in Section 3.1.2, conventional silencers usually comprise porous absorbers and/or acoustics resonators. For applications with weight restrictions and where installation space is limited, active noise control (ANC) approaches can be an alternative. Some general features of ANC are described in Section 3.1.4. Below the so-called cut-off frequency of higher order duct modes, which depends on the geometry of the duct cross section, the sound field in inlet and outlet ducts may be described as plane waves with equal pressure distribution across the duct cross section. Below the cut-off frequency, duct ANC-systems can be implemented as relatively simple SISO ANC systems. Similar to the discussion in Section 3.1.4, ANC for duct applications may be divided into feedforward and feedback ANC approaches.

Feed forward ANC systems for inlet and outlet ducts require an error and reference sensor (physical or virtual), a control transducer and a controller. Figure 5-3 illustrates the basic setup of a feedforward ANC System in a duct. A reference sensor (microphone) is used to sense a sound wave ‘upstream’ from an control loudspeaker location. This reference signal is filtered via a controller transfer function and played back via the control loudspeaker such that the primary and secondary sound fields destructively interact and cancel each other out. A challenge is that the controller transfer function needed is not known. In very early concepts of ANC, the magnitude and phase of single tones were adjusted manually in order to achieve the desired cancellation effect at the control loudspeaker. Today, in most practical applications, an adaptive control algorithm is implemented on a digital signal processor (DSP). Apart from the digital controller, such systems also require an ‘downstream’ error signal, which the system tries to minimise. Most often a so-called LMS algorithm or variations of it are used for online system identification and adaptation of the controller transfer function. Depending on the application, feedforward ANC systems may be implemented to reduce low frequency noise or to cancel out only tonal noise components (engine or fan orders).

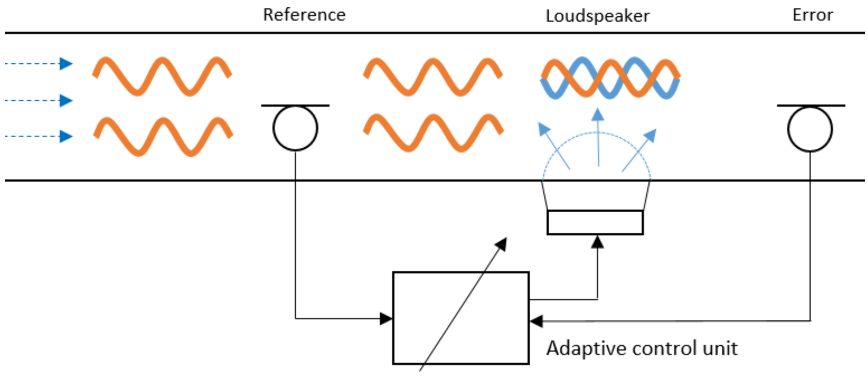


Figure 5-3: Schematic drawing of a typical feedforward ANC unit with reference and error microphone sensors and digital control unit.

Feedback ANC systems for inlet and outlet ducts usually comprise a sensor-actuator pair, coupled via a feedback control loop. An example of a successful implementation are Active resonator Silencer Cassettes (ASCs) for the control of low frequency noise [40]. An ASC consists of a loudspeaker mounted in an airtight casing and a microphone placed close to the loudspeaker membrane, and an analogue electronic controller circuit with adjustable gain. Figure 5-4 a) shows the basic components of a classical ASC. In essence, a passive ASC is a mass-spring damper system. The loudspeaker membrane is the mass and the membrane suspension together with the integrated air volume in the casing form the spring. The viscoelasticity in the membrane suspension and other effects lead to some damping of the resonance system [40]. Figure 5-4 b) shows that the absorption coefficient of the passive ASC (control off) has a relatively sharp peak at around 125 Hz, which is the resonance frequency of the spring-mass damper system of the loudspeaker. In this case, the loudspeaker is only moved through the incoming noise. When closing the feedback loop, the resulting peak in the absorption coefficient around the loudspeaker resonance frequency is much wider in the frequency range and has a much higher amplitude.

a) b)

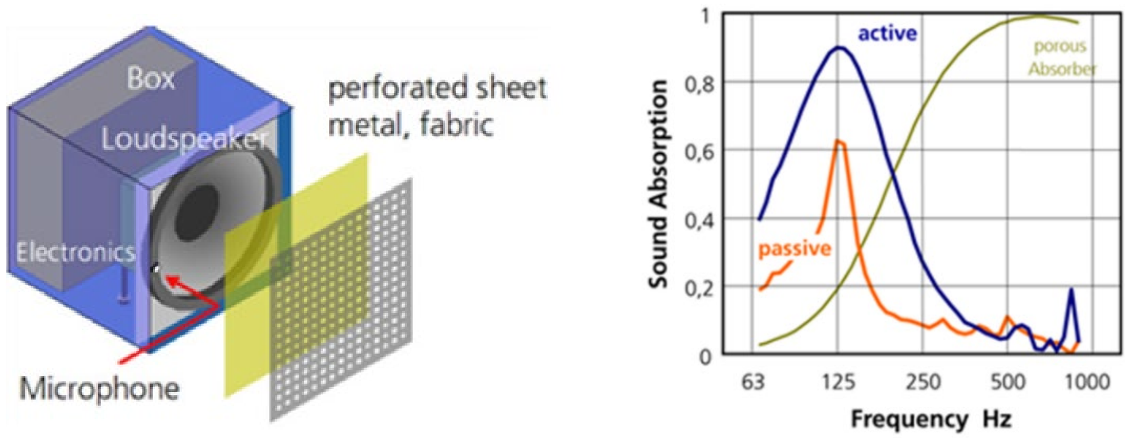


Figure 5-4: a) 3D model of an ASC b)sound absorption coefficient (normal incidence angle) of a porous absorber and an ASC with an open and closed control loop [40].

As reference, Figure 5-4 also shows the absorption characteristics of a porous absorber, which is not efficient at low frequencies. Effective silencer systems with a broad frequency range of performance can be designed by combining ASCs with porous absorbers.

A schematic drawing of a typical feedback ANC silencer systems with an ASC is shown in Figure 5-5. A microphone detects the time-dependent sound pressure. The signal is transferred to an analogue electronic control unit which controls the movement of the membrane. The passive absorber located opposite to the loudspeaker reduces the reflection on the duct wall and compensates for unwanted control spillover effects at higher frequencies [20, 41].

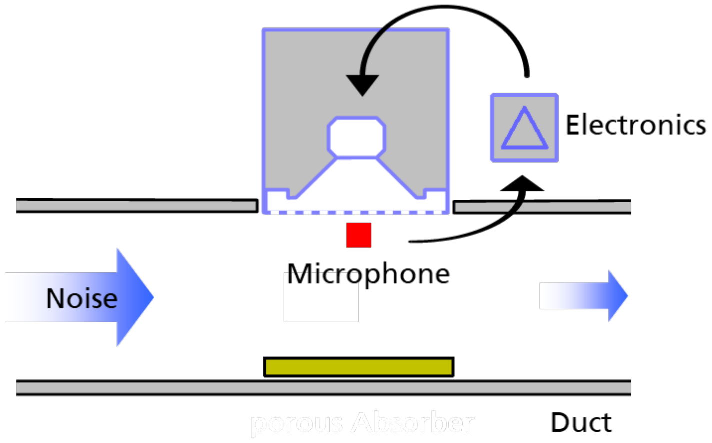


Figure 5-5: Schematic drawing of a typical feedback ANC unit.

5.2. Casing

Sound emissions from the casing occur due to sound radiation from the vibrating casing surfaces. The casing can be excited via acoustic sources inside the heat pump (e.g. via direct acoustic emissions of the compressor) or via structural excitation (e.g. via the vibration transmission from the compressor through the frame to the casing). In this section, various basic principles for reducing noise radiation and emissions from the casing are presented.

5.2.1. Noise insulation

In order to achieve good noise insulation, various design aspects have to be considered. First of all, the basic construction and the support frames of the heat pump are to be taken into account. As already mentioned in detail in Section 3.1.3, for a good airborne noise insolation, the mass per unit area and the internal structural damping has to be high. Accordingly, the designers have to find a compromise between the airborne noise insolation properties and the weight (including structural damping measures) of the casing. It is especially problematic if a tonal acoustic component of the heat pump fits with a structural resonance of one of the casing face plates. In this case, local masses or stiffening elements may help to detune the excitation and resonance frequencies. Another possibility to reduce airborne sound transmission from inside the heat pump to the outside is the reduction of the sound pressure levels inside the heat pump casing. For this purpose, various methods can be applied. In Section 3.1.1 absorption applications are mentioned which can be installed in different ways in the heat pump casing.

5.2.2. Retrofit solutions

Due to strict noise regulations made by local authorities, additional measures may be needed to avoid noise disturbances in densely populated urban areas. If the noise reduction measures applied to the heat pumps themselves do not meet the requirements, retrofitted noise control measures may need to be taken.

Typical retrofit solutions are so-called acoustic housings. Practical examples of such housings are illustrated in Figure 5-6. The aim of the acoustic housing is the further reduction of the acoustic noise emissions, while producing as little additional pressure loss as possible and also not affecting the operation and performance of the heat pumps. These retrofit housings work by providing an additional noise insulation combined with absorption lining inside the housing. By choosing the inlet and outlet geometry and orientation, the directivity of the sound emission can also be optimised and directed away from sensitive immission points.

In several practical cases, it has been shown that retrofitted casings can reduce the noise emitted from heat pumps by more than 20 dB [42, 43]. In addition to noise reduction, these housings protect heat pumps or air condition system from other environmental influences or vandalism.

a) b) c) d)

Figure 5-6: a) and b) Retrofit Solution, Kellner Engineering GmbH, Austria; b) and c) Retrofit Solution, Schallbox dB(A) GmbH, Germany.

5.3. Structure-borne noise emissions

Heat pumps transmit structure-borne noise into the structure which they are mounted to. If the heat pump is installed on a roof or to a wall, this can lead to unwanted noise transmissions into adjacent rooms. In order to select appropriate methods for vibration isolation, it is necessary to conduct a characterization of the components and the receiving structures with regard to structure-borne noise, which cannot be considered independently for the optimization of the vibro-acoustic behaviour. This section deals with the possibilities to structurally isolate the heat pump from its environment.

5.3.1. Passive vibration isolation mounts

The basics of vibration isolation are discussed in Section 3.2.1. Passive vibration isolation mounts are widely used in industrial applications and are already commonly used in heat pumps to isolate the compressor from the chassis or to isolate the heat pump chassis/casing from its base. In most cases these vibration isolation mounts are made from rubber or other elastic materials. Some commercially available vibration isolation mounts are made from steel coil springs or steel cables.

As discussed in Section 3.2.1, when choosing vibration isolation mounts, it is important to know the static and dynamic loads of the source and the vibration isolation frequency range of interest. The vibration isolation mounts must be selected such that they can withstand the static and dynamic loads and with respect to the resulting mass-spring system resonance frequency. If the base on which the source is mounted cannot be considered as sufficiently stiff/heavy (e.g. in the cases where heat pumps are mounted on a metal frame or grid), it is also important to consider the interaction between the source, the vibration isolation and the base structure. Depending on the requirements of the application, isolation systems can comprise multiple elastic components and be combined with vibration absorbers and vibration neutralizers. Vibration isolation mounts need to be carefully selected for each individual application to guarantee optimal vibration isolation performance. A discussion on how to model and design the vibration transmissibility of different isolation systems is provided in section 3.2.3.

5.3.2. Adaptive and active mounts

Most active vibration isolation systems available off-the-shelf are designed to isolate sensitive machinery parts from the base vibrations of the supporting structure. For these applications, a variety of active mounting systems is available such as workstations, benchtop platforms, breadboards, equipment platforms, and vibration-free islands.

The application that comes closest to the problem faced with heat pumps and their compressors is that of active engine mounts in automotive applications. Active engine mounts are not yet standard off-the-shelf-systems but are mostly designed specifically for a particular application [44, 45]. It may be possible to adopt concepts of existing active engine mounts from an automotive application where the weight and dimensions of the vibrating component are similar to those of a conventional combustion engine. Another field of application of active engine mounts is the use in heavy ship diesel engines [46]. Here, too, the solutions are usually tailored to the specific application which is possible due to the high overall cost of the machine.

It can be summarised that the principle of active vibration isolation is well documented in the academic literature but is not yet state-of-the-art technology which is commercially available as a generic product. Currently, there seems to exist no generic off-the-shelf solution which would guarantee to meet the requirements of applications in heat pumps.

5.3.3. Mountings, pipes

It is important to avoid vibration transmission bridges from pipes to the chassis/casing. Therefore pipes and cables should be fastened with appropriate elastic mounts.

6. Effect of Operating Conditions of Heat Pumps on Acoustic Behaviour

The operating conditions, especially transient operation modes influence the heat pumps acoustic behaviour and are an important aspect in the control of noise from heat pumps. In the Annex 51 Task 4 report »Analysis of the Effect of Operating Conditions of Heat Pumps on Acoustic Behaviour« this topic is addressed in greater detail. A short abstract of the Annex 51 Task 4 report is provided below:

The Annex 51 Task 4 report deals with tools that allow the modelling of the acoustic emissions of a heat pump as a function of its operating state. The noise emissions resulting from transient operations are examined, beginning with a description of the physical processes likely to produce the most obvious or bothersome noises, before describing them by order of time scale. Where possible, recommendations on how to reduce the influence of those acoustic emissions are discussed. Finally, the dependency on the type of heat source and the temperature and load levels are examined.

7. Placement and Installation

Placement and installation is an important aspect in the control of noise from heat pumps. In the Annex 51 Task 5.1 report »Report on heat pump installation with special focus on acoustic impact« this topic is addressed in greater detail. An executive summary of the Annex 51 Task 5.1 report is provided below:

Air-to-water heat pumps are also often chosen where space is limited or where there are obstacles in the building regulations. Compared to air-to-air heat pumps, water, which is more suitable for this purpose, is used for heat transfer. A permit is not required. The disadvantage of the air-to-water heat pump is its comparatively low efficiency and increased noise emissions. The latter are mainly caused by the motor of the air intake fan and by the compressor. The aim of this work thesis is therefore to select and place air-to-water heat pumps in such a way that the sound pressure level in the surrounding houses is kept low. In the following chapters, light will be shed on several topics surrounding the placement of heat pumps.

This report presents a selection of tools, which are used for calculating sound pressure levels. This includes simple formula based tools, which are often available online on websites of heat pump manufacturers or heat pump association. Examples shown include a Swiss, German and Austrian version. Two-dimensional visualization is based on the same formulas, but allows the user to see the sound pressure levels in a horizontal plane surrounding the freely placeable heat pump. All these tools neglect - apart from the corner- and wall-placement "penalties" - absorption, reflection or frequency dependencies in the calculation. The underlying formula is very ease and can be calculated by hand. To include the effects of directivity and frequency behaviour as well as absorption and reflection a much larger computational effort would have to be made. Some approaches, which try to shed light on these effects are visited next. Advanced sound propagation tools like CadnaA, SoundPlan, NoiseD3D, Mithra-SIG, IMMI, Olive Tree Lab Suite and OpenPSTD are listed. The full three-dimensional calculation of sound propagation is of course possible solving the acoustic wave equation using e.g. FEM and BEM.

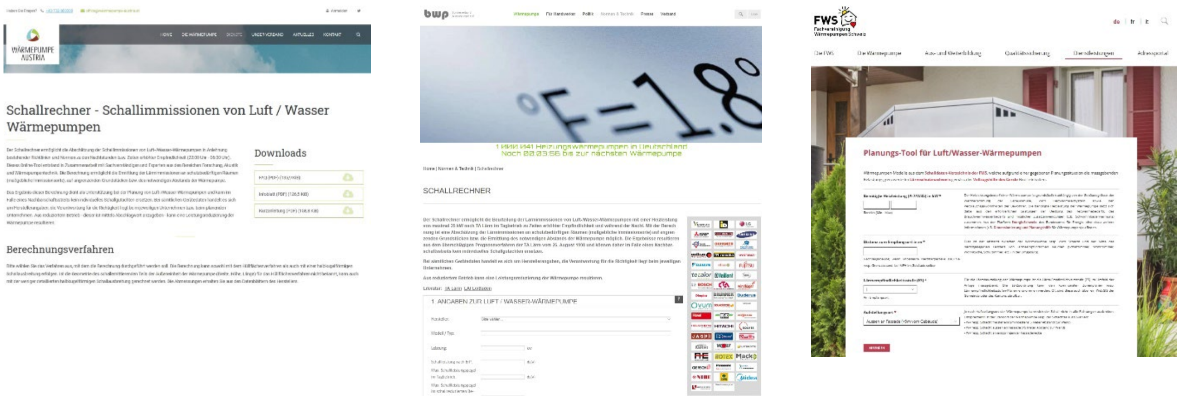


Figure 7-1: Simple web based calculation tools.

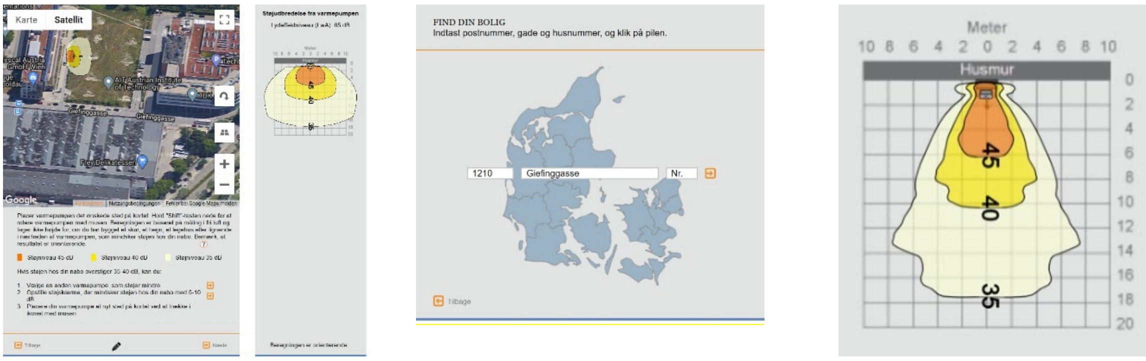


Figure 7-2:: Two-dimensional visualization of sound pressure levels.

The virtual placement of heat pumps using augmented reality is presented. This includes a description of acoustic measurements of noise sources, the auralisation approach and the methods for calculation sound propagation. The app is realized by a modelling and mapping approach and hardware and software for visualization and acoustics are described.

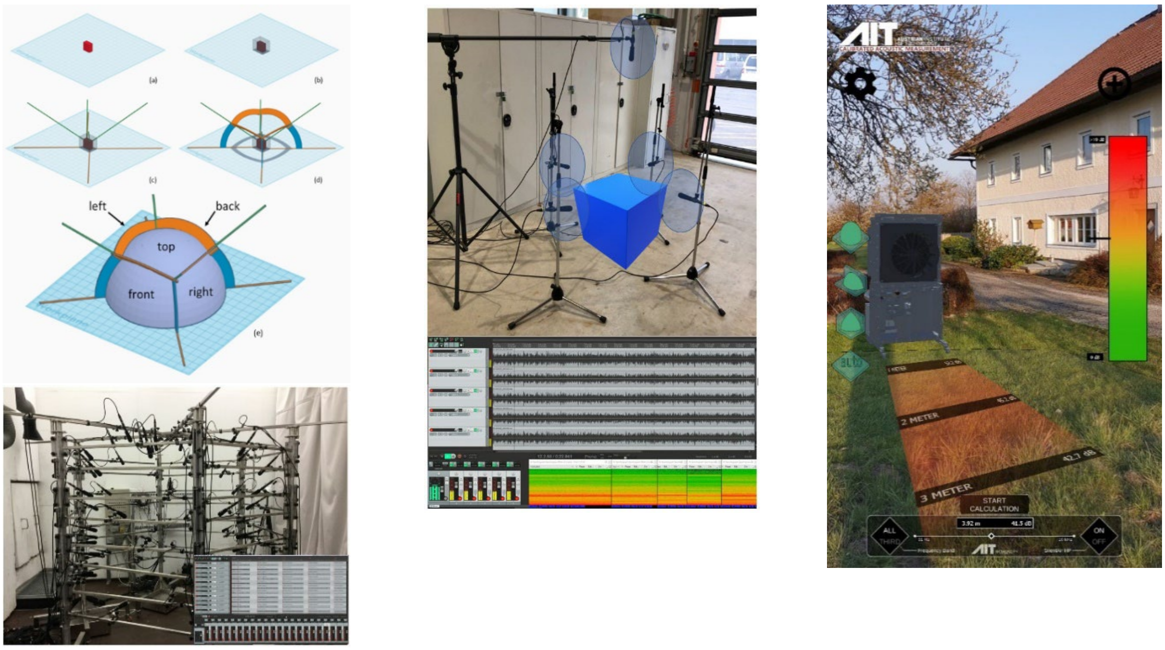


Figure 7-3: Augmented reality and acoustic app.

The acoustic interaction of multiple heat pumps including reduction measures is analysed using primarily the tool IMMI. First, the terraced housing estate chosen for an exemplary study is presented including the description of heating load, hot water provision, heating demand and the analysis of the neighbouring sites. The maximum sound propagation is calculated using IMMI following ÖNORM ISO 9613-2 and ÖNORM S 5021. Several scenarios have been compared: One heat pump per household, one heat pump per house and a local heating supply scenario. In all cases heat pump selection and placement are outlined. Results are compared using a method introducing penalty points on all defined immission points (doors, windows, borders). For a promising case, calculations have been repeated introducing noise barriers into the calculation. Time of day dependent sound propagation have been visited to introduce user profiles of the different buildings. Alternative tools like OpenPSTD and Olive Tree Lab Suite and the involved options are described.

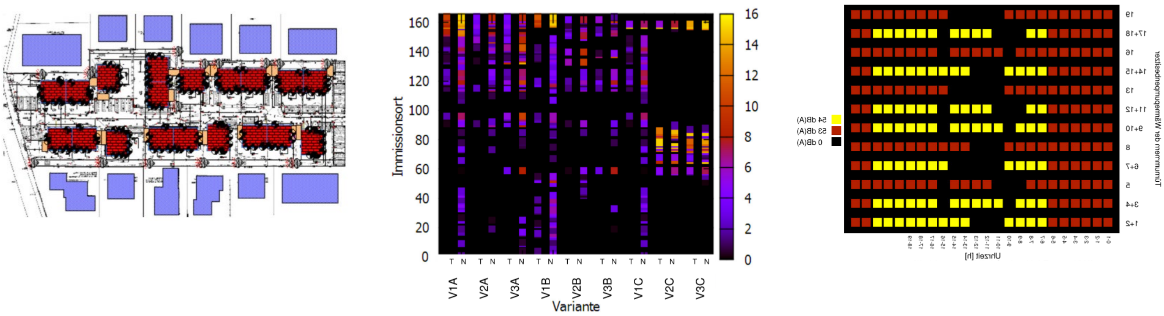
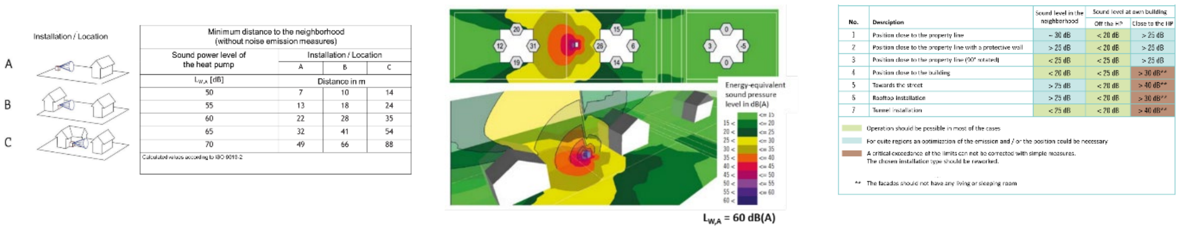


Figure 7-4: Sound field emission studies with multiple heat pump.

The report additionally is working on the analysis of unit placement, indoor & outdoor sound propagation. This includes the description of different installation locations and linked sound pressure maps showing the propagation of noise in various scenarios. A table summarizing the sound pressure level, which can be expected depending on the heat pump position is given.



Minimum distance to the neighborhood (without noise emission measures)			
Sound power level of the heat pump	Installation / Location		
	A	B	C
$L_{W,A}$ [dB]	Distance in m		
50	7	10	14
55	13	18	24
60	22	28	35
65	32	41	54
70	49	66	88

Calculated values according to ISO 9613-2

No.	Description	Sound level in the neighborhood	Sound level at own building	
			Off the HP	Close to the HP
1	Position close to the property line	~ 30 dB	< 20 dB	> 25 dB
2	Position close to the property line with a protective wall	> 25 dB	< 20 dB	> 25 dB
3	Position close to the property line (90° rotated)	< 25 dB	< 25 dB	> 25 dB
4	Position close to the building	< 20 dB	< 25 dB	> 30 dB**
5	Towards the street	> 25 dB	< 20 dB	> 40 dB**
6	Rooftop installation	> 25 dB	< 20 dB	> 30 dB**
7	Tunnel installation	< 25 dB	< 20 dB	> 40 dB**

Operation should be possible in most of the cases

For quite regions an optimization of the emission and / or the position could be necessary

A critical exceedance of the limits can not be corrected with simple measures. The chosen installation type should be reworked.

** The facades should not have any living or sleeping room

Figure 7-5: Analysis of unit placement, indoor & outdoor sound propagation

We outline the potential of sound absorption at nearby surfaces and tabulates the reduction effects of various measures taken. Finally common unclever decisions in heat pump placement are visited such as wrong locations, roof installation, the (unforeseen) development of the neighbouring properties, the selection of improper sound absorbing measures and finally the installation of further units in the neighbourhood.

8. Figures Index

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