FAULHABER

Vibrations and Noises in Small Electric Motors

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Thomas Bertolini | Thomas Fuchs

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Measurement, Analysis, Interpretation, Optimization

FAULHABER

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Dedicated to my esteemed teacher,
Prof. Dr.-Ing. Werner W. Freise

Thomas Bertolini, July 2011

Preface

It is hard for us to imagine today's world without electric motors. And their use keeps expanding into ever new areas of our lives. At the same time, we are becoming increasingly demanding when it comes to the acoustic quality of these motors. Manufacturers of motors and other equipment increasingly need to have broad competence in dealing with noise. In fact, success in the competitive marketplace is coming to depend on having such acoustic know-how.

Unfortunately, manufacturers often have a hard time finding solutions to their noise problems. It is often extremely difficult even to describe a problem in a qualified manner. Frequently, this is not even attempted. Instead, people come up with vague statements like "We have a noise problem!" Many times engineers and technical people who are not experts in acoustics are assigned the task of solving this noise problem. This book is intended to help them arrive at an efficient and effective procedure for eliminating irritating vibration and acoustic problems with small electric motors. It is based on real-world industrial experience and is not intended to be an academic text, but rather a practical manual.

Until now there has not been any literature on possible approaches to the analysis and elimination of vibrations and noises in small motors. The purpose of this book, therefore, is to provide users and manufacturers of small electric motors with the basic understanding needed for dealing with noise. In the first chapter we introduce the vibration and noise behavior of small electric motors and present key terms and the relationships between these terms. We then describe various options for reducing noise. Finally, we address the key principles of mechanical vibrations and acoustics needed to achieve success. The remaining chapters deal in detail with measuring vibrations and noises, with the analysis of these measurements, and with the problems associated with noise testing in large-scale manufacturing.

The principal methods are illustrated, together with their advantages and disadvantages. The book concludes with a series of examples from actual industrial practice.

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1 Introduction to the topic

When motors of all types and sizes are evaluated, the main concerns involve the quality of the desired functions, service life, and purchase and operating costs. Side effects such as heating, vibrations and noise are generally undesirable, and they play a very important role in the decision to use a specific motor.

Small motors are usually installed in tight spaces in equipment, so that the heat they produce can be particularly disadvantageous, even when the actual heat output is small, since the surrounding equipment itself often offers little opportunity for removing heat. Since these motors are usually located close to humans and to their ears and sense of touch, the noise and vibrations that they are permitted to generate are significantly less in relative terms than would be permitted for larger motors.

For large electrical machines and motors there are standards that must be used to measure and evaluate noise and vibrations. Considered in isolation, a small motor is barely audible because of its small dimensions. Even the vibrations it produces are generally not found to be objectionable. A small motor does not have a significant impact until it is installed in or on a piece of equipment and then evaluated subjectively, usually only in this installed environment. This subjective evaluation of the noise and vibration characteristics of the motor installed in the equipment must be quantified by means of measurement technology. From this information, specifications for the vibration and noise limits for the motor alone must be derived and agreed upon and then met when the motor is manufactured. The description and derivation of such limits can be very complex in individual cases; therefore test standards like those used for large motors do not make sense for small motors.

Above all, the small motor manufacturer wants to be successful in the marketplace. So he will work to identify the causes of objectionable vibrations and noises in his product and try to eliminate them as much as possible. In

the final analysis the motor is the source of the noise, and the equipment in which it is installed is "merely the loudspeaker." Working together with his customer, the manufacturer must also try to figure out how an anticipated or existing vibration or noise problem can be optimally solved with regard to the entire system in which the motor operates. It may often be easier and more cost-effective to overcome an undesirable vibration or noise along the path from the motor itself to our sense of hearing or touch than it would be to eliminate the problem at the source. Frequently, if certain rules are not followed, the vibrations caused by the motor along its transmission path become "the mouse that roared." This means that the path itself must be analyzed and taken into account.

In the section below we shall first explain some important terms relating to the noises and vibrations produced by small motors. We will then describe the basic principles relating to the transmission paths. This will be followed by a look at the causes of noises and vibrations in small motors, as well as the unique characteristics of such noises and vibrations and the technical options for reducing them.

Before any measurements are made, the vibration behavior of the equipment in which a motor is installed must first be considered conceptually. Systematic measurements and testing derived from this conceptual analysis should then follow to support or reject the conceptual model; but efficiency demands that the brainwork comes first. Only if the measurements are carefully planned and systematic will their consequences be clear. Therefore, a detailed description of acoustic and vibration measurement technology, combined with suitable methods, follows. In this way, it is possible to move from subjective judgments to descriptions based on actual measurements in order to obtain limit values that can be used in the development, manufacturing, application, and quality testing of motors.

1.1 Basics of vibrations, structure-borne noise, and airborne noise

Noise and vibrations are oscillations, namely changes of states or conditions that occur with periodic regularity. We describe an oscillation by the dura-

tion of its period or **frequency** and by the maximum value of its state over time (**amplitude**). There are many kinds of oscillations. In the text below we shall only consider those whose states involve periodic movement in space (**mechanical vibrations**). Such movements can be generated by periodically changing forces, such as those encountered in a crank mechanism (**forced excitation**). But they can also be produced independently by a spontaneous exchange of energy between various energy stores, such as that which occurs with elasticities (stores for deformation energy) and inertial masses (stores for kinetic energy), if these stores can in some way be energized (for example: a swing, bell, violin string, whistle). An oscillation that is controlled by means of these energy stores is called a **natural oscillation**. Its frequency (**natural frequency**) is often determined solely by the properties of the energy stores and not by the energy that is stored in them. Because of the unavoidable **damping** that causes energy to be "lost" (for example: the energy that is consumed when materials change shape), a natural oscillation cannot be maintained indefinitely unless a source of appropriate energy is applied further to the system. In addition to the supply of energy by means of an appropriate forced excitation, a supplier of power that is constant over time can, as a result of the properties of an oscillatory system, produce self-controlled, i.e. self-generated, natural oscillations (such as friction-caused vibration, whistling noises).

In addition to its manifestation over time, a natural oscillation also always has a specific spatial distribution of its peaks and valleys (**natural waveshape**), which can only form if no disruptive forces can act upon the oscillatory system via mounting systems. A natural waveshape can simultaneously have identical motion excursions throughout the waveshape; when this occurs, it has the **mode** $r = 0$. This is the case, for example, when a ring is "breathing," i.e. when it changes its diameter in all directions in the same way over time.

If a natural waveshape is characterized by simultaneously having two points of maximum but opposite oscillation excursion amplitudes (**oscillation antinodes**) and two points at which no motion is present between them (**oscillation nodes**), it has a mode of $r = 1$. The flexural oscillation of a motor illustrates how mounting systems can hinder the formation of waveshapes: radial vibrations are hindered if not prevented at the bearing points. However, the shaft can vibrate radially. In this case at two opposite points on the circumference of the rotor the radial movement is maximal but is equal to zero at the

points offset by 90°. Seen in the circumferential direction, the mode is $r = 1$. But in the axial direction, this is not the case because only a single antinode can be seen between the bearings. The bearings hinder the free formation of $r = 1$, and therefore radial alternating forces (vibratory forces) occur in the bearings. If the rotor were not hindered by the bearings, it would vibrate on both ends in the direction opposite to the vibrations in the middle of the rotor, and two nodes would form between its ends. If the bearings were placed at the nodes and the outer ends of the shaft were allowed to oscillate freely, the free rotor flexural oscillation would not be impeded. In fact, $r = 1$ also establishes itself when hindering bearings are present because the bearing system, with its elasticity and mass, simply needs to be included as part of the oscillatory system. The corresponding flexural natural frequency, of course, is different from that of the rotor without bearings.

The oval deformation of a tubular metal package is characterized by the mode $r = 2$ (two full waves along the circumference, four antinodes, four nodes, etc.). Antinodes and nodes can change location relative to time, but the deformation pattern, in other words the mode, remains intact (see also chapter 4.6, p. 50 f.).

A configuration with more than two possibilities for storing various forms of energy (such as potential and kinetic energy) is capable of having a number of different natural oscillations with various corresponding natural frequencies and waveshapes, i.e. with a number of different antinodes and nodes superimposed on each other in time and space.

In industrial applications, forced excitation and the ability to engage in natural oscillation usually occur in combination. If a forced excitation acts upon a system that is capable of natural oscillation, the system continuously oscillates at the rate of the forced excitation, but also temporarily at its own natural frequencies, provided that the forced oscillation is not applied precisely at the node of the corresponding natural waveshape. If the forced excitation frequency and the natural frequency are identical, the term **resonance** is used. In this case the corresponding waveshape movements can become very large if damping is small. If the natural frequency is less than the excitation frequency, **operation above resonance** occurs. In the reverse case **operation below resonance** occurs. Given that industrial equipment generally has a number of natural frequencies (and natural waveshapes) and given that the

forced excitation can also contain a number of frequencies, operation above resonance and operation below resonance are often both done. Tuning can greatly alter the effect of forced excitations.

The point at which the forced excitation is applied or how the excitation is spatially distributed also plays an important role! If the forced excitation is only applied at a single node of a waveshape, then no resonance occurs (at least not theoretically), even if the natural frequency of this waveshape and the frequency of the forced excitation are the same. In general: **For resonance to occur, the frequencies and the modes of the waveshape and the forced excitation distribution must be the same.** If forced excitation is applied at specific points outside of the natural waveshape node, it is important to consider that such excitation at specific points is comprised of excitation components that theoretically can have an infinite number of modes, so that a resonance can result with any one of these components.

Oscillations from and in solid and liquid materials are referred to as **structure-borne noise**; in the case of liquids, they are also referred to as **fluid-borne noise**. We humans experience such structure-borne noise with our sense of touch. Vibrations from and in gases (air) are referred to as **airborne noise**. We perceive this noise with our sense of hearing and, at very low frequencies and high amplitudes, also with our sense of touch.

While structure-borne noise is transferred as a result of the elasticity of solid substances, airborne noise or fluid-borne noise is transferred as a result of the compressibility of gases and liquids; the losses that result from the change in the shape of objects (material damping) and friction in the air weaken the transfer along the transfer path.

Our sense of hearing has a large range of perception across approximately six powers of 10 of the amplitude of the vibration between the (lower) **threshold of perception** (**auditory threshold**) and the **threshold of pain**, and it has a large frequency range with highly frequency-dependent sensitivity. In addition, there are also individual evaluations of frequency mixtures (**spectrum**) and changes in noises over time, as well as the **stereo effect** that results from hearing with both ears, along with overlapping individual psychological effects. Therefore, the effect of airborne noise on human beings cannot be completely emulated by using measurement technology (see also chapter 5, p. 72).

1.2 Transmission paths

A **transmission path** is the path over which the oscillation is carried from the generator to our sense of hearing or touch. It extends from the source of the oscillation as structure-borne noise to the surface of the source (in this case the motor) and from there as structure-borne noise across the mounting system into the driven device and from there as structure-borne noise to the outer surface of the device housing. At the same time, airborne noise is produced at the surface of the motor. In the equipment in which the motor is installed, this noise strikes the inside surface of the housing and from there is transferred as additional structure-borne noise through the housing wall to the outer surface of the equipment. If the housing has openings, the "inner" airborne noise also escapes directly to the outside and is added on to the airborne noise that comes from the oscillating housing surface and is radiated inward and outward. With our sense of touch we feel the vibrations on the housing surface and, in some cases, also the vibrations that are transferred from the housing surface into a cover or mounting system. The airborne noise reaches our sense of hearing.

The effect of the properties of the structure-borne noise path (distribution of masses and elasticities) to the surface that generates airborne noise is shown by the comparison of the noise produced by a small motor in installed condition (light blue spectrum) and in uninstalled condition (dark blue spectrum) (Fig. 1.1): An uninstalled motor is almost always significantly quieter because of its smaller surface area and its vibration distribution.

Therefore, the airborne noise that it generates is small and is largely short-circuited acoustically because its dimensions are small in comparison with the wavelengths of the sound oscillations. The fact that it can often still be heard is due to the large range of sensitivity provided by our sense of hearing. In this case, though, we only hear the high-frequency noise components since these components short-circuit acoustically somewhat less than the low-frequency noise components, to which our sense of hearing is also less sensitive. This is one of the reasons why the sound impressions produced by an uninstalled motor and those produced by an installed motor usually differ significantly. Another reason is that the system used to mount the motor in the equipment only transfers the structure-borne oscillations that are present at the mounting point. Therefore, the nature of the mounting system plays a very important role in acoustics and vibration!

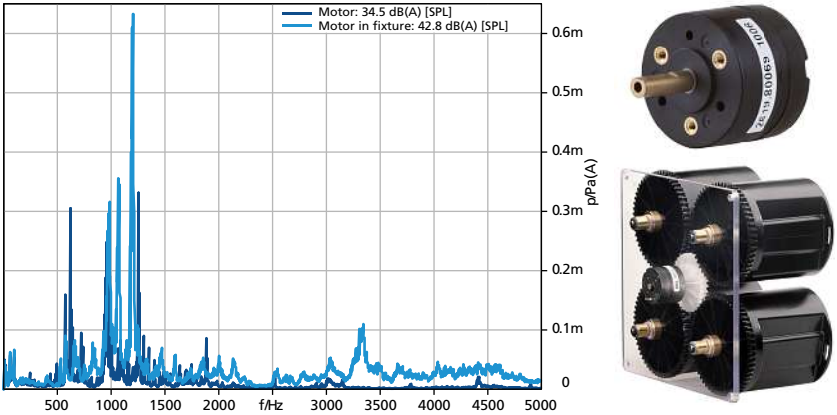


Fig. 1.1: Effect of installation situation on noise radiation characteristics

In the example shown in Figure 1.1, the motor is mounted directly on an acrylic plate without any decoupling. The center of the acrylic plate was chosen as the mounting location – theoretically the worst case. The antinode of the first natural oscillation mode with the corresponding lowest natural frequency of the acrylic plate is located here. Applying a vibration at this point therefore has the maximum effect and will generate the largest conceivable noise radiation – which is by no means desirable.

The transmission path to our sense of touch is limited to the transfer of vibrations by means of structure-borne noise to the surface that is touched. Airborne noise and structure-borne noise often work together; an everyday example is an electric shaver whose motor vibrations are felt on our skin and heard with our ears. Measures to reduce undesirable airborne noise are generally the most effective when they are applied to the noise source and/or the path of structure-borne noise, in the latter case in particular at the motor mounting area.

Airborne noise is also produced under variable flow conditions in blowers, sirens, etc., causing localized fluctuations in air pressure. If we want to reduce this noise, we must design the blower in such a way that the flow of air ahead of, inside, and downstream of the blower is as homogeneous and constant over time as possible. Airborne noise caused by oscillating surfaces can be

reduced by decreasing the vibration excursions at the surface of the object, by reducing the surface area, and by utilizing time-phase positions of the surface vibrations that are different at different locations (natural waveshape distribution and therefore acoustic short-circuiting). Such a short-circuit can also be achieved by openings in housings. Oscillating perforated panels are therefore quiet compared with solid panels.

Exactly the opposite approach is used in designing a loudspeaker. The coil that vibrates in the magnetic field is small (Fig. 1.2) and therefore, like a small motor, it can only produce a small amount of airborne noise on its own. However, because it is desired that as much airborne noise as possible be produced, the coil is mounted as rigidly as possible on a speaker diaphragm. In order for this diaphragm to vibrate as much as possible in the same phase at all locations on its surface, it is designed to be lightweight and essentially rigid and therefore is not flat but rather conical or concave and is flexibly and elastically attached to the housing at its outside edge, and only there. In order to prevent acoustic short-circuiting between the front and back of the diaphragm, the loudspeaker is installed in a relatively large wall (acoustic wall) or a housing.

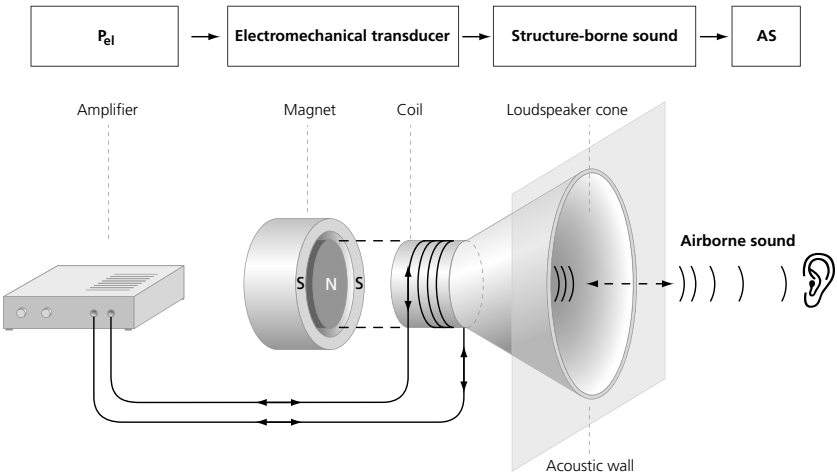


Fig. 1.2: Transmission path in the production of sound in a loudspeaker

2 Causes of vibrations and noises in small motors

In each motor undesirable forces, torques, and motions are unavoidably produced in addition to those that are desired. Undesirable fluctuations (oscillating torques) are superimposed on the desired electric motor torques. This results in oscillating rotational movements. Radial forces caused by imbalance and magnetic effects cause radial movements. Friction forces that fluctuate over time occur in bearings and on sliding contacts and cause undesirable movements. When gearboxes are installed in equipment, undesirable rotational oscillations are caused by the gears. All these movements constitute structure-borne noise, and they are transferred as such to the vibrating surfaces of the motor. Figure 2.1 shows typical examples of the main locations at which vibrations and noises can be produced in an electric motor.

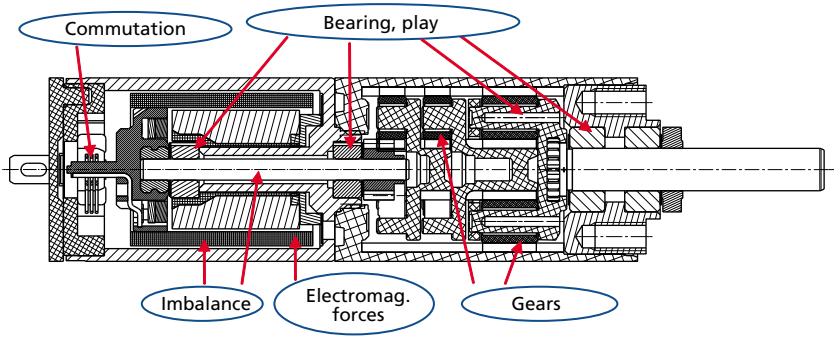


Fig. 2.1: Sources of vibrations and noises in an electric motor

2.1 Electromagnetically induced vibrations

The surface and the shaft of each electrical machine also move in undesirable ways. This is the result of the heteropolar concept that is always used for electrical machines for functional reasons and that has significant advantages over unipolar machines. As a result of this concept, the distribution of the magnetic energy density that is needed to generate the machine's torque in the air gap between the stator and rotor of an electrical machine must unavoidably fluctuate in time and space. Slotting (the distribution of the magnetic permanence for the magnetic air gap), the design of the permanent magnets, the distribution of the current-carrying winding, the way the current is applied, and the curve of current over time are all factors in these fluctuations. On the peripheral surface of the rotor and on the air gap side of the stator, the fluctuation of the spatial energy density distribution over time causes fluctuating tangential and radial forces that are applied at different locations – so-called **force excitations** (Fig. 2.2).

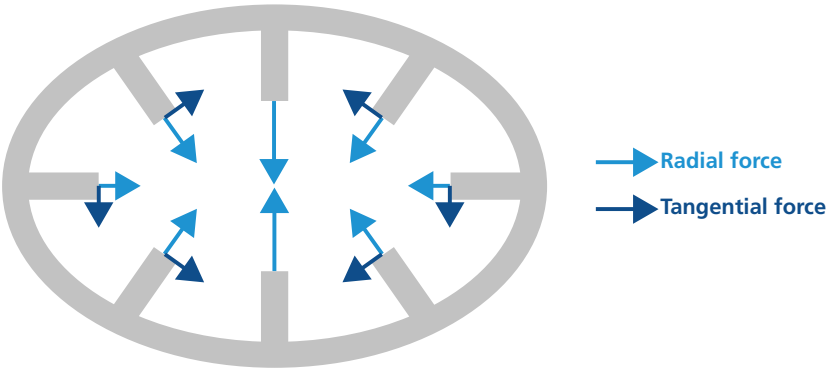


Fig. 2.2: Force excitations on the stator of an electrical machine

If the motor is suitably designed, the fluctuations in the tangential forces within the torque have very little effect because, when they are added up along the circumference, the large number of local fluctuations offset each other

over time. In this way, oscillating torques can be kept low. This also applies to the cogging torques of motors equipped with permanent magnets. Locally caused tangential force fluctuations, however, cause tangential flexural deformation oscillations – for example, on the teeth of the metal packages of the stator and rotor (if teeth are present). The distribution of the radially oriented magnetic tensile forces causes radial movements and deformations of the stator and the rotor. Depending on the nature of the spatial distribution of these forces, this can be a resting or circumferential shift or a vibratory movement. In the stator and rotor, these shifts in movements are out of phase, and the overall center of gravity is maintained. Likewise, a flexing of the shaft and a possible deformation of the bearing cover with a modal order number of $r = 1$ (see also chapter 4.6 "Oscillation modes," p. 50 f.) or an oval, triangular, or polygonal elastic deformation of the stator ($r = 2, 3, ...$) occur. In this case the rotor body is barely deformed at all because of its greater radial flexural rigidity. The deformation of the stator can vary over time depending on the force distribution, and it can be circumferential, in other words it can oscillate. As a rule, movements having several modal order numbers r overlap at the stator. Generally, with small motors only movements with $r = 1$ are objectionable because small motors are small and inherently have sufficient rigidity. (With large motors movements with $r > 1$ are noticeable, although this is seldom the case in those with $r > 4$.)

Movements with $r = 1$ are usually the result of an eccentric position of the rotor in the stator, which can be static, for example when the rotor bearing locations in the stator are eccentric or circumferential, as would be the case with a bent shaft. They are harmonic or pulsating, usually with a number of harmonics, and they act in a spatially fixed radial direction and/or are circumferential, in other words the direction in which they are applied rotates. In the worst case, nearly all the frequencies of the movements lie within the audible range. Additional details on the causes of the radial and tangential forces of the various modal order numbers r can be found in the specialized technical literature on electrical machines.

In motors that utilize permanent magnets (DC motors, stepper motors, synchronous machines), a "cogging torque" occurs in addition to the desired torque. This cogging torque occurs in a no-current state as a result of the variations of magnetic permanence that depend on the position of the ro-

tor. In operation the cogging torque is superimposed on the desired torque, which does not change over time, and results in torque fluctuation or ripple. As a consequence of the law "for every action there is an equal and opposite reaction," an alternating force acts on the stator. Depending on the mounting system used, this force can manifest itself as a vibratory force at the motor mounting site.

2.1.1 Effect of electronic commutation

In electronically commutated motors, the currents applied to the individual motor phases are turned off and on by electronic circuitry. Depending on the pattern of these phase currents over time, radial and tangential forces act upon the stator teeth, and forces act upon the electrical conductors in the motor phases. The latter case occurs in particular with unslotted motors with air gap windings. In the case of rectangular current patterns over time, local rectangularly shaped forces are also produced over time, causing abrupt oscillatory excitations at the stator. This excites oscillations at the switching frequency (fundamental oscillation) of the current phases, as well as their harmonics.

With ideally rectangular currents and ideally symmetrical motors, the torque remains constant over time. However, if in reality there are geometric asymmetries, differences in the phase inductances and resistances as well as differences in the electronic circuitry, then current time gaps or overlaps will result in the individual phases. These will cause torque fluctuations and additional vibratory excitations at the stator. The resulting noises are often called "commutation noise," and their importance must not be underestimated! The use of electronic control circuitry that works with sinusoidal current patterns results in poorer motor efficiency, but it does eliminate or at least greatly reduce this type of noise.

2.1.2 Special characteristics of stepper motors

"Electronically commutated motors" (EC motors) as well as "stepper motors" belong to the class of synchronous machines excited by permanent magnets. While EC motors typically are conceived for continuous drive tasks, stepper motors are designed to be able to execute defined rotational angles with as

little deflection as possible and to be able to maintain these defined stepper positions; they should have the highest possible holding torque. However, this requires that the magnetic circuit be dimensioned in such a way that a high cogging torque is produced (no-current torque in the phases); and this cogging torque overlaps the holding torque (current flow into the phases). Therefore, in addition to stator deformations, torque fluctuations and sudden torque changes are to be expected, just as with EC motors. For all intents and purposes, sinusoidal currents are not used with stepper motors. So-called "micro-stepping," in which the phase currents can be turned on and off in fine increments in order to achieve additional intermediate positions between two hold positions, is frequently used.

Reluctance stepper motors do not require any permanent magnets and therefore do not have any cogging torque. However, these motors ("switched reluctance motors") utilize very low air gap widths; 50 µm is typical. This causes extremely high radial forces, which massively deform the stator and cause high noise generation.

2.2 Mechanically induced causes of vibrations in small motors

2.2.1 Vibrations in bearings

In **plain bearings** such vibrations are the consequence of the intermittent mechanical contact between the shaft and the bearing surface. When a lubricant film carries the load between the shaft and bearing around the full circumference, such contact does not occur, nor any noise. However, contact does occur upon start-up, when the radial forces on the bearing are excessive (belt drive, gear, air gap field), when the shaft and/or the bearing sleeve is not round or if they are crooked, if the sintered bearing surfaces do not have sufficient porosity, if the shaft running surface is too smooth, or if there is not enough lubricant in the bearing (mixed friction). The consequence is vibrations at the roughness peaks of the bearing surfaces, which are dependent on the elasticity of the shaft or bearing, at numerous frequencies (frequency band in spectrum) in the audible range. The frequency of rotation and multiples

thereof are particularly pronounced. The amplitudes of the higher frequency vibrations are often simply modulated, which causes the friction noises to become particularly objectionable. As the lack of lubricant worsens, the mechanical contacts increase, and mechanical friction occurs and wear increases sharply. This results in natural vibrations in the bearing (with the self-induced addition of energy from the bearing friction), which is perceived as a squeaking or squealing noise. In plain bearings, the deformations are primary and the resulting deformation forces are secondary (**displacement excitation**).

In **roller bearings**, rolling elements – in small motors these are generally ball bearings – roll in inner and outer rings, usually with mechanical contact. They are surrounded by a layer of lubricant, which has a slight cushioning effect and enlarges the contact surface somewhat. If the rolling elements and the ring raceways are sufficiently round and undamaged, only a broad frequency band (spectrum) of oscillations results as a consequence of the circumferential elastic deformations at the contact points caused by the compression forces (**force excitation**) and as a consequence of lubricant movements (displacement excitation). The lubricant dampens the vibrations. So if not enough lubricant is present or if the viscosity of the lubricant is incorrect, vibration will increase ("metallic, hard-sounding" noise) because the pressures increase at the contact points. Material fatigue increases. Raceway damage, in particular that caused by axial overloading of the bearings, also leads to faster material fatigue.

Radial ball bearings have radial bearing play due to how they are manufactured and how they operate. If this radial play is "compressed to zero" as a result of an improper elastic axial preload on the outer or inner ring, the bearing balls cannot transfer the radial forces optimally. The installation fits of the bearing and the assembly quality play an important role in this case. At certain rotational speeds the balls actually run in synch instead of in a circular, wavy path. As a result, self-induced axial vibrations, which are heard as howling noises, occur in the bearing bracket.

Thus in bearings, we mainly encounter motions that generate elastic forces, which in turn lead to vibrations, as well as forces that lead to oscillating motions. This means that there is displacement excitation as well as force excitation, so that both must be considered when the individual case is being analyzed in theoretical terms.

2.2.2 Vibrations at sliding contacts

Many motors have sliding contacts to transfer current between static and rotating parts. If these contacts consist of closed **slip rings** and metal brushes or carbon brushes (Fig. 2.3), the vibration excitations that are to be expected are theoretically identical to those of nonlubricated plain bearings. The slip rings themselves, because of their large mass and rigidity compared with metal and carbon brushes, are not excited to the extent that acoustically perceivable vibrations are produced. How the metal or carbon brushes vibrate is

Fig. 2.3: Examples of various carbon and metal brushes

highly dependent on their inherent elasticity and mass and also on the system used to hold them (spring, box, hammer brush holder) as well as the slip ring bearing surface. The resulting noise from the metal or carbon brushes themselves is very low because of their small surface area. What is perceived, though, are the structure-borne vibrations that are transferred to the holder and the area close to it since these parts have many relatively large radiating surfaces and prevent acoustic short circuits. Eccentricities in the slip ring, damage on its bearing surface, and deviations from the circular shape become noticeable at the rotational frequency and multiples thereof. Some modulation occurs, as is typical of bearings. If the bearing surface of the slip ring is

very smooth or becomes very smooth in operation, and if too little humidity is present in the air, an air cushion forms between the bearing surfaces of the carbon brush and the slip ring as the rotational speed increases, and this reduces the coefficient of friction. The coefficient of friction, which decreases as the frictional velocity increases, and the resulting frictional force, which also fluctuates, allow natural vibrations to occur in the carbon brush, just as vibrations are produced when a violin bow is drawn across the violin strings. These vibrations have a natural waveshape, which is related to the carbon brush holder, and a corresponding natural frequency. They are heard as a loud whistling or squealing noise in the range from 3 to 10 kHz. If there are smooth spots in some locations and the motor is running at a slower speed, they are heard as a chirping noise. This effect is similar to brake squeal. It has the same cause and can be controlled by design features. If metal brushes are used instead of carbon brushes, undesirable natural vibrations usually do not occur.

A **commutator** – i.e. a slip ring consisting of segments that are located next to each other in the peripheral direction, that are insulated from each other, and that are connected to the ends of the coil sections induced by the air gap field – has further vibrations in addition to those referred to above. The segments are embedded in an insulating polymer which is inherently elastic and plastic. This means that even when the commutator running surface is machined to the finest precision, sawtooth-like radial steps from segment to segment (in the 1-µm range and smaller!) cannot be avoided. This excites radial vibrations in the metal or carbon brushes, similar to what happens when car tires travel across the washboard roads. At low rotational speeds, the brushes can follow the unevenness in the commutator. The vibrations then have the segment frequency f_L of

$$f_L = n \cdot z_K \qquad \text{[Hz]} \tag{2.1}$$

where n is the motor rpm and z_K is the number of commutator segments and multiples thereof. At higher speeds, the metal or carbon brushes lift off the commutator for some of the time, depending on their characteristic elasticity, characteristic mass, and spring contact pressure. This tends to occur more with carbon brushes because of their higher mass (metal content in low-voltage motors!) than with metal brushes. The latter also have the advantage that

they do not all lift at the same time. In addition, sound radiation is very low because of their small surface area. These advantages are countered by their shorter service life (small wear mass, no graphite lubrication).

2.2.3 Vibrations caused by imbalance

It is impossible in manufacturing technology to completely align the main axis of inertia and the axis of rotation of a rotor. While it is possible to keep this deviation low, the typical high speed of small motors causes substantial centrifugal forces – i.e. radial forces about the circumference having the modal order number $r = 1$ at the bearing points (hence: force excitation). Depending on elasticity, this results in larger or smaller circumferential radial deflections of the bearing point. Depending on the deviation of the main axis of inertia from the axis of rotation (parallel or angular) there can be radial forces that, in spatial terms, are in phase or out of phase, or combinations of the same, and as a result motions caused by these forces (ranging from shaking to tumbling motions). A parallel mismatch between the axes is referred to as **static imbalance**; angular mismatch **is referred to as dynamic imbalance** (see chapter 9.2, p. 153 ff.).

2.2.4 Vibrations with geared power transmissions

There are two main causes of vibrations when gears are involved. The first is that gears are never so precise that there are no tangential or radial vibrations; the second is that, as the teeth roll over each other, the momentary gear ratio will change because of the changes in the ratios of the radii. With large gears that have many teeth, several teeth are usually involved in the power transfer, so that with sufficient play and elasticity the variations between the radii are evened out, just as with helical gears. Small motors, however, have small gears with a small number of teeth – in other words, fewer teeth are simultaneously engaged (overlap), and seldom by means of helical gearing. Therefore, even a slight fluctuation in the ratios of the radii makes itself quite noticeable. This excites rotational vibrations on the shafts and radial vibrations at the bearings equal to the product of the rotational frequency and the number of teeth and multiples thereof. These vibrations are occasionally modulated by

the rotational frequency or its multiples, for example when plastic gears warp and become unround. The vibrations are a form of displacement excitation, since they are primarily forced motions that result in pressures or forces. The magnitude of the vibrations depends on the elasticity of the teeth and other elasticities, on the moments of inertia, and on the play between the teeth and the lubricant cushion. Plastic gears are advantageous with regard to noise because of their elasticity, low mass, and material damping, but their dimensions often increase to an undesirable extent as temperatures and humidity rise. Therefore, they must be designed to have play. Thus, they are not suitable for relatively high loads. With multiple-stage gearboxes and planetary gears, the variations in the radii largely compensate for each other in the overall transmission system.

During the service life of a motor, wear frequently causes play and backlash between the gear teeth.

2.2.5 Play and backlash

When geared systems are turned on and when direction reversals occur, play and backlash cause impactive noises in the drive train. During operation, friction between moving parts can cause stochastic, pulse-like noises as well as continuous noises. Continuous noises usually contain many harmonics. Play and backlash can occur at and between many components. Typical examples are the flange mounting of the motor, gearboxes that are not under load, inadequately tensioned ball bearings, or a loose ball bearing inner ring on a shaft.

Play between the teeth of a gearbox causes vibrations (displacement excitation). The sides of the gear teeth strike each other abruptly and spring back elastically. The resulting repetition frequencies and amplitudes depend to a very large extent on the amount of play, and they become very distinct under no-load conditions, during start-up, or when rpm changes occur. The vibrations disappear as load increases. These vibrations can be minimized by means of good lubrication and elastic teeth that provide material damping (plastic). In large motors that have play or worn metallic gear teeth, such vibrations can be extremely objectionable.

3 Options for reducing and optimizing noises

The motion vibrations that occur in motors are transmitted from the motor surface as airborne noise and at the shaft and motor mount as structure-borne noise. From here they are transmitted further into the environment as structure-borne noise and to some extent also as airborne noise.

To reduce noise, the general approach is to interrupt the noise chain from source to transmission path to ear, or better still, to reduce the generation of noise directly at the source. If this is not possible, one can at least try to make the noise more pleasant and less objectionable. This activity is referred to as "noise optimization," even though such a systematic effort to alter the noise being produced is not an optimization in the true sense. In the individual case, the measures that are taken must always take economic considerations into account.

3.1 Insulation and deadening

Sound barriers can be achieved by means of acoustic and vibration insulation. Insulation must be clearly distinguished from **damping**, in which vibrational energy is converted to frictional heat. In solid bodies, such frictional heat is caused by the movement of molecules or relatively large particles against each other inside the body, but also in material that is installed outside of the equipment (such as foam, nonwoven materials, elastomers) and that exhibits substantial internal friction. In order for such material also to have a damping effect, it must be attached at vibration antinodes on the surfaces, in other words at those locations where the vibrations cause the greatest deformations in the material. Such material is often called insulating material, even if it does not insulate but rather dampens.

With liquids, viscosity has a damping effect, but only in combination with compressibility or a significant deformation of the liquid "body" in its container. The damping ability of water is very low because of its low internal friction and because it is nearly incompressible. Oil also is nearly incompressible. Its significantly higher viscosity only has a damping effect when it is forced through narrow openings, in other words changes shape, such as is the case in shock absorbers. Gases are compressible, but because of the large distances between their particles, they have very low internal friction and therefore are low in damping ability. However, if gas flows through barriers such as narrow screens, filters, foam, or if the gas particles oscillate within such barriers, then the friction, sound pressure, and sound particle velocity increase, so that the sound volume decreases and the sound energy is "rubbed" into heat. Thus, screens, filters and similar devices are sound dampers.

In general, insulation and damping measures must be considered separately. They often are mutually exclusive. However, in many cases insulation and damping measures make sense, provided that they are carried out at the right location.

3.2 Reducing sound radiation

The radiation of airborne noise to the outside can be significantly reduced by encapsulating the entire motor. In this way the propagation of the airborne noise is limited and is "dammed in"; in this case we speak of sound insulation. Resonances that are caused by the encapsulation itself, as well as cavity resonances, must be taken into account.

Frequently, the entire motor cannot be completely encapsulated because of the connection to the unit being driven or to the environment. In the case of openings, attention must be given to achieving a desirable mismatch of the (sound) wave resistances that are involved in the transmission of the sound, and to avoiding objectionable reflection. Covering the capsule with sound insulating material helps to prevent cavity resonances and helps to dampen the vibrations of the capsule itself. In the case of sound dampening – in contrast to sound insulation – sound energy is "destroyed" (converted to friction). In the case of small motors, covering the capsule with insulating

material often is not possible for space-related reasons, or it is not done for cost reasons.

Surfaces that radiate noise can be quieted by providing them with openings. This reduces the size of the radiating surface (compare this with a loudspeaker diaphragm!) and also fundamentally changes the vibrational behavior of the surface. In this way, natural frequencies can be shifted and objectionable vibration modes together with their nodes and antinodes can be rendered harmless. Additional reinforcements or bracing can have an effect similar to that of openings.

3.3 Reducing sound and vibration transfer

In theory, the measures referred to for reducing sound radiation also apply to preventing motor vibrations from being transferred across the shaft and the motor mounting system in the equipment (or in the environment). However, there are some general "recipes for success": Mounting should always be done as close as possible to the node sites of the most objectionable vibratory motion. As a rule, the most important nodes are found near the bearings. The vibratory motions that are still present there should transfer the lowest possible force oscillations. In other words, the mounting system in the direction of vibration should be as flexible as possible and have as little damping as is possible with the given motor application and other conditions (such as transport shocks). If the force oscillations are small, then they only result in small vibratory motion in the attached parts, even with small masses (lightweight equipment). More equipment mass, in particular in the area where the motor is mounted, is often advantageous. Of course, the oscillating mass of the motor, the elasticity of the mounting system, and the mass of the equipment in the vicinity of the motor mount must be matched to each other so that no resonance with the objectionable motion frequency results. The system must be tuned so that the resonance is below the operating point.

Figure 3.1 illustrates the principle of operation above resonance in greater detail. If excitation near the resonance frequency is found in a device, this will be accompanied by correspondingly high vibration amplitudes. One approach is to make the device more rigid in order to shift the natural frequency

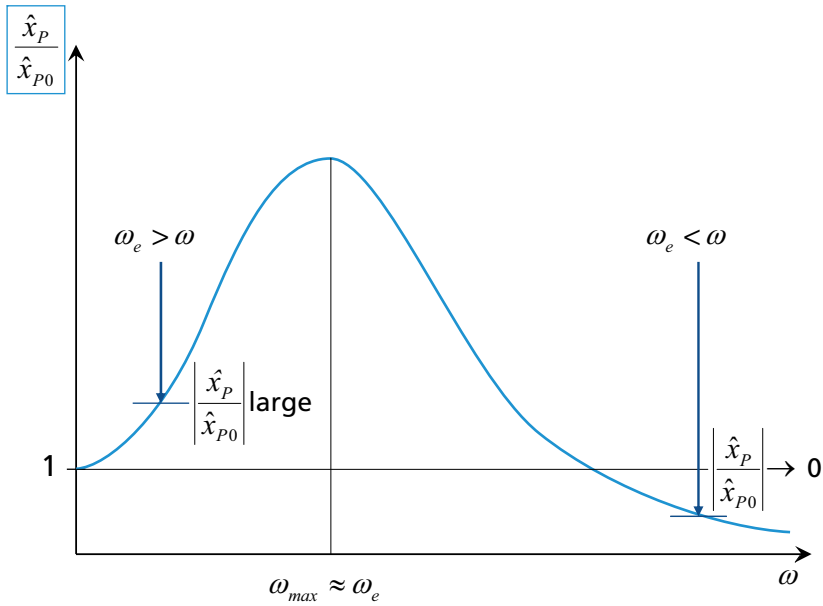


Fig. 3.1: Frequency dependency of the oscillatory amplitudes in operation above resonance

of the device upward to higher values. However, this step often is not successful because the resonance frequency cannot be increased sufficiently. This is because the natural frequency only increases at the square root of rigidity. Moreover, excitation in the form of vibration can still be expected in the range $\omega < \omega_e$. If the opposite approach is taken and the natural frequency is reduced so that $\omega > \omega_e$, the amplitude of the vibration can become very small. This is referred to as operation above resonance. In actual practice, it is important to make certain that the resonance frequency is passed through quickly.

When the motor is mounted, it is important to make certain that it is preferably connected to the structure at vibration nodes – in other words, that the corners of a housing and not the middle of a surface are used as the mounting points. If the middle of the plate is nevertheless used for space-related

reasons, a suitable decoupling measure must be provided. This is illustrated in the layout shown in Figure 1.1 (see chapter 1, p. 19).
Figure 3.2 shows a physical model that corresponds to the example from Figure 1.1: A motor is rigidly mounted in the center of an acrylic plate of mass *m* and flexural rigidity c_m. This motor generally produces **displacement excitation** on its surface. Therefore, the motor produces oscillatory excursions in the outward direction, and thus the mass of the motor is not significant for the excitement of the plate. The middle of the mounting plate experiences displacement excitation of $x(t)$. The mounting plate possesses a natural frequency that is dependent on its mass and flexural rigidity. In order to decouple or isolate the motor, it must be flexibly attached to the mounting plate. To accomplish this, an additional elasticity c_{added} is interposed between the displacement excitation and the excited mass m, thus creating a **single-degree-of-freedom system**. This decoupling causes the motor to act like **a loudspeaker without a diaphragm**; sound radiation is correspondingly low, and the low frequencies are short-circuited. If the decoupling were not present, the case would

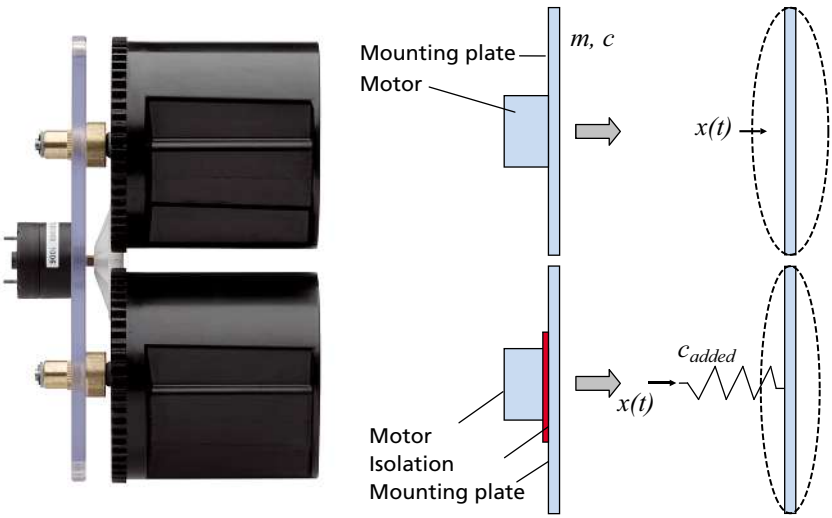


Fig. 3.2: Diagram of motor isolation

be that of a **loudspeaker coil equipped with a diaphragm**: The airborne noise level increases and the perceived spectrum changes in the direction of lower frequencies.

In practice, a layer of elastomer is normally used for decoupling, for example an elastomeric foam mat. This mat not only has an elastic (spring-like) effect, it also produces a certain amount of damping. However, complete decoupling does not occur in this damped system. Instead, a certain portion of the excitation is transferred through the damping components. Therefore, a damping-free decoupling system using spring elements is a better option relative to the decoupling effect. However, this is not always practicable for strength reasons or because of space limitations. Increasing the mass of the mounting plate is another measure that is suitable for "quieting" the system. The mathematical basis of decoupling is described in detail in chapter 4.7 (p. 52 ff.)

Additional measures, such as using active mass dampers or shock absorbers, are also theoretically conceivable. However, these options for reducing the transfer of noise and vibrations will not be described further here because they are not relevant to small motors.

3.4 Reducing sound and vibration excitation

Ideally, noise and vibrations are best reduced where they are created: at the source. However, in electric motors, forces and torques are needed, and they often unavoidably include undesirable components (oscillating torques, cogging torques, etc.), which cannot be completely avoided. The wide variety of motor concepts that is currently encountered with their varying operating principles also leads to various noise excitations. Asynchronous motors behave differently than synchronous motors (including electronically commutated motors and stepper motors!), and DC machines behave differently than the piezo motors. Noise and vibration excitation therefore can often only be minimized by very carefully selecting a suitable motor and a suitable motor size.

With motors that are operated in electronic equipment, noise is often excited by undesirable motor current timing patterns. The switching of

phase currents in electronically commutated motors can result in radial force excitations (torque fluctuations) which are so strong that the motor radiates airborne noise at the switching frequency. Likewise, pulse width modulation (PWM) in the audible range can be objectionable, as is experienced with inverter drives.

Noise and vibrations are also frequently excited by motor imbalance. In many cases this problem cannot be eliminated with very small motors because the balancing that would be needed often cannot be as precise as the motor would require. Since the small motors are also operated at very high speeds of 10,000 to 100,000 rpm, the noise produced by the imbalance can be extremely objectionable.

The noise level of a small electric motor naturally increases with its operating speed. Therefore, reducing the motor's operating speed is one way to successfully reduce the noise level it causes. As a rule, drive motors – in contrast, for example, to blowers – are not operated continuously at a fixed speed, but rather are used for positioning tasks. During a positioning cycle, the motor passes through all speeds from zero up to the maximum operating speed relatively quickly. Resonances can be produced in the process. In particular, very objectionable noise levels can result when operating speeds that are close to resonant frequencies are present for a relatively long time. A slight change in the motor's rotational speed may be helpful here if the task that the motor is performing and the positioning velocity permit. However, caution should be observed in making rpm changes: With small motors, rotational speed variations are much greater than with larger motors because friction and component tolerances have a much more pronounced effect in relative terms. Thus, a slight change in the rotational speed of a small motor may be helpful with one motor, but harmful in the next motor of the same model from a different production lot.

Thus, the only options that remain for reducing undesirable noise and vibration excitation directly at the source are changing the design of the motor, modifying the electronic control system if present, reducing imbalance, or changing the operating point.

In recent years, there has been increasing discussion of **active noise compensation**. In everyday language, this is a sound that is generated artificially in order to cancel out existing sound (noise) by means of destructive inter-

ference. To implement this method, it is necessary to generate a signal that corresponds to the undesirable sound but that is precisely out of phase. Literature in English also uses the terms **Active Noise Reduction (ANR)** or **Active Noise Cancellation (ANC)**. This approach is already being used successfully. It is used, for example in earphones to eliminate environmental noises by combining these noises with the earphone signal, but out of phase. Pilots, for example, no longer need to be exposed to the jet engine whine. The approach is severely limited, however, if the out-of-phase airborne noise cannot consistently be supplied to the human ear at the required volume. Numerous attempts at using this technique in automotive engineering thus far have only met with moderate success.

3.5 Optimization: efforts to systematically influence sound and vibration excitations

Noise optimization is defined as systematically altering the acoustic quality of the noise in order to achieve the best possible value. The **acoustic quality** represents the degree to which the requirements relating to the totality of individual requirements in an auditory event are met. Acoustic quality includes three different types of independent variables:

- Physical independent variables: These variables relate to the sound field.
- Psychoacoustic independent variables: These variables describe how the sound is perceived in human hearing.
- Psychological independent variables: These variables relate to how the sound is evaluated when it is heard.

Ideally, the objectionable sound field is reduced to the threshold of hearing. Since this often is not technically or economically feasible, one can try to affect the noise and alter it in such a way that undesirable and objectionable noise components are eliminated. Chapter 5 shows in detail how such undesirable and objectionable noise components can be evaluated (see p. 72 ff.).

If one wishes to define acoustic quality, the perception threshold of various soundscapes must be taken into account, above all in conjunction with

undesirable and objectionable noises. Here, it is likewise important not to establish any requirements that are not technically or economically feasible. These thresholds of perception were investigated by ZWICKER, and they are stated more precisely in [5]. While the human ear can register a sound level difference of 0.2 dB with a 1-kHz tone at a sound pressure level of 80 dB, at 20 dB it can only distinguish a difference of 1 dB.

4 Mechanical oscillations

4.1 Basic principles of oscillations

An oscillation refers to a process in which physical variables (distances, velocities, forces, etc.) repeat at identical time intervals **and** in which a periodic change of energy occurs between at least two different energy stores. With mechanical oscillations, these stores consist of a store for kinetic energy (moved mass) and a store for potential energy (tensioned spring). With electromagnetic oscillations, magnetic energy from an inductance is changed into electrical energy, which is stored in a capacitor. In each case, oscillation cannot occur unless there are two different energy stores. Therefore, temperature does not oscillate, but rather fluctuates, because there is only a single form of energy store for thermal energy, namely heat capacity.

The simplest oscillation can be described by means of harmonic functions (cyclical functions). For example, the following applies to the movement of a mechanical oscillator

$$x(t) = \hat{x} \cdot \cos(\omega t + \varphi) \tag{4.1}$$

where $\hat{x}$ is the amplitude of the oscillation, $\omega = 2\pi f$ = the angular frequency, f is the frequency, t is the time, and φ is the zero-phase angle at $t = 0$. Linear oscillators are those whose defining variables (mass, inductance, spring constant, etc.) do not depend on time nor on the momentary value of the amplitude of their oscillation. Therefore, the defining variables of linear oscillators are constant.

The degree of freedom determines how many coordinates are needed to describe the oscillation. In a mechanical oscillator, there are a maximum of three translational degrees of freedom and three rotational degrees of free-

dom. In engineering applications, the description of a single degree of freedom is generally sufficient.

The number of elements of an oscillator refers to the number of energy stores of the same type that it possesses. For example, a mechanical oscillator consisting of one mass and one spring is a single-element oscillator. A multiple-element oscillator is present when a number of masses are elastically connected to a number of spring elements. In the case of an infinite number of finely distributed masses and springs (such as elastic beams subject to mass loads), the number of elements is infinite. To make a system easier to understand, one should always try to simplify multiple-element systems and, if possible, reduce them to two-element systems.

4.2 Single-element linear oscillatory system

The basic behavior of an oscillatory system can easily be studied using the example of an apparatus consisting of an inertial mass m and a spring having a spring constant c and damping d. An arrangement of this type is shown in Figure 4.1. If one assumes the same counting direction z for the displacements x, velocities $\dot{x}$, and accelerations $\ddot{x}$ of the mass m, and also for all forces, the spring force F_c is:

$$F_c = -c \cdot x \tag{4.2}$$

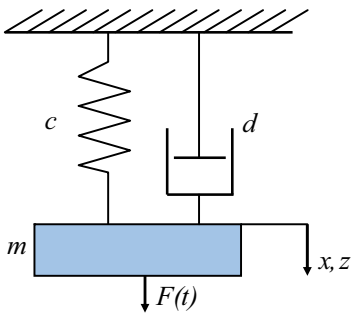


Fig. 4.1: Single-element spring-mass system with damping

For the damping force F_d, which is assumed to be proportional to speed:

$$F_d = -d \cdot \dot{x} \tag{4.3}$$

According to Newton's second law, the total of the forces applied to a mass m is equal to their acceleration times their mass. Thus,

$$F_c + F_d + F(t) = m \cdot \ddot{x} \tag{4.4a}$$

If Equation 4.4a is restated, one obtains:

$$m\ddot{x} + d\dot{x} + cx = F(t) \tag{4.4b}$$

Equation 4.4b is a linear differential equation, which is a second-order equation because of the twofold derivation of the deflection component x over time. There are two parts to the solution of this second-order differential equation.

One part is referred to as the general solution x_h of the homogeneous differential equation. This solution is found when we set the right side of Equation 4.4b equal to zero. Thus, no external exciting force is present. In this case, the homogeneous solution of Equation 4.4b with

$$m\ddot{x}_h + d\dot{x}_h + cx_h = 0 \tag{4.5}$$

allows us to draw conclusions as to the natural oscillation behavior. The second part of the solution is referred to as the particular solution x_p, which describes the reaction to the interfering function $F(t)$.

$$m\ddot{x}_p + d\dot{x}_p + cx_p = F(t) \tag{4.6}$$

If the force that is applied is harmonic

$$F(t) = F_{max} \cos(\omega t + \zeta) \tag{4.7}$$

where ζ is the zero-phase angle of the force excitation, a harmonic solution is to be expected using

$$x_p = \hat{x}_p \cos(\omega t + \varsigma) . \tag{4.8}$$

Here, the frequency of the oscillation corresponds to that of the excitation. However, a phase shift can occur in the deflection at a phase angle of ς. From the solution of the homogeneous differential equation, we then obtain the natural angular frequency ω_e as

$$\omega_e = \sqrt{\frac{c}{m} - \left(\frac{d}{2m}\right)^2} . \tag{4.9a}$$

The system will oscillate at this frequency if it is deflected from its resting position and then left to respond on its own. Without damping ($d = 0$), the undamped natural angular frequency ω_0 becomes:

$$\omega_0 = \sqrt{\frac{c}{m}} \tag{4.9b}$$

With periodic excitation from outside $F(t)$ corresponding to Equation 4.6, an amplitude $\hat{x}_p$ which is dependent on the frequency of excitation ω results:

$$\hat{x}_p = \frac{F_{max}}{\sqrt{\left(c - m\omega^2\right)^2 + \left(d\omega\right)^2}} \tag{4.10}$$

This curve is shown in Figure 4.2, standardized to the deflection x_{p0} at $\omega = 0$. A frequency-dependent phase shift occurs between the exciting force $F(t)$ and the deflection x. With very low frequencies, excitation and deflection are

(nearly) in phase; at very high frequencies, they are out of phase. At the natural frequency without damping ω_0 the phase shift is $\varphi - \xi = \pi/2$.

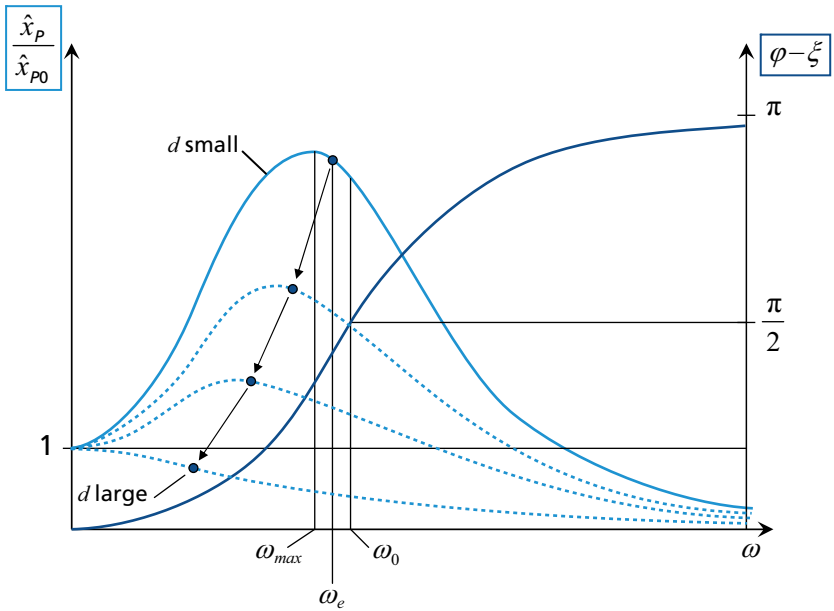


Fig. 4.2: Amplitude and phase curve of a single-element oscillatory system

As damping d increases, the natural angular frequency ω_e moves down to lower frequencies in accordance with the relationship

$$\omega_e = \omega_0 \sqrt{\frac{1 - d^2}{4mc}} \tag{4.11}$$

with a simultaneous reduction in the amplitude. The maximum of the oscillation amplitudes is found at the frequency

$$\omega_{max} = \omega_0 \sqrt{\frac{1-d^2}{2mc}} \tag{4.12}$$

as naturally expected slightly below ω_e, and with increased damping d it also moves toward lower frequencies.

4.3 Multiple-element linear oscillatory system

A two-element linear oscillatory system is obtained (Fig. 4.3) if once again a similar system with mass m_2, a spring having the spring constant c_2, and damping d_2 is applied to the mass of the system shown in Figure 4.1. If it is assumed that the positive counting directions are the same z, the force equilibrium can be established for each individual mass:

$$F_{c_1} - F_{c_2} + F_{d_1} - F_{d_2} + F_1(t) = \Sigma F = m_1 \ddot{x}_1 \tag{4.13a}$$

$$F_{c_2} + F_{d_2} \qquad\qquad + F_2(t) = \Sigma F = m_2 \ddot{x}_2 \tag{4.13b}$$

If the terms for spring forces and damping forces are inserted in the equation and sorted, one then obtains:

$$m_1 \ddot{x}_1 + c_1 x_1 + c_2 (x_1 - x_2) + d_1 \dot{x}_1 + d_2 (\dot{x}_1 - \dot{x}_2) = F_1(t) \tag{4.14a}$$

$$m_2 \ddot{x}_2 + c_2 (x_2 - x_1) \qquad\qquad + d_2 (\dot{x}_2 - \dot{x}_1) = F_2(t) \tag{4.14b}$$

This second-order differential equation system can be solved using the same method as in the previous section. From the solution of the homogeneous components (external force excitations are zero), **two** natural frequencies with **two** decay time constants are obtained: In general, the number of natural frequencies in an oscillatory system corresponds to the number of elements it has (number of energy stores of the same type).

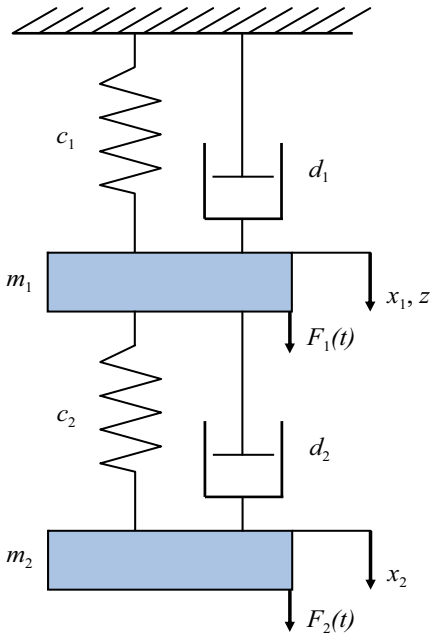


Fig. 4.3: Two-element linear oscillatory system

In the example shown above, the exciting forces $F_1(t)$ and $F_2(t)$ were assumed in order to derive the basic relationships. This case occurs frequently, and is referred to as force excitation. Another case that often occurs is so-called displacement excitation, at which the curve of displacement x is specified. This is the case, for example, with a crank mechanism or with a vibrating surface ("loud" motor) that is adjacent to a potential radiating surface.

In addition to force and displacement excitation, oscillations can also be caused by variable parameters, which thus far have been considered to be unchanging. These oscillations are then referred to as parameter-excited oscillations. The most well known parameter-excited oscillation is a playground swing. The swinging oscillation occurs due to the periodic change in the center of gravity of the system. In addition, varying coefficients of friction,

damping values, dimensions, and elasticities can lead to parameter-excited oscillations.

4.4 Multiple-element torsional oscillation system

In motor technology, oscillations of motor surfaces and components are frequently encountered, as are torsional oscillations in the motors themselves. A typical case is an arrangement consisting of a motor, gearbox, and load. The rotating elements of such a system can be described as inertial moments J_1 to J_3. These moments are linked by elastic shafts to the torsional rigidities c_{d1} and c_{d2} and to each other. The torques M_1 to M_3 are applied to the inertial moments, causing an angular acceleration $\ddot{\varphi}_1$ to $\ddot{\varphi}_3$. As shown in Figure 4.4, this arrangement can be represented and treated as a three-element torsional oscillator.

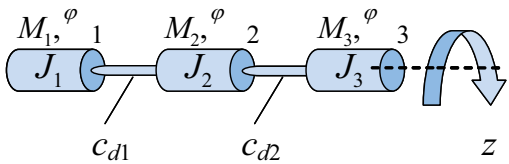


Fig. 4.4: Three-element torsional oscillatory system

If the simplifying assumption is made that the system is free of damping, it then follows for the torque equilibria:

$$c_{d1}(\varphi_2 - \varphi_1) + M_1 = J_1\ddot{\varphi}_1 \tag{4.15a}$$

$$-c_{d1}(\varphi_2 - \varphi_1) + c_{d2}(\varphi_3 - \varphi_2) + M_2 = J_2\ddot{\varphi}_2 \tag{4.15b}$$

$$-c_{d2}(\varphi_3 - \varphi_2) + M_3 = J_3\ddot{\varphi}_3 \tag{4.15c}$$

For simplification purposes, we can assume that the arrangement shown in Figure 4.4 is symmetrical, so that $c_{d1} = c_{d2} = c_d$ and $J_1 = J_3$. The solutions to the homogeneous differential equations – in this case without driving torques – are harmonic oscillations. Corresponding to the three elements that are present in the system, there are three natural frequencies at

$$\omega_1 = 0\,, \tag{4.16a}$$

$$\omega_2 = \sqrt{\frac{c_d}{J_1}}\,, \tag{4.16b}$$

$$\omega_3 = \sqrt{\frac{c_d\left(J_1 + \frac{J_2}{2}\right)}{J_1\left(\frac{J_2}{2}\right)}}\,. \tag{4.16c}$$

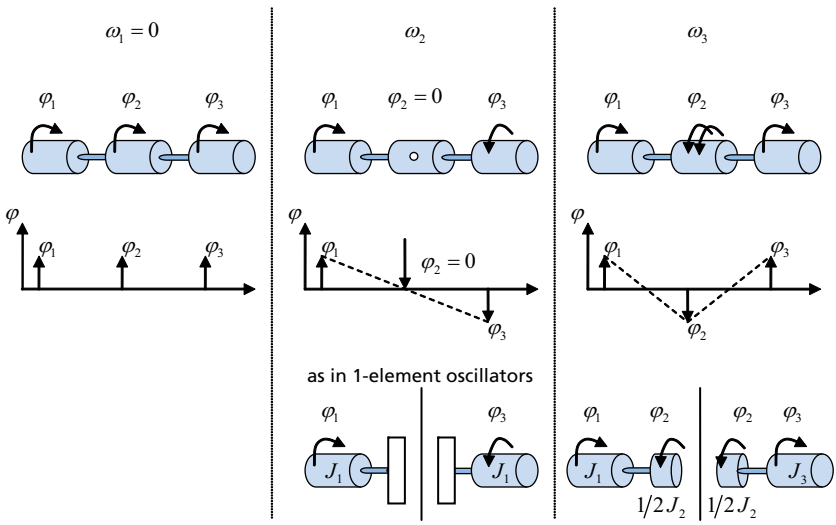


Fig. 4.5: Possible oscillatory shapes of a three-element torsional oscillator

If these natural frequencies are inserted in Equation 4.15 (a, b, c), we find for ω_1 the condition $\varphi_1 = \varphi_2 = \varphi_3$. This means that free motions at ω_1 can only occur if no relative torsions occur in the system. This is the trivial case of uniform rotational motion. For ω_2 we obtain $\varphi_1 = -\varphi_3$ and $\varphi_2 = 0$. Here, both ends oscillate in opposite directions, and the center section is at rest. For ω_3, $\varphi_1 = \varphi_3$ and $\varphi_2 = \varphi_1\left(\frac{2J_1}{J_2}\right)$. In this case, the ends oscillate in the same direction, and the center oscillates in the opposite direction. Different combinations or additional combinations of natural frequencies and oscillation modes are not possible for this example. This is illustrated by Figure 4.5.

4.5 Oscillations in systems having distributed masses and elasticities

The examples used thus far each consider concentrated (point) masses and moments of inertia, and separate spring elements. These simplifications help us to arrive at a basic understanding of oscillations and oscillatory systems. Because such simple models are easy to grasp, one should always try to reduce complex systems to such a simple model.

However, in actual practice, masses and elasticities are never concentrated at a single point nor can they always be separated from each other. Instead, they are distributed in space. An analytical calculation of the natural frequencies of such systems is generally only possible for simple objects, such as a flexural beam subjected to mass loading, the oscillation of a string, or for the oscillations of a circular ring. Today, complex geometric systems are generally modeled and calculated with the aid of the finite element method. The results that are obtained with these calculations usually correlate very well with measurements made on the components.

4.6 Spatial distribution of oscillations: oscillation modes

Each natural frequency of an oscillation possesses a specific spatial distribution of amplitudes. This "natural waveshape" is referred to as the "normal mode of oscillation" or simply "oscillation mode." Since a number of oscillation modes often occur simultaneously, the individual oscillation modes together with their corresponding natural frequencies must be determined from the overall oscillation behavior so that the cause of the oscillation can be found.

An extensive analysis of this oscillatory behavior is referred to as modal analysis. Here, the spatial distribution of oscillations with their corresponding natural frequencies and possible damping factors as well as the phase position of the various oscillations relative to each other are determined.

The individual vibration modes can be distinguished from each other based on the number of vibration nodes that occur. The number of nodes is divided by two to yield the so-called modal order number r of an oscillation distribution. Thus, the modal order number also gives the multiple of the fundamental oscillation. The fundamental oscillation at $r = 1$ has two oscillation nodes and two oscillation antinodes.

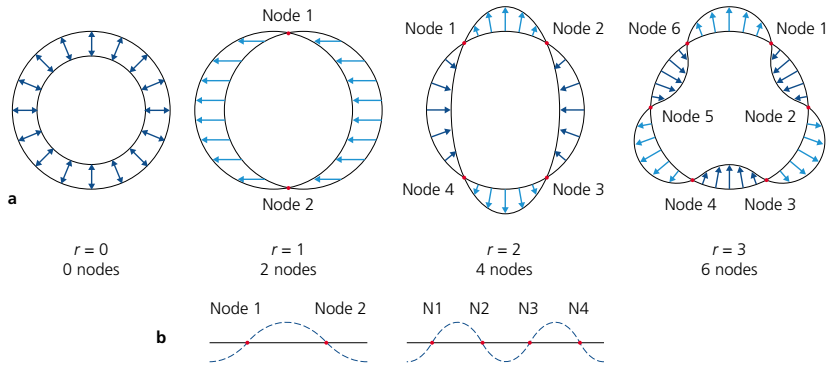


Fig. 4.6: Oscillatory modes of a circular ring (a) and of a beam (b)

Figure 4.6a shows some possible vibration modes of a circular ring. The oscillation pattern having the modal order number $r = 0$ (also referred to as zero oscillation) describes the circular ring when it is "breathing." $r = 1$ describes a movement such as that which can be caused in a rotor by an imbalance – which then leads to a circumferential oscillation. In Figure 4.6b the natural waveshapes of a beam are also indicated schematically for $r = 1$ and $r = 2$. It is important to consider that the natural waveshapes must be seen as existing free in space (in other words without clamping systems or other constraints). The total of all mass forces is always zero, so that the torques that act in an outward direction are also equal to zero. The natural oscillation does not cause either a translational nor a rotational movement; only local flexural deflections occur. A beam for $r = 1$ therefore has an antinode in the middle, out-of-phase antinodes on the ends, and two nodes between them. The distribution of mass in the beam determines precisely where these two nodes will lie. If the ends of the beam are clamped in place (corresponding to the conditions of a rotor that is held by bearings at its two ends, see chapter 1.1, p. 14 ff.), then forces result on these ends. Depending on whether the clamping system is elastic or rigid, the free natural waveshape will dissipate to a greater or lesser extent. Oscillation modes can also be produced via excitation under such constrained conditions, but they are no longer natural oscillations!

An example of the natural waveshapes of a brush holder is shown in Figure 4.7. It is now possible to obtain such calculations with clear representations of the vibration modes fairly easily from the CAD data contained in parts drawings.

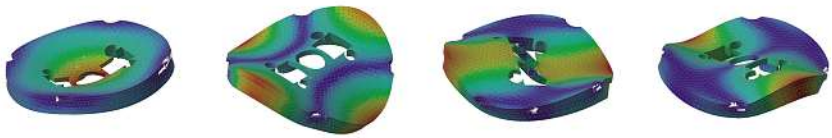


Fig. 4.7: Oscillatory modes of a brush holder

4.7 Mathematical/physical analysis of the decoupling of motors

There are two general "recipes" that work well for decoupling motors:

- Mounting and driving should always be done as close as possible to the node sites of the most objectionable vibratory motion in the vibration source. As a rule, the most important nodes are found near the bearings.
- The vibratory motions that are still present there should transfer the lowest possible force oscillations. In other words, the mounting system or the drive coupling in the direction of vibration should be as flexible as possible and have as little damping as is possible with the given motor application and other conditions (material characteristics, transport shocks). The force vibrations caused by the introduced motions will then be small and, even when the masses of attached parts are small (lightweight equipment), the force vibrations will only produce small vibratory movements in these parts. This is important because these parts direct their acoustic behavior in an outward direction.

More equipment mass, in particular in the area where the motor is mounted, is often advantageous. Of course, the oscillating mass of the motor, the elasticity of the mounting system, and the mass of the equipment, in particular in the vicinity of the motor mount but also elsewhere, must be matched to each other so that no resonance occurs at the undesirable frequency. The system must therefore be tuned so that the resonance is below the operating point.

In greatly simplified but basically sufficient terms, the above mounting situation can be formally described with a simple oscillation model in which the properties of the masses, elasticities, and damping effects are each considered individually independent of the movements. The system is also assumed to be linear, so that the forces and movements, instead of being considered as totalities, can be considered individually as sinusoidal components, and the resulting components can be superimposed on each other. Let the motor in this system be characterized by mass m_0, upon which a component of the oscillation-exciting forces f_0 or an oscillatory movement x_0 acts (force or displacement excitation, Fig. 4.8). We use c and d to describe the

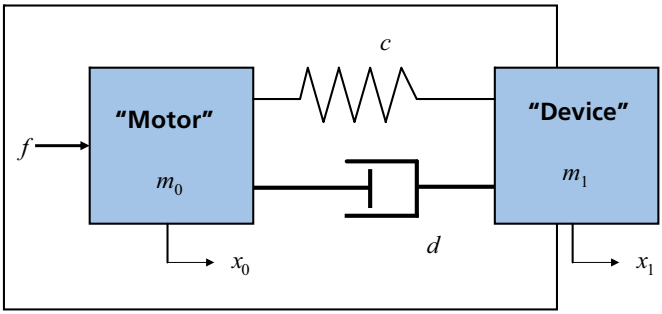


Fig. 4.8: Model of a device with motor as a two-mass oscillator

elasticity and the damping effect of the motor mounting system in the device (provided that they act in the direction of the oscillatory movement x_0). Let m_1 summarize the mass of the device in simplified terms, and let x_1 be the movement components of the device resulting from the excitations f_0 and x_0, respectively. In the simplest case this is the device surface area radiating the airborne noise. Combining the components into a single mass m_1 is permissible if the device is sufficiently rigid. Of course, the device can also be described as a configuration consisting of distributed masses and elasticities, or as a multiple-element oscillation model. But then one would not obtain any clear guidance as to effective ways to mount the motor. A more complex and less straightforward approach is only needed if the rigidities in the device are low and the results of the simpler method are uncertain, or if the more complex method is the only way to obtain any results at all.

The elasticity c results in a spring force $c \cdot x$; the damping factor d results in a damping force $d \cdot \dot{x}$ that is assumed to be proportional to the oscillatory velocity; the mass results in an inertial force $m \cdot \ddot{x}$ that is proportional to the oscillation acceleration. Thus, the following motion equations are obtained for the masses m_0 and m_1:

$$c \cdot (x_0 - x_1) + d \cdot (\dot{x}_0 - \dot{x}_1) + m_0 \cdot \ddot{x}_0 = f \tag{4.17a}$$

$$c \cdot (x_1 - x_0) + d \cdot (\dot{x}_1 - \dot{x}_0) + m_1 \cdot \ddot{x}_1 = 0 \tag{4.17b}$$

Using

$$f = \hat{f} \cdot \cos \omega \cdot t, \tag{4.18}$$

$$x = \hat{x} \cdot \cos(\omega \cdot t - \varphi), \tag{4.19}$$

$$\dot{x} = -\hat{x} \cdot \omega \cdot sin(\omega \cdot t - \varphi), \tag{4.20}$$

$$\ddot{x} = -\hat{x} \cdot \omega^2 \cdot \cos(\omega \cdot t - \varphi) \tag{4.21}$$

these equations can be solved for x or f. The complex form with the imaginary unit $j = \sqrt{-1}$ can be used:

$$f = \mathrm{Re}\left(\hat{f} \cdot e^{j \cdot \omega \cdot t}\right) \tag{4.22}$$

$$x = \mathrm{Re}\left(\hat{x} \cdot e^{j \cdot \omega \cdot t} \cdot e^{-j \cdot \varphi}\right) \tag{4.23}$$

$$\dot{x} = \mathrm{Re}\left(j \cdot \hat{x} \cdot \omega \cdot e^{j \cdot \omega \cdot t} \cdot e^{-j \cdot \varphi}\right) \tag{4.24}$$

$$\ddot{x} = \mathrm{Re}\left(-\hat{x} \cdot \omega^2 \cdot e^{j \cdot \omega \cdot t} \cdot e^{-j \cdot \varphi}\right) \tag{4.25}$$

This allows us to solve the motion equations for x_0, x_1, and φ_0, φ_1. We now see that the lowest possible values of x_1 can be obtained by properly selecting c, d, and m_1. The diagram in Figure 4.8 and the motion equations already show that d always causes x_1 to increase. Therefore, the damping d must be kept as small as possible. If we ignore the summand d in the motion equations, we get very simple, clear results for the motions

$$x_0 = \frac{f}{m_0 \cdot \omega_{e0}^2} \cdot \frac{1 - \left(\dfrac{\omega}{\omega_{e0}}\right)^2}{\left(\dfrac{\omega}{\omega_{e0}}\right)^2 \cdot \left[\left(\dfrac{\omega}{\omega_{e0}}\right)^2 - \left(\dfrac{\omega_{e1}}{\omega_{e0}}\right)^2\right]}, \tag{4.26}$$

$$x_1 = \frac{f}{m_0 \cdot \omega_{e0}^2} \cdot \frac{1}{\left(\frac{\omega}{\omega_{e0}}\right)^2 \cdot \left[\left(\frac{\omega}{\omega_{e0}}\right)^2 - \left(\frac{\omega_{e1}}{\omega_{e0}}\right)^2\right]} \tag{4.27}$$

using

$$\omega_{e0}^2 = \frac{c}{m_1} \tag{4.28}$$

and

$$\omega_{e1}^2 = \frac{c\left(m_0 + m_1\right)}{m_0 \cdot m_1} = \omega_{e0}^2 \left(1 + \frac{m_1}{m_0}\right) \tag{4.29}$$

as typical (natural) frequencies that are dependent on c, m_0, and m_1.

If we assume that the motor executes a movement x_0 at the mounting point that is introduced by the motor ("low-end" excitation), the excitation frequency f and the motor mass m_0 are not included in the calculation, and from the second motion equation we obtain the housing deflection:

$$x_1 = \frac{x_0}{(1 - (\omega/\omega_{e0})^2)} \tag{4.30}$$

In the following, the ratio of the housing mass m_1 to the motor mass m_0 is (arbitrarily) assumed to be $m_1/m_0 = 2$. Then, according to Equation 4.29 $\omega_{e1} = \sqrt{3}\omega_{e0}$. The diagram that appears in Figure 4.9 below shows the normalized curves for the drive motion

$x_0 \cdot \frac{m_0 \omega_{e0}^2}{f}$, the housing motion $x_1 \cdot \frac{m_0 \omega_{e0}^2}{f}$, and the ratio of the oscillation amplitudes $\frac{x_1}{x_0}$ over $\frac{\omega}{\omega_{e0}}$. We see that the natural angular frequency ω_{e0} must always be significantly lower than the angular frequency $\omega = 2\pi f$ that is part of the excitation frequency of the motor in order for x_1 to become small.

If ω_{e0} is too high, an undesired amplification of the motion amplitude x_1 can even occur in some circumstances, and consequently an intensification of the noise. ω_{e0} becomes small in accordance with Equation 4.28 if the elasticity c is as small as possible. This is the case with flexible mounting systems and the largest possible mass m_1 in the vicinity of the mounting point.

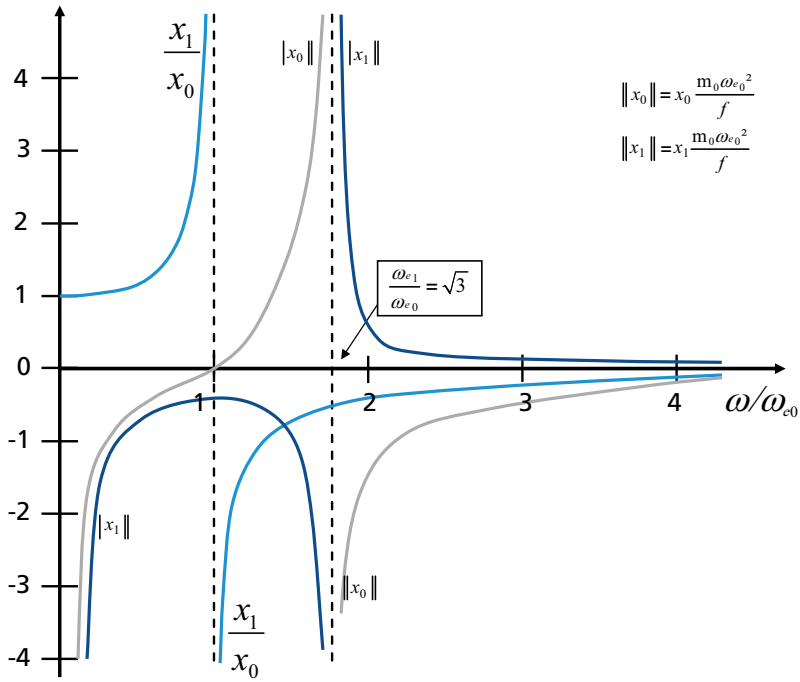


Fig. 4.9: Motion deflections of a two-mass oscillator as a function of frequency (d=0, m_1/m_2=2)

In general, meeting this requirement is sufficient because the introduced displacement excitation x_0 mainly comes from the motor. However, ω_{e1} must also be kept low if in special cases the force component F acts on the mounting system instead of x_0. This leads to the further requirement that

m_1 must not be too large relative to m_0 since $\omega_{e1}^2 = \omega_{e0}^2 \cdot (1 + m_1/m_0)$. In this special case, it is best if both masses are of similar magnitude and are sufficiently large.

Similar requirements apply to the case of rotational oscillation excitations transferred from the drive via its mounting system and the shaft. With the simplifying but useful assumptions referred to above, the same oscillation model will apply, in which the masses m are simply replaced by the mass moments of inertia J and the elasticity c is replaced by the torsional elasticity c_d, and the static and rotating components are considered separately from each other. The static and rotating inertial moments J_0, J_1 of the motor and the drive end, and the torsional rigidity c_d of the mounting system and the output shaft train must therefore act in such a way that their natural frequency is lower than the frequency of the strongest undesirable rotational oscillations, and they definitely must not be close to the resonant frequency. Taking damping d into account does not change these simple consequences. Damping reduces the deflections at the poles with ω_{e0} and ω_{e1} to finite values, ensuring larger amplitudes at frequencies above the poles. The former is advantageous in the case of slow start-ups in which many excitation frequencies in the motor pass through resonance points (poles). However, if start-up is sufficiently fast, the resonance increases remain low, even without damping d, because there is not enough time to generate high amplitudes. Above the poles, any damping d that is present always has a deleterious effect on the desired oscillation reduction or isolation. Unfortunately though, compromises are often necessary and unavoidable.

5 Basic acoustic concepts

This chapter provides an introduction to acoustics. It explains how sound is propagated and describes the most important characteristics of sound, the basic principles of human hearing, and the auditory metrics that are frequently used in acoustics.

5.1 How sound is propagated

In physical terms, sound is made up of waves. This means that sound consists of oscillations that can change over time and that spread out or propagate in space. This propagation can occur in solid, liquid, and gaseous media. Depending on the propagation medium, the following terms are used:

- structure-borne noise in solids,
- fluid-borne noise in liquids, and
- airborne noise in gases.

While the waves that occur in gases and liquids are solely longitudinal, in solids, transverse or longitudinal waves can occur. Sound can be transferred

Infrasound	0 < f < 20 Hz	Seismic activities, waves (seasickness), communication between whales, elephants
Audible sound	16 < f < 16 kHz	Human voice
Ultrasound	16 kHz < f < 1.6 GHz	Medical ultrasonic devices, plastic ultrasonic welding, sonar, cleaning
Hyper sound	1 GHz < f	Aeronautics and aerospace, motors

Table 5.1: Sound frequency ranges

from one medium to another. The frequency range in which sound can occur is normally divided into four areas (Table 5.1).

In chapter 4.7 (see p. 53), for purposes of simplification a model using two essentially rigid masses m_0, m_1 and a mass-less elasticity acting between them c is used to emulate structure-borne noise. This is theoretically correct and serves the purpose for the case that is described. In solid and liquid substances and in gases, however, elasticities and masses are finely distributed, so that they have a combined effect throughout the substance. This fact must always be considered in the general case of the propagation of sound and vibrations.

For example, if an alternating, i.e. oscillating, force is applied to a single point on the surface of the object, elastic oscillating deformations will be produced in the object beginning at this point. This oscillating deformation occurs at a deformation velocity that is also oscillating. The changes in this oscillating deformation over time create oscillating mass forces in the moving particles of mass. These mass forces react as pressure variations to oppose the excitatory pressure of the elastic forces. However, they also are radiated outward in all directions into the object, where they affect the elastic stresses that have also been transmitted to these locations. If such an object did not possess any mass, the elastic oscillating stresses that are generated by the outwardly radiating oscillating force would spread out unhindered into the object. Depending on the nature of the object, these forces would act in the form of oscillating forces or oscillating pressures on other points or portions of the surfaces of the object. If, as above, we assume that the object has no mass, these propagated forces would be just as large as the exciting oscillatory force. Only the elastic deformations at the points at which force enters and exits the system are different – a result of their elasticity. This example can easily be applied to liquids and gases, where it is even more obvious. This is so because there is no distinction between the modulus of elasticity and the modulus of rigidity in liquids and gases.

Oscillations occurring within solid and fluid substances (structure-borne noise) or gases (airborne noise) are best described by means of pressure and motion oscillations. Depending on the given excitation, the pressure oscillation can consist of partial oscillations at various frequencies. The partial oscillations are defined below as **sound pressure** (components) p_v by their peak value $\hat{p}_v$, frequency ω_v, and angle φ_v as

$$p_v = \hat{p}_v \cdot \cos(\omega_v \cdot t + \varphi_v) \qquad (5.1).$$

Instead of the oscillatory motion component of the oscillation, its derivative over time, in other words its vibrational velocity v_v, is used. This velocity is also described by the peak value $\hat{v}_v$, frequency ω_v, and angle ψ_v.

$$v_v = \hat{v}_v \cdot \cos(\omega_v \cdot t + \psi_v) \qquad (5.2)$$

This velocity is also referred to as **sound velocity** (component). The effective value of the total of all sound pressure components is referred to as the (total) sound pressure. The (total) sound velocity is the effective value of the total of all sound velocity components.

Sound pressure and sound velocity are propagated outward in all directions in the form of waves having a migration or propagation velocity c_s. This velocity depends on the modulus of elasticity / modulus of rigidity and the density of the medium. In air at 20°C it is 343 m/s; in metals it is about 6000 m/s.

Equations 5.3 to 5.6 describe the relationships that apply. The following equation applies to the speed or **velocity of sound in an ideal gas** c_{Gas}:

$$c_{Gas} = \sqrt{\kappa \frac{p}{\rho}} = \sqrt{\kappa \frac{RT}{M}} \qquad (5.3)$$

Here κ is the adiabatic exponent, p is the pressure of the gas, ρ is the density of the gas, R is the gas constant (8.3145 J/(mol K)), T is the absolute temperature of the gas (in K), and M is the molar mass of the gas.

The following applies to the **velocity of sound in liquid media** c_{FL}:

$$c_{FL} = \sqrt{\frac{K}{\rho}} \qquad (5.4)$$

Here, K stands for the bulk modulus of the liquid, and ρ stands for the density of the liquid.

For the **velocity of sound in solids**, distinctions are made between **longitudinal** (c_{long}) and **transverse waves** (c_{trans}; with the direction of oscillation being parallel or transverse to the direction of propagation):

$$c_{long} = \sqrt{\frac{E(1-\mu)}{\rho(1-\mu-2\mu^2)}} \tag{5.5}$$

$$c_{trans} = \sqrt{\frac{E}{2\cdot\rho\cdot(1+\mu)}} \tag{5.6}$$

Here, E is the modulus of elasticity of the solid, μ is Poisson's ratio of the solid, and ρ is the density of the solid.

Table 5.2 states various sound velocities for some typical gases, liquids, and solids. For solids, the velocity of sound c_{long} is stated.

Medium	**Velocity of sound** *c* in m/s
Oxygen	315
Nitrogen	336
Air	343
Carbon dioxide	268
Helium	970
Fresh water	1440
Salt water	1510
Flexible rubber	50
Hard rubber	2300
Concrete	4000
Stainless steel 1.4571	5720
Aluminum	6320

Table 5.2: Velocity of sound in selected media

The (spatial) wavelength is $\lambda_v = 2\pi c_s / \omega_v$. This gives us the following typical wavelengths in the medium air at a propagation velocity of 343 m/s:

- 20 Hz → 17.15 m
- 1 kHz → 34.3 cm
- 16 kHz → 2.14 cm
- 20 kHz → 1.72 cm

The large wavelength of low frequencies relative to a room in a home is the reason why a bass loudspeaker (woofer) has virtually no directionality in such an environment. Therefore the location of the woofer has no significant effect on how the sound is perceived.

The curve of sound pressure $\tilde{p}_v$ with respect to time and space may be expressed by

$$p_v(x,t) = \hat{p}_v \cdot \cos(x/\lambda_v - \omega_v t - \varphi_v) \tag{5.7}$$

where x is the sound travel movement. For the sound pressure and sound velocity components the following applies:

$$p_v = \underline{Z}(f) \cdot v_v \quad \text{where } \underline{Z} \text{ is the } \textbf{sound wave resistance} \tag{5.8}$$

$\underline{Z}$ has the following actual meaning when damping is not present and the wavefront is "flat":

$$\underline{Z} = Z = \rho \cdot c_s \tag{5.9}$$

Here, ρ is the specific density/mass of the material through which the sound travels. If damping is present, the sound pressure and sound velocity are phase-shifted. Therefore, $\underline{Z}$ is complex and also larger than when no damping is present. Two examples help to explain what we mean by "flat":

Let us assume that a flat panel that is very large relative to wavelength λ is oscillating in phase perpendicular to its surface at $\tilde{v}$. In this case, a flat acoustic wavefront, i.e. flat in the above sense, is emitted from the surface of the panel at least from its middle area. The sound velocity is longitudinal,

in other words it only occurs in the direction of travel. This is also true of the sound pressure in this case. As its distance from the panel increases, the sound wavefront maintains its amplitude if no damping occurs. In the areas along the periphery of the panel, pressure and velocity components that move laterally are encountered. This is due to the absence of oscillatory excitation in the area. A phase shift between $\tilde{p}$ and $\tilde{v}$ occurs.

A sphere of radius r_0, whose surface is oscillating in phase with the same amplitude throughout, is said to "breathe" at the oscillation frequency (mode $r=0$). It therefore emits a spherical and purely radial longitudinal wavefront. This wavefront is also "flat" in the above sense, and if no damping is present, the sound pressure and the sound velocity are in phase. Since the spherical wave radiation surface increases in this case as the distance *r* from the center of the sphere increases, the amplitudes of $\tilde{p}$ and $\tilde{v}$ decrease at $(r_0/r)^2$. If the surface of the sphere oscillates with a mode of $r \neq 0$, pressure and velocity components occur perpendicular to the radial direction (acoustic short circuiting), similar to those which occurred in the flat panel. In general, area limitations (relative to the wavelength λ_ν) and modes with $r \neq 0$ cause a reduction in sound radiation, in some cases combined with directional effects. This phenomenon is referred to as **relative radiated power** or **relative sound radiation**. To determine the relative sound radiation, the sound intensity that is **actually present** at a defined distance from the sound source (a number of wavelengths) is expressed relative to the sound intensity that **could theoretically be achieved** with an infinitely large flat oscillator surface. The mode $r = 0$ is selected as a reference value for oscillators.

The wavelengths of audible airborne noise lie in the range from approximately 20 to 6000 mm. Therefore, the low-frequency sound radiation that occurs with small motors is very low because of their small dimensions. However, it can become objectionable when motors are attached to larger structures.

The product of the momentary values of $\tilde{p}_\nu$ and $\tilde{v}_\nu$ at the same location is the momentary value of the sound power density i_ν at this location

$$i_\nu(t) = p_\nu(t) \cdot v_\nu(t) \tag{5.10}$$

for sound component ν. Its effective value is

$$I_{veff} = \frac{\hat{p}_v \cdot \hat{v}_v \cdot \cos(\varphi_v - \psi_v)}{2}. \tag{5.11}$$

The effective value of the total of all effective values of the sound components at various frequencies

$$I_{eff} = I = \sqrt{\sum_v I_{veff}^2} \tag{5.12}$$

is the (total) sound power density, which is also referred to as the sound intensity and the sound volume. In isotropic media the sound can radiate freely. However, in industrial applications the media are subject to limitations. Air spaces are enclosed by housings; housings have limited wall thicknesses, etc. If airborne noise strikes a wall, reflection will occur. A sound pressure wave p_{Lv} that strikes the wall exerts pressure on the wall. The corresponding sound velocity v_{Lv} causes the wall to move. The ratio of p_{Lv} and v_{Lv} describes the sound wave resistance of the air Z_L:

$Z_L = \frac{p_{Lv}}{v_{Lv}}$; the ratio of the pressure p_W and the velocity of motion v_W into the wall describes the sound wave resistance of the wall Z_W: $Z_W = \frac{p_W}{v_W}$.

The velocity of motion v_W is the velocity into the wall and not the velocity of the wall itself! Therefore, Z_W is the sound wave resistance of the wall material. If $Z_W \neq Z_L$, then reflected sound waves pLr, vLr must be produced in such a way that the following conditions exist at the wall surface:

$$p_{Lv} + p_{Lr} = p_L = p_W \quad \text{and} \quad v_{Lv} + v_{Lr} = v_L = v_W \tag{5.13}$$

Since $Z_L = \frac{p_{Lr}}{v_{Lr}}$ also applies to the reflected components, the sound reflection factor is

$$r = \frac{p_{Lr}}{p_{Lv}} = \frac{(Z_W - Z_L)}{(Z_W + Z_L)} \tag{5.14}$$

and the sound transmission factor is

$$d = 1 - r = \frac{p_W}{p_{Lv}} = 2 \cdot \frac{Z_L}{(Z_W + Z_L)}. \tag{5.15}$$

The relationships also apply with complex values for Z, in other words when there are phase shifts between $\tilde{p}$ and $\tilde{v}$. As Z_W becomes larger relative to Z_L, more sound is reflected and less is transmitted. If the Z_W of housings is significantly larger than that of air, then housings will be able to shield (dampen) sound well because they only transmit the sound in the form of airborne noise at the sound velocity v_W of the wall. However, this approach describes the transfer of sound between media in a way that is highly simplified and therefore is not a reliable way to estimate the sound insulation achieved by housings.

A different model can be used, though, to estimate the sound insulation that is achieved by a housing. For example, if "flat" airborne noise "1" coming from a perpendicular direction strikes a wall that has the specific mass ρ_W and thickness d, and if the sound wave resistance of the wall Z_W is large relative to the sound wave resistance of the air Z_L, the following apply to the side of the wall that is being struck:

$$p_{Lv1} + p_{Lr1} = p_{L1}, \tag{5.16}$$

$$v_{Lv1} - v_{Lr1} = v_{L1} = v_W. \tag{5.17}$$

On the other side of the wall, which is oscillating in parallel at v_W, airborne noise "2" is radiated at p_{L2}, v_{L2}, and Z_L, where $v_W = v_{L2}$. The resulting sound pressure from both sides $p_{L1} - p_{L2}$ accelerates the mass $d\rho\dot{v}_W$

$$p_{L1} - p_{L2} = d \cdot \rho \cdot \dot{v}_W \,, \tag{5.18}$$

from which at $v_W = \dfrac{j\omega \underline{p}_{L2}}{Z_L}$ (5.19)

(represented in complex form with the angular frequency of the sound $\omega = 2\pi f$) the ratio of the sound pressures and sound particle velocities behind and in front of the wall is obtained as

$$\frac{\underline{p}_{L2}}{\underline{p}_{Lv1}} = \frac{\underline{v}_{L2}}{\underline{v}_{Lv1}} = \frac{1}{\left(1 + \dfrac{j\omega\rho m}{2Z_L}\right)} \,, \tag{5.20}$$

expressed in complex form, or as the ratio of the amplitudes

$$\frac{p_{L2}}{p_{Lv1}} = \frac{v_{L2}}{v_{Lv1}} = \frac{1}{\sqrt{1 + \left(\dfrac{\omega\rho m}{2Z_L}\right)^2}} \,. \tag{5.21}$$

As expected, one can see that good sound insulation is possible if the wall has a large specific mass and thickness. This is always true. However, the sound-insulating effect of the wall also depends on the direction from which the sound strikes the wall and the flexural properties of the (housing) wall as well as the ability of the housing to resonate as a single body.

5.2 Sense of touch and hearing: hearing in humans

The sense of touch and hearing come into play at the end of the chain of structure-borne noise and airborne noise. They are the senses that humans use to evaluate noise. This is a subjective process, in other words it depends on the individual person and his or her physiological and psychological constraints. To determine the suitability of motor manufacturers and their prod-

ucts, this subjective evaluation must be transformed into objective, reproducible test conditions so that it can be measured by actual test instruments.

The sense of touch is very difficult to describe. Touch stimuli are experienced very differently depending on whether our fingers, faces, or feet are the receptors. With small motors, the sense of touch **tends to be of lesser importance**. It is studied in detail, though, when it comes to the haptic characteristics of a product. Therefore, we shall not consider the sense of touch further in this text. However, there are obvious analogies to the sense of hearing.

Humans perceive sound across a wide range of amplitudes. This range roughly extends across 10 to the power of 6 of the sound amplitude from its frequency-dependent (lower) **threshold of perception** (**threshold of hearing**) up to the **threshold of pain**, which is also frequency-dependent. A wide range of frequencies, which are dependent on the individual's age, are involved. In addition, there are also individual evaluations of frequency mixtures (**spectrum**) and changes in noises over time, as well as the **stereo effect** that results from hearing with both ears, along with simultaneous individual psychological effects. Therefore, the effect of airborne noise on human beings cannot be modeled completely using measurement technology. Because of the wide range of amplitudes that humans can hear, sound pressure $\tilde{p}$, sound velocity $\tilde{v}$, and sound volume I are also expressed as levels L. The level value is the logarithm of the **auditory threshold** (subscript 0) for the frequency 1000 Hz that was established for the **sound volume** I, in other words

$$L_I = 10\log\frac{|I|}{I_0}[\mathrm{dB}], \tag{5.22}$$

referred to as the sound volume level, sound intensity level, or simply **sound level** (stated in "bels" or more commonly in "decibels").

The sound volume level can also be described using the sound pressure with the aid of the **sound wave resistance** *Z*, which is assumed to be a real number, and the averaging time T_m:

$$L_P = 20\log\left(\frac{\tilde{p}}{p_0}\right)[\mathrm{dB}] \text{ with } \tilde{p} = \sqrt{\frac{1}{T_m}\int_0^{T_m} p^2(t)\,dt} \tag{5.23}$$

This is then called the **sound pressure level**. With a real Z it is therefore equal to the sound level. Statistical tests on many human beings have shown that the effective value of the sound pressure at the auditory threshold p_0 and at the frequency 1000 Hz is

$$p_0 = 20 \cdot 10^{-6}\ \text{Pa}. \tag{5.24}$$

The corresponding effective airborne sound velocity v_0 is

$$v_0 = p_0 / Z_{\text{air}} = 50 \cdot 10^{-9}\ \text{m/s}, \tag{5.25}$$

the oscillation deflection amplitude $\hat{x}_0$ is

$$\hat{x}_0 = \sqrt{2}\,\frac{v_0}{2 \cdot \pi \cdot 1000} = 11 \cdot 10^{-12}\ \text{m}\ . \tag{5.26}$$

The corresponding numerical values for the threshold of pain (subscript s), which are also determined as statistical mean values, are higher by a factor of 10 to the power of 6 at the same frequency:

- $p_s = 20\,\text{Pa}$
- $v_s = 50\,\text{mm/s}$
- $\hat{x}_s = 11\,\mu\text{m}$

Accordingly, the range of this level between the threshold of hearing and the threshold of pain is 120 dB at 1000 Hz, in other words 10 to the power of 12 of the sound volume.

However, our hearing not only perceives sound logarithmically – in other words as levels. It also evaluates the sound level differently depending on its frequency; it "measures" the **loudness** or **loudness level** L_S. The graph of the so-called audible area (Fig. 5.1) shows that the threshold of hearing increases as one moves toward low frequencies; in other words that the auditory sensitivity of the ear decreases in this zone. This also applies to varying degrees to frequencies above ca. 4000 Hz. Between 1000 and 4000 Hz the sensitivity is

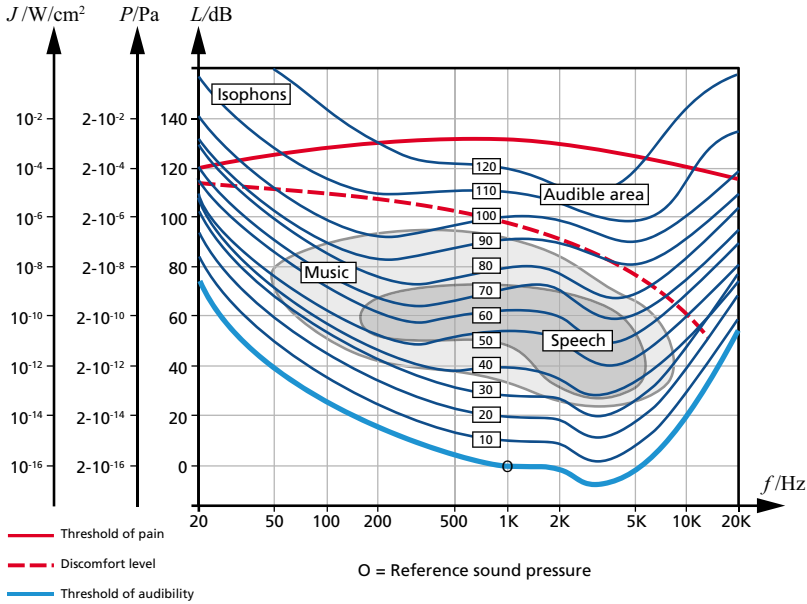


Fig. 5.1: Curves of equal perceived loudness

even somewhat greater than at 1000 Hz. However, the threshold of pain is less frequency-dependent. Sound frequencies below 15 Hz and above 20 kHz normally are not audible, i.e. in these ranges the auditory threshold increases sharply to the threshold of pain. Within the audible area we see statistically determined curves in which the sound volume is perceived to be equally loud, depending on the frequency. The listener hears the same loudness level. The loudness or loudness level L_S is stated in the unit **phons** and is scaled in such a way that at 1000 Hz, the phon values are equal to the sound volume levels in decibels. Hence, at 1000 Hz the phon scale is identical to the dB scale. At low and high frequencies, the dB values are greater than the phon values. The human ear can distinguish differences in loudness of approximately 1 phon, in other words sound volume differences of about 25%.

The term **perceived loudness** S is used to describe the factor, i.e. multiple, by which a sound is subjectively perceived to be louder or softer than a different sound. It is stated in **sones**, and the following is defined:

$$S = 2^{\frac{(L_s - 40)}{10}} \tag{5.27}$$

For the loudness level $L_s = 40$ phons, $S = 2^0 = 1$; at 50 phons $S = 2$, etc. Therefore, the perceived loudness doubles with every 10-phon increase in the loudness level. This is extremely important for efforts to control noise: If the perceived loudness of a motor is to be reduced by half, then the reduction in the loudness level must be 10 phons; in the case of a sound at 1000 Hz, the reduction in the sound volume level must therefore be 10 dB. This is equivalent to reducing the sound volume to 1/10 of the previous sound volume and therefore to a reduction of sound oscillation to about one third of what it previously was.

Perceived loudness and loudness level are merely rough measures of sound as it actually exists and as it is perceived by individual persons. This auditory perception of sound depends on many sonic effects that, although physically measurable, are very difficult to evaluate physiologically. Such effects must be taken into account in individual cases when evaluating sound sources.

Loudness and perceived loudness cannot be measured. But the auditory perception of loudness in phons or of the perceived loudness calculated from it in sones can be gaged from physically measurable structure-born sound and airborne sound and their frequencies by applying special metrological techniques to the data. This is accomplished with electrical filters that weight (attenuate or amplify) the components of the measured metrics according to their frequencies. Then the weighted components are added together and converted to logarithms. This produces the so-called weighted sound pressure level. This final result can also be illustrated with information on special characteristics (presence of noticeable partial oscillations/tones, frequency spectrum, change in the sound level over time, etc.).

It is often useful to determine level values across the entire measurement period. The variables obtained in this way are referred to as **time-range values**. In addition to the previously described levels found in acoustics, the following time-range values are also frequently encountered in acoustics and in vibration measurement technology:

- RMS value
- Crest factor
- Kurtosis

The **RMS value** (root mean square) corresponds to the quadratic mean value of the signal and therefore is only used in a mathematically correct manner for sinusoidal, periodic signals.

The **crest factor** describes the ratio of the maximum value to the effective value (RMS value). For signal analysis, this factor is used with measured signals that have brief, pulse-like peaks over time. The main advantage of this factor is that it is easy to calculate and interpret. A big disadvantage, however, is that with signals that have a high noise component or in measurements of various objects whose mean values differ greatly, the signal peaks can get "lost in the noise." Likewise, individual peaks in the measured signal that are due to malfunctions at the sensor can produce a high crest factor. The crest factor F_C is calculated from the ratio of the maximum amplitude V_{pK} to the quadratic mean value V_{RMS} as follows:

$$F_C = \frac{V_{pK}}{V_{RMS}} \tag{5.28}$$

With **kurtosis**, which is also referred to as the "standardized fourth central moment about the statistical mean," the repetition of an event is also evaluated. For example (in comparison to the crest factor), individual spurious signal noise can be "eliminated." A problem that cannot be solved by kurtosis, though, is that of elevated mean values in which periodic peaks cannot be detected. The statistical value is calculated from the mean α_1

$$\alpha_1 = \frac{1}{N} \cdot \sum_{n=0}^{N-1} x(n) \text{ (first statistical moment)} \tag{5.29}$$

using the number of measured values N and the measured values $x(n)$ and the standard deviation σ

$$\sigma = \sqrt{\mu_2} = \sqrt{\frac{1}{N-1} \cdot \sum_{n=0}^{N-1} \left(x(n) - \alpha_1\right)^2} \tag{5.30}$$

to obtain the kurtosis

$$\delta_4 = \frac{1}{N} \cdot \sum_{n=0}^{N-1} \left[\frac{x(n) - \alpha_1}{\sigma} \right]^4 \quad \text{(fourth statistical moment).} \tag{5.31}$$

5.3 Evaluation of auditory events: subjective variables used to take auditory sensation into account

We have already considered how vibrations are propagated as structure-borne noise and how they then reach the human ear via the resulting airborne noise. We then explained in detail what and how human beings hear. At this point the process of physical hearing is finished. However, in human beings there is still an individual evaluation of noises that have been heard (but not yet perceived!). This evaluation and the factors that affect it and the subsequent perception of the noise (= sensation) are discussed in the following section.

5.3.1 Weighting curves

Weighting curves have been developed to permit airborne noise recordings to be evaluated in a way that accurately reflects what we hear. These weighting curves attenuate or amplify certain frequency ranges in order to match physiological reality. All of the curves have one thing in common: 0 dB amplification or attenuation at 1 kHz. This means that the measurement and subsequent weighting of a 1-kHz tone produces the same results for all weighting curves. There are curves that are appropriate to the various requirements shown in Figure 5.2. This diagram shows that our hearing is the most sensitive in the range between 3 and 4 kHz. The A-weighted curve roughly corresponds to the downward-facing curves of equal loud-

ness. These curves of equal loudness are also the results of research by E. Zwicker [5].

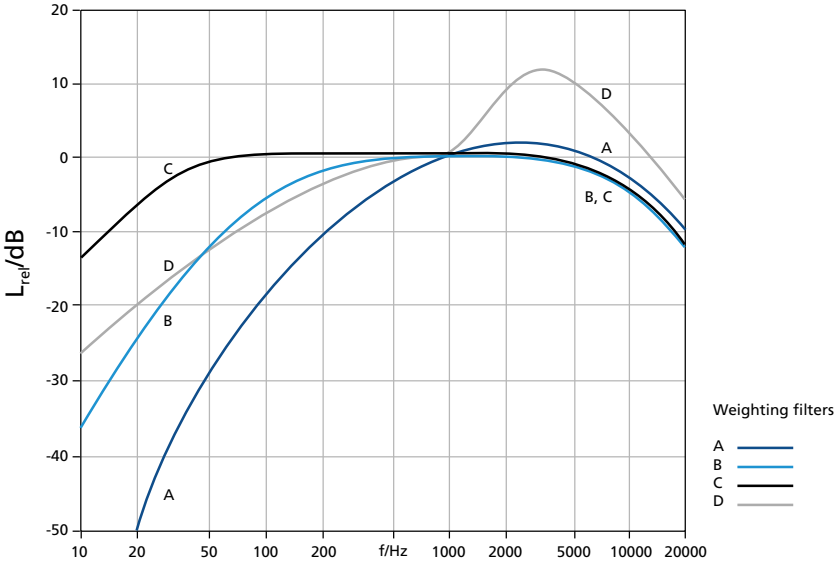


Fig. 5.2: Weighting curves [9]

Using the following example, we shall examine the effects of various weighting curves on an actual measured signal. This signal is from a measurement of airborne noise from a motor operating at 5000 rpm. The weighted signal is then subjected to a Terzanalysis (see chapter 7.3.3, p.112 f.) in order to illustrate the effects in the various frequency ranges.

The progress of a noise event over time is shown in Figure 5.3. If a one-third-octave analysis of this event is performed with various weighting filters, one obtains the curves relative to frequency that are shown in Figure 5.4. These curves differ greatly from each other. The greatest differences occur in the low-frequency range; the dB(A), dB(B) levels found in this range extend from 33.6 to 60.5!

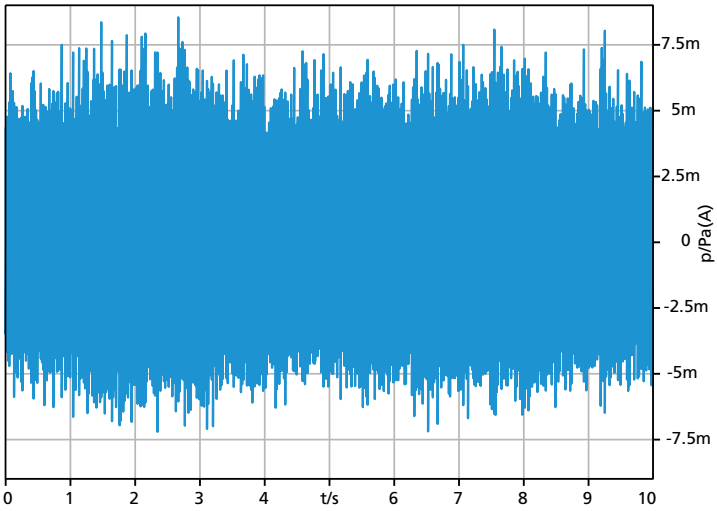


Fig. 5.3: The progress of a noise event over time

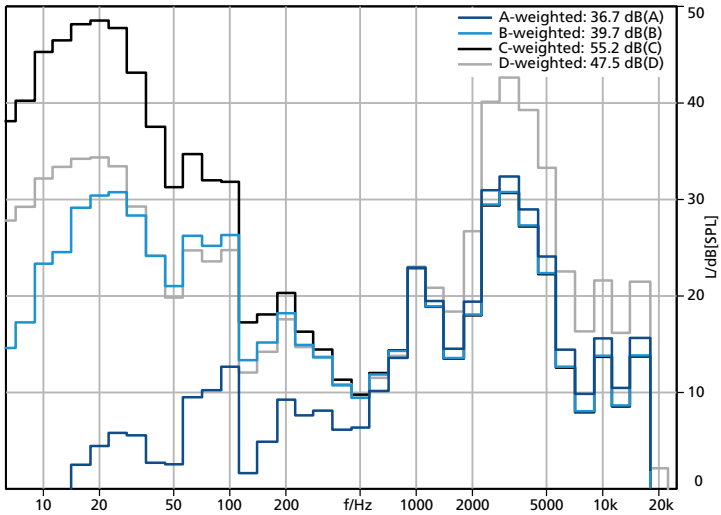


Fig. 5.4: Effect of weighting curves on the one-third-octave analysis of a signal (16 Hz to 20 kHz)

The various weighting curves are found in the following standards:

- A-weighting: Measurement of noise per DIN-IEC 651
- B-weighting: Measurement of noise per DIN-IEC 651
- C-weighting: Measurement of noise per DIN-IEC 651
- D-weighting: Measurement of noise per IEC 537; aircraft noise
- G-weighting: Infrasound weighting per ISO 7196

The most frequently used weighting is A-weighting. The letter for the given weighting system that is used is appended to the unit designating the weighting. This results in nomenclature like dB(A) or dB(C).

5.3.2 Psychoacoustic metrics

Unlike classic signal analysis, psychoacoustics is not based on physical parameters, but rather takes the characteristics of how human beings perceive noise (cognitive aspects) into account. These characteristics were obtained from tests performed with human subjects. The psychoacoustic characteristics were then modeled with the aid of various bandpass filters that best reproduced human auditory impressions.

The purpose of the psychoacoustic metrics was to generate metrics that could be calculated and that reflect human sensation. This also means that such characteristics cannot be measured directly but can only be calculated from other measurements. For the most part, the following parameters and units are used in this process: Perceived loudness [sone], sharpness [acum], perceived pitch [mel], roughness [asper], and fluctuation strength [vacil]. The definitions and derivations of the individual metrics are found in [5].

Using psychoacoustic metrics it is possible to express subjective auditory impressions with the aid of variables that can be calculated. It is even possible to define these impressions in a reproducible manner. The following example shows that psychoacoustic metrics are in fact able to describe noise events more precisely.

Figure 5.5 shows the frequency spectrum of two noise events, each with the same dB(A) value [44.9 dB(A)]. However, each noise produces a distinctly different impression: The noise event indicated by the "dark blue" spectrum

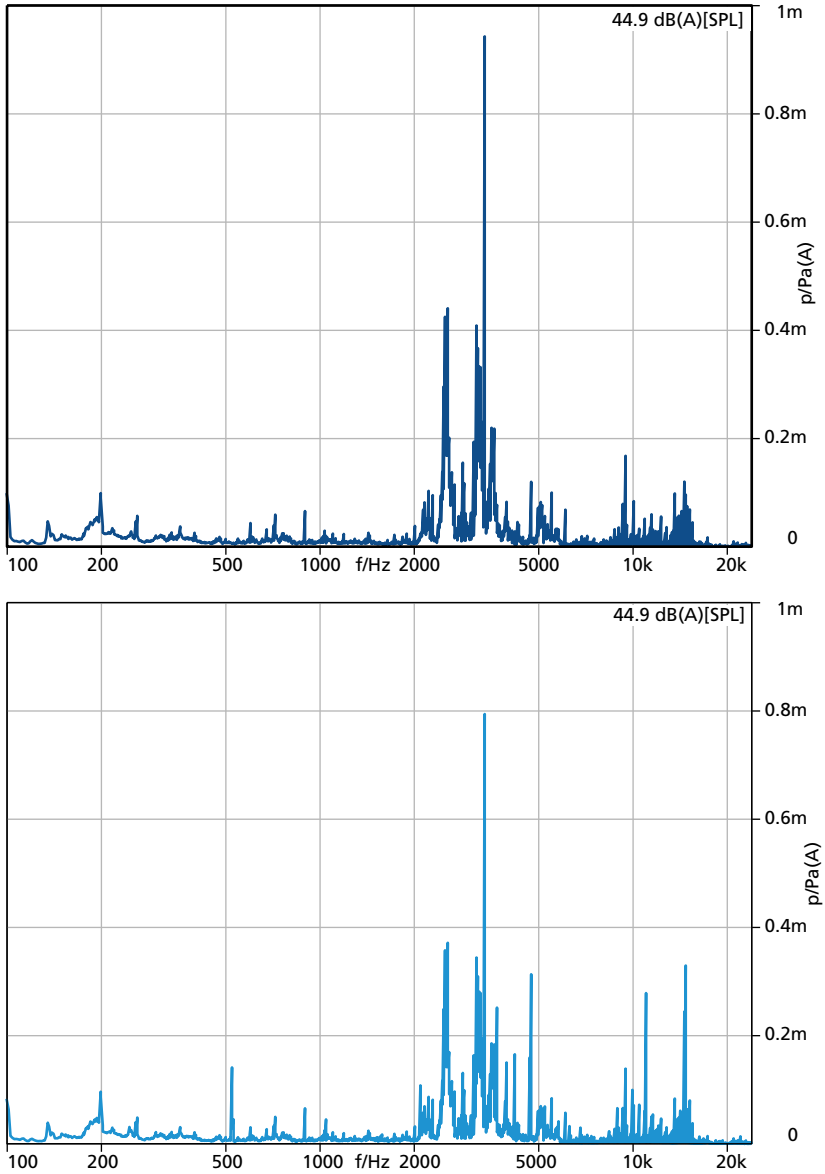


Fig. 5.5: Frequency spectra having the same dB(A) value (example)

is merely perceived as a kind of random noise, while the "light blue" spectrum produces a random noise that has clearly pronounced tonal components at 11 kHz and 14.7 kHz. These tonal components have an extremely irritating effect. In actual practice the component of such a noise that is actually irritating is not always as distinct as in this example. It is therefore useful to record the noise events with a dummy head and to listen to the signal repeatedly with various filters. Once the objectionable component has been detected, the cause needs to be found. If it can be attributed to a part that is resonating, the next step is to try to modify the part (geometry, material, location of mass, etc.) or to make a change at the location of the part (such as surface damping) in order to change the natural frequency or to systematically prevent sound radiation.

Here, it is clear that the airborne noise level rarely permits an objective evaluation. So we see that the psychoacoustic characteristics, for example, can be used to evaluate progress achieved through design changes. Table 5.3 shows the differences in the psychoacoustic metrics for this example, some of which are substantial.

Characteristic	**"Dark blue" signal**	**"Light blue" signal**
Airborne noise level in dB(A)	44.9	44.9
Perceived loudness in sones	3.09	3.27
Tonality in tus	0.143	0.268
Roughness in aspers	0.400	0.417
Sharpness in acums	3.32	3.72

Table 5.3: Airborne noise level vs. psychoacoustic metrics

One major advantage of using psychoacoustic metrics is that the relationship between the auditory impression and the calculated metric is linear. Unlike logarithmic levels, these metrics are easier for the user to understand. For example, a sensory impression that is twice as loud also corresponds to a perceived loudness value that is twice as large. For this reason, these values are becoming more and more popular for use when a direct comparison must be made between two noise patterns. On the other hand, the presen-

tation of a single absolute value is less popular because we lack the empirical knowledge that tells us which perceived loudness value corresponds to which sensory impression. For example, a user can roughly estimate what corresponds to an airborne noise level of 50 dB(A). However, the user often does not have enough experience to state a corresponding perceived loudness value in sones.

In the area of quality assurance, this type of perceived loudness value is becoming increasingly more important since its evaluation in standard production or supporting standard production can also provide information on the state of the manufacturing processes or components. As with level values used in acoustics, however, these characteristics cannot indicate the cause of the noise, but merely can be used to make decisions in standard production as to whether a part is good/bad.

5.4 Subjective perception of noise and weighting of noise

Weighting noises is a very complex and difficult undertaking. There are countless factors that affect the individual subjective assessment of these weightings. The purpose of this chapter is to explain this area and the problems that can result.

The expectation that the user has for the product that is to be evaluated probably has the greatest effect on how values are sensed. Furthermore, there are many factors that lead to this expectation When we hear a noise event, our brain evaluates it. Figure 5.6 shows the independent variables that our brain considers to be the most important and that it uses to evaluate the event. There are many factors that also affect how we evaluate objectionableness, pleasantness, and other criteria.

5.4.1 Influence on the perception of noise by individual factors

As Figure 5.6 shows, there are some physiological factors that can affect the evaluation of noises by human beings "from within." One such factor is age. As humans become older, the highest frequency that they can perceive grad-

ually decreases. This means that the evaluation of noises in the range from 10 to 16 kHz is highly age-dependent.

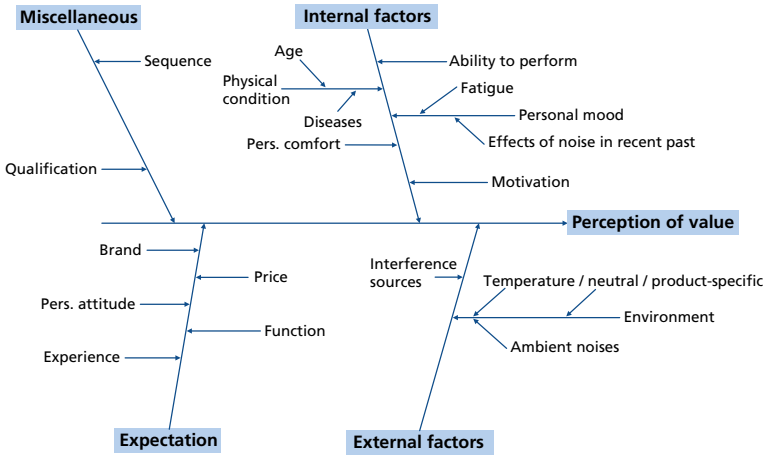


Fig. 5.6: Factors affecting subjective evaluation

Likewise, the psychological state of the test subject cannot be ignored, since a person who is stressed out or has not got enough sleep perceives certain noise characteristics differently than a person who is completely relaxed and rested. In the worst case, for example, spending the previous evening in a disco with the poorer hearing that results will affect how noise amplitudes are evaluated. Diseases that affect hearing sensitivity or perception can have a similar effect. The test subject's ability to perform well also determines how long the subject can participate in a subjective evaluation without changing his or her subjective impression. This can be tested during the evaluation, for example by having the subject evaluate identical noise patterns a number of times. Likewise, the test subject's motivation has a decisive impact on the results of the evaluation. An unmotivated person tends to provide random and arbitrary evaluations that have nothing to do with the noise pattern. Such an evaluation is of no value.

5.4.2 External factors and the ambient level

External factors also affect the subject's subjective impression of the noise pattern. For example, the environment can have a direct as well as an indirect effect on the evaluation.

The environment has a direct effect if it is neutral or product-specific. It is important to consider that the ambient noise can have a decisive impact on whether the subject perceives the noise that is to be evaluated as being "loud" or "soft." Studies have shown that a noise is not perceived to be "loud" or "soft" unless the difference between the sound pressure level that is to be evaluated and the ambient sound pressure level is > 6 dB.

Furthermore, the environment directly affects how the subject feels and therefore indirectly affects the evaluation result. The factors environmental noise or temperature can have the same indirect effect. If a person feels ill at ease in the test environment, this directly affects the evaluation result.

5.4.3 Expectations

The expectations that a person has of a product play an important role in how he will evaluate the noise produced by the product. For example, if the subject recognizes the brand and the associated perceived quality or price of the product, a certain expectation is established along with a tendency to want to meet this expectation. Even more specific are the expectations that can result if a subject is already familiar with an earlier version of the product and can make a direct comparison. Likewise, the subject's personal attitude with regard to a product or brand can have a strong impact on the evaluation. For example, if the subject considers the product to be unnecessary and superfluous, the evaluation will certainly be worse than that of a product that is considered to be necessary and essential.

There are also expectations as to the noise pattern that relate to the function of the product. For example, the noise expectations for a translational movement are different from those for a rotational movement; likewise, the noise expectations for an electric motor differ from those for a gasoline-powered engine.

Another factor is the specific feedback relating to the function of the motor. In other words, the user will hear that the device is working properly (example: a vacuum cleaner). With this feedback, the user would have an adverse impression if the noise were too quiet or did not fit the function.

5.4.4 Miscellaneous

In addition to what we have explored thus far, there are some other factors that can also affect the subjective evaluation of noises. To subjectively evaluate noises, the test subject must be adequately qualified. In other words, the subject must be able to arrive at a meaningful conclusion as to the noise produced by the product.

The sequence in which the test samples are presented also has a major effect on the evaluation of noises. For example, if one begins with a supposedly very "poor" example of a noise, the following examples will always tend to receive better evaluations than they would have received without the effect of the comparison noise. The reverse effect – if one starts with a supposedly very good example – also occurs. In order to avoid these effects and to obtain an evaluation that is as neutral as possible, individual examples should be played repeatedly in different order. Another possibility is to play some sample files that cover the spectrum of the evaluation for the test subject without asking the subject to evaluate them.

5.4.5 Procedure of typical auditory tests

The subjective evaluation by the test subjects through the use of an auditory test is a very useful way to attain an impression of various noises. Depending on the purpose of the test, there are various types of auditory tests that can be used.

In the case of a **ranking**, for example, various examples of noises can be placed in a given order by the test subject. It may not be immediately apparent why the subject selected the given ranking, but one can quickly see which noise pattern is the favorite. The disadvantage of this method is that it gives a ranking for the noise patterns, but does not evaluate the distance between the indi-

vidual examples. Thus, it cannot be determined whether the second place in the ranking is just barely acceptable or perhaps already completely unacceptable.

Another way to obtain a result very quickly is through **pair comparison**. In this test, two noises are always presented one after another, and the subject decides, for example, which noise pattern is "louder" than the other. Problems can result with this method if noise A is perceived to be just as loud as noise B. Therefore, the option A = B must be available in addition to the two options A > B and A < B.

In order to obtain a more precise response as to noise quality, a **category-based evaluation** can be performed. In this method it is first necessary to determine the criterion that must be evaluated. Then the test person must assign the various noises to a scale. The 5-level Rohrmann scale or a 10-level scale are often used to evaluate objectionable noises in automotive vehicles in accordance with VDI-Richtlinie 2563 published by the Association of German Engineers (Table 5.4).

Rohrmann	**VDI-Richtlinie 2563**
Not at all	Cannot be detected even by experienced evaluators
	Can only be detected by experienced evaluators
Little	Can only be detected by critical persons
	Can be detected by everyone
Moderate	Experienced as objectionable by some persons
	Experienced as objectionable by all persons
Predominant	Experienced as a defect by all persons
	Experienced as a severe defect by all persons
Complete	Only partially acceptable
	No longer acceptable

Table 5.4: Commonly used scales for evaluating noises

In order to obtain an optimal evaluation of subjective noise quality, the use of a **semantic differential** is recommended. In this approach the test subject must classify a noise according to a number of criteria. The definition of the evalua-

tion criteria is an important part of preparing such a semantic differential. This differential should be appropriate to the product and should be set up in such a way that the evaluation is as simple as possible for the specific case. With some products, for example, it may even make sense to use unconventional evaluation criteria (such as a male versus a female noise). Figure 5.7 shows an example of such an evaluation form. In addition to the option shown here, the opposite of the descriptive adjective can be listed in order to simplify the classification and to make the scale clearer. A final method of evaluating noises, which is also very elaborate, is **AISP (Exploration of Associated Imagination on Sound Perception)**. In this method the test subject is not given any requirements or criteria that are to be used to evaluate the noise. Instead, the subject states his or her impressions of the noise, free of any external influences and questions. Here, first impressions, associations, as well as emotions are important. These statements are recorded by the tester and then evaluated.

The following must be considered for all of the types of auditory tests referred to above: The test subjects must be properly instructed before the test is performed. The subjects will not feel comfortable and not be able to provide meaningful results unless they have understood these instructions. It is important to communicate to the test subjects that there are no wrong responses but rather that sensory impressions can differ among test subjects. In addition, the approximate length of the test should be communicated so that the test subject knows what to expect. It may be necessary to ask the subject about how he or she is generally feeling so that the results that are obtained can be weighted properly.

A room in which the subject can feel at ease should be selected for the test environment. This might be a quiet room or an environment that is specific to the application (such as an automobile interior). The test signals must all have a uniform quality. Here, for example, it is useful to use dummy head recordings (chapter 6.1.2, see p. 88 f.) with the reproduction equipment needed to reproduce the data properly for human hearing. Additional factors that must be considered in recording the signals are the duration of the signal, the measurement environment, as well as the reproduction by means of loudspeakers or headsets.

After the test data have been assessed, they must be subjected to a detailed analysis. The individual criteria used in this analysis go beyond the scope of present work.

Questionnaire

Noise no.: _____ Subject no.: _____

Evaluate what you felt about the sound by circling one of the numbers from 1 to 10. 0 means "not at all" and 10 means "to the highest degree possible."

unpleasant	*0*	*1*	*2*	*3*	*4*	*5*	*6*	*7*	*8*	*9*	*10*
irritating	*0*	*1*	*2*	*3*	*4*	*5*	*6*	*7*	*8*	*9*	*10*
loud	*0*	*1*	*2*	*3*	*4*	*5*	*6*	*7*	*8*	*9*	*10*
pleasant	*0*	*1*	*2*	*3*	*4*	*5*	*6*	*7*	*8*	*9*	*10*
rough	*0*	*1*	*2*	*3*	*4*	*5*	*6*	*7*	*8*	*9*	*10*
intrusive	*0*	*1*	*2*	*3*	*4*	*5*	*6*	*7*	*8*	*9*	*10*
acceptable	*0*	*1*	*2*	*3*	*4*	*5*	*6*	*7*	*8*	*9*	*10*
continuous	*0*	*1*	*2*	*3*	*4*	*5*	*6*	*7*	*8*	*9*	*10*
reverberating	*0*	*1*	*2*	*3*	*4*	*5*	*6*	*7*	*8*	*9*	*10*
large	*0*	*1*	*2*	*3*	*4*	*5*	*6*	*7*	*8*	*9*	*10*
pulsating	*0*	*1*	*2*	*3*	*4*	*5*	*6*	*7*	*8*	*9*	*10*
mechanical	*0*	*1*	*2*	*3*	*4*	*5*	*6*	*7*	*8*	*9*	*10*
expensive	*0*	*1*	*2*	*3*	*4*	*5*	*6*	*7*	*8*	*9*	*10*
surprising	*0*	*1*	*2*	*3*	*4*	*5*	*6*	*7*	*8*	*9*	*10*
droning	*0*	*1*	*2*	*3*	*4*	*5*	*6*	*7*	*8*	*9*	*10*
suggesting a musical tone	*0*	*1*	*2*	*3*	*4*	*5*	*6*	*7*	*8*	*9*	*10*
sharp	*0*	*1*	*2*	*3*	*4*	*5*	*6*	*7*	*8*	*9*	*10*
powerful	*0*	*1*	*2*	*3*	*4*	*5*	*6*	*7*	*8*	*9*	*10*

Overall, I like this sound:

A lot											*not at all*
	0	*1*	*2*	*3*	*4*	*5*	*6*	*7*	*8*	*9*	*10*

Fig. 5.7: Questionnaire for the subjective evaluation of noises

6 Measuring noises and vibrations

Nowadays, subjectively perceived noises can be recorded, analyzed, and reproduced in an auditorily appropriate manner using suitable measurement equipment. We have at our disposal a wide range of measuring equipment that can be used to perform all manner of measurements on airborne and structure-borne noise. Given the extensive number of motor designs that are possible, it is not always easy to select the correct instruments. The next section introduces the most important measuring instruments and provides guidelines on how to use them effectively.

6.1 Equipment for measuring airborne noise

Airborne noise is measured using microphones that either react to air pressure changes or sound oscillation velocity (sound velocity). The development of the technology used to measure airborne noise coincided with the development of the telephone. The first devices capable of converting sound to electrical energy – electroacoustic transducers – come from this period. Since then, various types of transducers have been developed and optimized for specific applications. The main applications for electroacoustic transducers continue to be found in the telecommunications industry. However, high-quality electroacoustic transducers are also used in the music industry and in measurement technology. The following sections provide an overview of the various transducer types and the areas in which they are used.

6.1.1 Microphones

Microphones (Fig. 6.1) are devices whose design allows them to convert a change in air pressure to a change in electrical voltage. Hence, they are also

referred to as electroacoustic transducers. The first microphones were developed in the middle of the 19th century. Figure 6.2 shows the various forms of these transducers. Additional detailed explanations of the various types of transducers may be found in [4] and [8].

Today, airborne noise measurement technology primarily uses various types of condenser microphones. These microphones are generally classified according to the diameters of their diaphragms. The most common sizes are: $^1/_8$", ¼", ½", 1". However, the limit of fre-

Fig. 6.1: Microphone

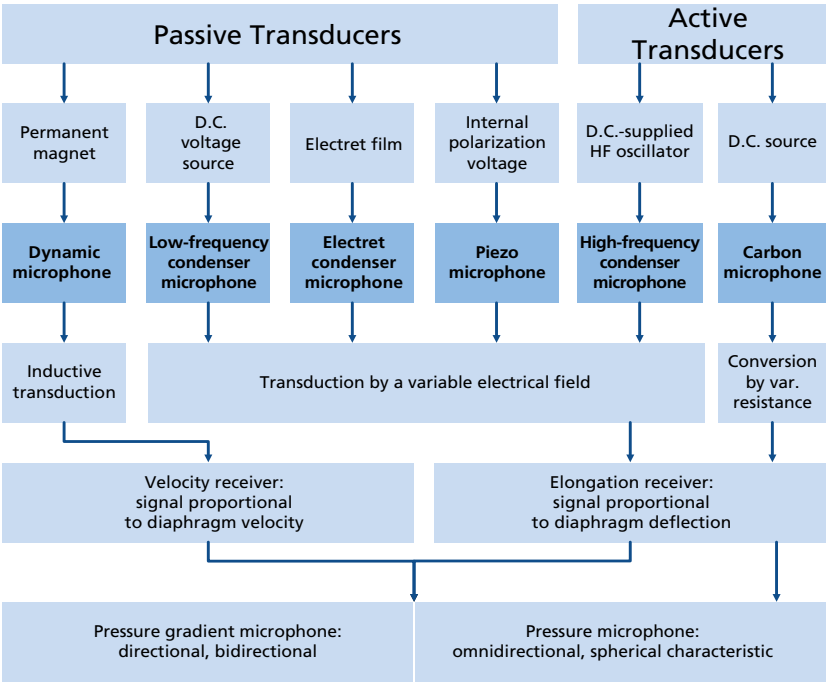


Fig. 6.2: Transducer principles (source: Wikipedia)

quency response changes according to the area of the diaphragm. The larger the area, the lower the frequency response limit. These microphones work by converting air pressure changes caused by the sound that is to be measured into voltage changes between two condenser plates.

All microphones have **directional characteristics**. The directional characteristics, i.e. directivity, typical of a microphone will vary depending on how the microphone is constructed. Typical directivity shapes are omnidirectional, bidirectional, and shotgun. Hybrid shapes such as cardioid, subcardioid, supercardioid, and hypercardioid are also encountered. Figure 6.3 shows the sensitivities of the microphones in the form of polar patterns. It is important to understand that the directional characteristics of microphones will be lost in a diffuse field (a field in which the energy density is constant at all locations).

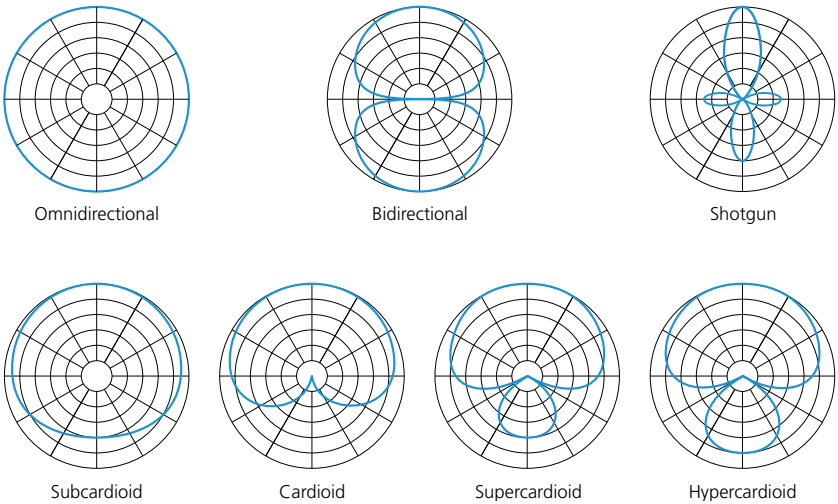


Fig. 6.3: Typical directivity patterns of microphones

A microphone always picks up the entire noise pattern. This can be an advantage or a disadvantage. Compared with an accelerometer, a microphone is not good at measuring radiating surfaces selectively. On the other hand, it has

the advantage of being a single sensor that can be used to acquire the entire spectrum of radiated sound.

Microphones only sense those sound components that can be radiated by the part. Therefore, only information that can be perceived subjectively is evaluated or analyzed. Other vibrations – such as structure-borne noise within the part – cannot be studied with microphones. One big disadvantage of using measurement microphones is that the measurements must be performed in a test booth in order to reduce or, ideally, eliminate interference from the environment. Aside from the wide range of microphone technologies that are available, in recent years special microphone systems that have markedly expanded the spectral range and possibilities of airborne noise measurement have appeared (see the following sections).

6.1.2 Dummy head

Dummy head (binaural) measurement technology is a special form of airborne noise measurement. In this technology, two microphones are integrated into a model of a human bust (shoulder and head area) in place of ears (Fig. 6.4). With this arrangement, auditory events can be recorded and reproduced in a manner that is auditorily correct for human beings. This technology is based on the fact that our sense of hearing is greatly affected by filtering and reflexions in the shoulder and head area. Since humans have learned to localize sounds by means of these reflections and above all through filtering phenomena, such recordings can among other things make it possible to determine the direction of the noise.

Fig. 6.4: Dummy head

Fig. 6.5: Dummy head used to perform measurements

Because it permits excellent auditorily correct recording, the dummy head is ideal for recording reference signals such as borderline samples. At the same time, the dummy head makes it possible to replicate realistic conditions of actual use, as is shown in Figure 6.5. Here, an operations helmet with a built-in fan motor is being tested in an actual-use position in order to gage the effect on the user.

6.1.3 Microphone arrays

In array measurement technology, a large number of microphones are used in a large area to measure the totality of sound emissions (Fig. 6.6). The location of the sound event is determined with the aid of the run-time differences between the individual microphones for the event. Careful data analysis and superimposition of a video image reveals which amplitudes and frequencies are emitted at which locations on an object being tested. The big challenge that is presented by this technology is not the evaluation

Fig. 6.6: Microphone array

of the data but the need for sensible ways to store and process the data. This measurement method often uses arrays of 120 microphones and a scanning rate of 192 kHz.

It is important to recognize that this technology has inherent limits and that microphone arrays are not a universal solution for noise analysis. They only produce the desired result when they are used correctly. On the other hand, valuable information that can help find the causes of noise can usually be obtained with "simple measurements."

6.1.4 Methods for measuring airborne noise with microphones

Using the correct procedure to measure airborne noise signals is of critical importance for the subsequent evaluation and interpretation of the signals. Reproducible and interpretable results can only be obtained if the data are acquired correctly. Because of the high degree of sensitivity that microphones have to background and reflected noise, the proper measurement booth must always be used when making measurements with microphones. One exception is when measurements are made in defined environments or in a specific situation (e.g. when an automobile or train passes by).

The directional characteristics of microphones can be used in the near field – i.e. in the vicinity of the radiating surface – to eliminate some interfering factors. However, a microphone loses its directional characteristics in a diffuse field. Therefore, selective measurement can only be relied upon under certain conditions. The measured amplitudes and levels follow the **laws of distances** $1/r^2$ or $1/r$. With all energy variables (physical variables that describe the energy of a physical field), for example the sound power, the amplitude follows the $1/r^2$ law, while the field variables (physical variables that describe the condition of the field), for example sound pressure, sound velocity, and sound displacement follow the $1/r$ law. For the level calculation, the level decreases by 6 dB according to both laws when the distance doubles.

The measuring distance is important for measuring airborne noise. The sound field around the object being measured can be divided into the near field and the far field. Measurements must be made in the far field in order to record level values correctly. In the near field, constructive or destructive interference occurs depending on the frequency and the measurement location. As a result of frequency dependency, there is no precise rule for when the near field becomes the far field. This transition is directly associated with the wavelength and the size of the radiating surface. As a general rule of thumb, the minimum measurement distance for airborne noise having a maximum wavelength of λ_{max} is:

$$\text{Measurement distance} \geq 2 \cdot \lambda_{max} \tag{6.1}$$

6.2 Equipment for measuring structure-borne noise

A broad range of sensors is available for measuring structure-borne noise. In addition to the basic dynamic variables (distance x – velocity v – acceleration a), it is possible with a single component to obtain information simultaneously on these variables in three orthogonal directions ("triaxial sensor"). The basic dynamic variables relate to each other with regard to frequency as follows:

$$a = \omega \cdot v = \omega^2 \cdot x \tag{6.2}$$

As a general rule, the measurement sensors must be sufficiently small and light or contactless so that they can perform measurements at individual points and not affect the vibration. One technology that is now mature is contactless vibration measurement using a laser vibrometer. With these systems it is also possible to measure the three dynamic basic variables uniaxially and triaxially.

6.2.1 Accelerometers

Fig. 6.7: Accelerometer

An accelerometer (Fig. 6.7) supplies an electrical signal that is proportional to acceleration. It contains a piezoelectric quartz crystal that acts as a spring between the measuring location and a (free) mass; in other words it is an oscillatory system. Such a system has limits with regard to its sensitivity and frequency measurement range. So there are different sensors for different measurement tasks. For low frequencies, relatively heavy sensors are used. They are limited with respect to measurement sensitivity on the high end. For high frequencies, lightweight sensors that only have a slightly disruptive effect on the vibrations but whose measurement sensitivity is limited on the low end are used.

Accelerometers are used to measure and monitor vibrations and to perform shock tests. The advantage is the low price and wide range of sensors that are available for various requirements (design size, mounting system, frequency range, etc.). The biggest disadvantage of these sensors is that the applied mass directly affects the measured object. In particular in the area of small motors, standard sensors can easily exceed the mass of the object under test and therefore corrupt the results dramatically. The following equation shows the approximate effect of the mass of the sensor on the level:

$$\Delta L = 10\log\left(1+\frac{\left(2\pi f\cdot M)^2\right)}{\left(2{,}3\cdot c_L\cdot\rho\cdot h^2\right)^2}\right) \tag{6.3}$$

When accelerometers are used, attention must be given to the fact that contact resonance (which can result in incorrect measurements!) can occur depending on the mounting type. Typical mounting types are:

- Adhesive bonding
- Magnetic
- Probe
- Mounting pins
- Wax

Figure 6.8 shows the measurement areas that correspond to the typical mounting types and that are limited by the resonant frequencies. In the laboratory, attachment types can usually be selected freely as needed, but when testing is done in standard production, the adaptation options are severely limited or totally constrained by the given situation. Therefore it often is not possible to circumvent the resonance frequencies. Instead, one must be aware of them and take them into account when data are evaluated.

The electrical effects that occur due to friction are another interference factor that must not be ignored. The movement (friction) of the cable produces a static charge, which can cause measurement errors. For this reason, the cable to the amplifier must be securely attached.

To avoid electromagnetic interference, single- or better still, double-shielded cable must be used, depending on the measurement environment.

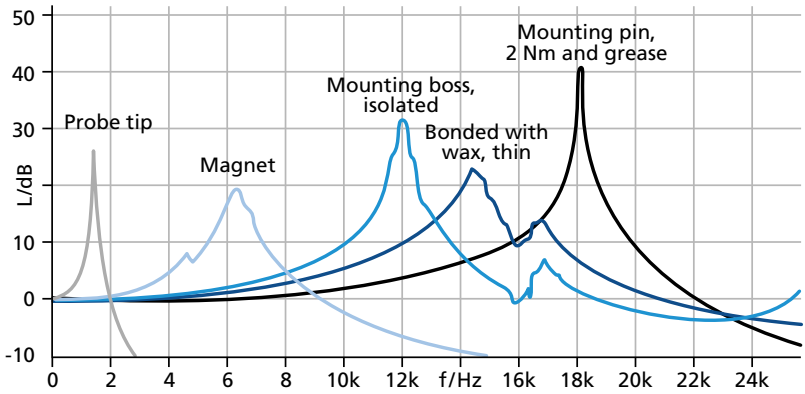


Fig. 6.8: Resonance frequency depending on adaptation

6.2.2 Laser vibrometer

While the size of the device under test plays a lesser role in airborne noise measurements, when structure-borne noise is being measured with conventional accelerometers with small noise or vibration generators, the limits of what is possible in structure-borne noise measurement are quickly reached (feedback effect on vibration behavior due to the change in mass). In such cases, it makes sense to turn to the contactless measurement methods offered by laser Doppler vibrometry. The two main advantages of this approach are that the measurement is free of contact, so that the associated measurement does not feed back to the device under test, and that the measuring location is of small size, in the case of a laser vibrometer (Fig. 6.9) only a few micrometers. Thus, vibrations on brushes can be measured without difficulty, for example, (width about 100 µm), something which would not be possible with contacting technology. The disadvantage of such measuring methods is their price, which can easily be 20 times the purchase price of an accelerometer.

Fig. 6.9: CLV 2534 laser vibrometer

When measurements are performed with the aid of laser vibrometers, there is no need for modifications to be made on the sensor and device under test. However, certain things must be considered in order to obtain

a noise-free signal. The design and operation of the Doppler interferometer results in ideal working distances (so-called visibility maxima), which are determined by physical requirements. These distances depend on the laser light source and the optical system that are used. They are specified by the manufacturer.

The reflection from the object under test must be sufficiently large to allow an adequate measurement signal to be obtained. In the case of dull black surfaces, this can cause problems. Such problems can be overcome by applying reflective foils or sprays. It is also important to aim the beam as perpendicularly as possible to the surface being measured in order to also obtain maximum reflection and minimize the amplitude error ("cosine error") described in chapter 6.2.5 (see p. 96). In addition, movements perpendicular to the beam direction must be prevented since they can cause signal interference or increased random noise.

6.2.3 Displacement sensors

Displacement sensors tend to play a lesser role in measurements made on small motors. These devices are classified as being inductive, capacitive, optical, or eddy-current-based depending on their operating principle. Since the amplitude of the vibrations is governed by quadratic functions, amplitude resolution must be substantially higher when displacement is being measured. For example, an oscillation of $a = 1\ \mathrm{g}$ at 1000 Hz corresponds to a displacement amplitude of only 0.25 µm. These requirements apply of course equally to measured data acquisition systems.

6.2.4 Force sensors

In addition to the classic variables of distance, velocity, and acceleration, it is also possible to analyze the vibrations in a motor using the forces that occur. Since the sensor cannot be integrated directly at the point where the forces are produced, the motor must be adapted as much as possible to the sensors. But this usually affects the vibration characteristics significantly, which is why this method of measurement is not used with small motors.

6.2.5 Methods for measuring structure-borne noise

When structure-borne noise is being measured, incorrect measured values can also be obtained if some important considerations are ignored. For this reason, we shall examine the most important rules in the next sections.

Accelerometers and laser vibrometers differ with respect to their measuring principles. While the accelerometer measures in absolute terms, the laser vibrometer measures relative to the sensor head. This means that two different requirements must always be met in the design of the measurement setup.

If an accelerometer is used, the measurement setup must be be completely isolated in order to totally eliminate any effects on the setup that might come from the outside. In order to completely prevent the interfering effects that can occur when a laser vibrometer is used, the "only" thing that needs to be done is to ensure that there are no relative movements between the sensor and the object being measured. Vibration components that cause a phase shift to occur between the sensor and the object being measured also result in an inaccurate measurement.

With all measurements, the measurement direction must be perpendicular to the direction in which the structure-borne noise is propagated in the body. If a perpendicular orientation is not possible, a cosine error must be taken into account when the amplitudes are subsequently evaluated. The cosine error describes the error in amplitude that occurs in measurements in which the measurement direction is not the same as the propagation direction. The measurement angle (deviation from the perpendicular) must be taken into account in order to determine the actual amplitude. The requirements shown in Figure 6.10 apply.

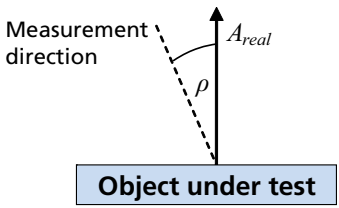


Fig. 6.10: "Cosine error"

When a laser vibrometer having three measurement heads (also called a scanning vibrometer) is used, the oscillation components of the three orthogonal directions in space are calculated from the three measurement signals. The same applies when accelerometers are used. The use of triaxial sensors (see p. 91) is recommended for recording the three orthogonal components, provided that these sensors can be used without any feedback effects (the mass of the sensor must be "much smaller" than the mass of the component being tested!). It is also important to make certain that the vibration that is to be measured is measured as close as possible to its point of origin in the vibration direction that is to be tested (Fig. 6.11) in order to avoid damping and the nonlinear effects of connecting pieces and material transitions as much as possible.

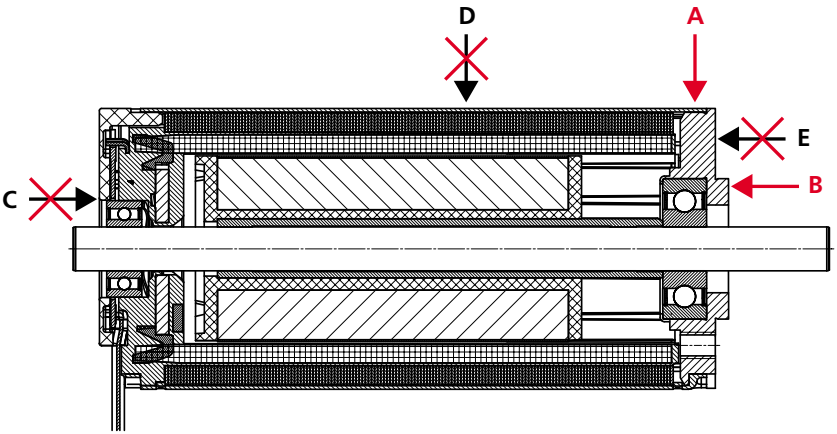


Fig. 6.11: Measuring point⟶measuring direction

Ideally, the measuring point should not be located at a vibration node, but rather as much as possible at a vibration antinode. Since this is difficult to achieve in various frequency components, care must be taken to ensure that all of the components are being measured, or are measured at a number of locations.

Table 6.1 shows the possibilities and limits of various sensor types. The parameter ranges listed in the table cannot be measured using a single sensor. The parameter ranges that are stated here therefore show the entire bandwidth that is covered by various designs of the given measurement device.

Measuring equipment / Characteristic	Microphone	Accelerometer	Laser vibrometer
Frequency range	0–140 kHz	0–54 kHz	0 to 600 MHz
Amplitude range (max.)	180 dB	300,000 g	100 m/s
Resolution (max.)	1 mV/PA	10 V/g	0.5 V/mm/s
Weight (min.)	–	0.2 g	–
Measuring point size (min.)	–	Dia. 5 mm	Dia. 1.5 µm
Price (min.)	€600	€1000	€15,000

Table 6.1: Table of measuring equipment

6.3 Reproducible measurement

Being able to measure reproducibly is an essential characteristic of noise and vibration test facilities and equipment. Before meaningful analysis is possible, noise and vibration values must be able to be measured reproducibly. The measurement must be reproducible and also be able to be calibrated. Calibratability ensures that when a test setup is duplicated, all of the test rigs produce identical results (taking accuracy into account). If a noise cannot be reproduced and also calibrated, it cannot be analyzed, and it therefore makes no sense to attempt to interpret the data. Reproducibility ensures that the measurement results are always identical, independent of the technician and other possible environmental factors (see chapter 8, p. 134 ff.).

If the devices that are being tested are freely suspended in an anechoic chamber, it would seem to be easy to implement a reproducible measurement. But when tests are integrated into the production environment or in particular in the case of in-line tests, measurement is particularly challenging. Two completely different basic requirements arise:

- Absolute-value measurement without any effects from the environment or the mounting system (suspension)
- Comparison measurement with a potentially reproducible effect of the mounting system on the device under test

6.3.1 Absolute-value measurement

At first, it would seem to be trivial to implement absolute-value measurement. And this is in fact true, provided that one is willing to test a motor in an unloaded condition. However, as soon as a measurement must be performed under operating conditions (under axial, radial, torque load), such a measurement is difficult to perform and usually ends up being a comparison measurement in which the effect of the mounting system is reproducible. However, this effect should be as low as possible, as should the sound radiation from any additional components that are necessary, such as brakes, couplings, or bearings.

Fig. 6.12: Setup for measuring airborne noise

Figure 6.12 shows an example of a setup in which the device under test is operated under load and a sound pressure measurement is performed. It is necessary to measure under load since the load torque has a substantial effect on the sound pressure level. When the setup was developed, care was taken to ensure that no interfering covers or setup components would have a damping effect on the airborne noise in the direction of radiation. Care was also taken to ensure that the vibrations occurring in the device under test did not excite natural frequencies in the test setup. Therefore, the base plate of the mounting system was of a very massive design relative to the device under test.

6.3.2 Comparison measurement

Figure 6.13 shows the frequency spectrum of a freely suspended motor (dark blue) and that of a motor that is clamped into a test fixture (light blue). One can clearly see the effect of the fixture on the measurement result. The fixture would not be suitable for obtaining an absolute value.

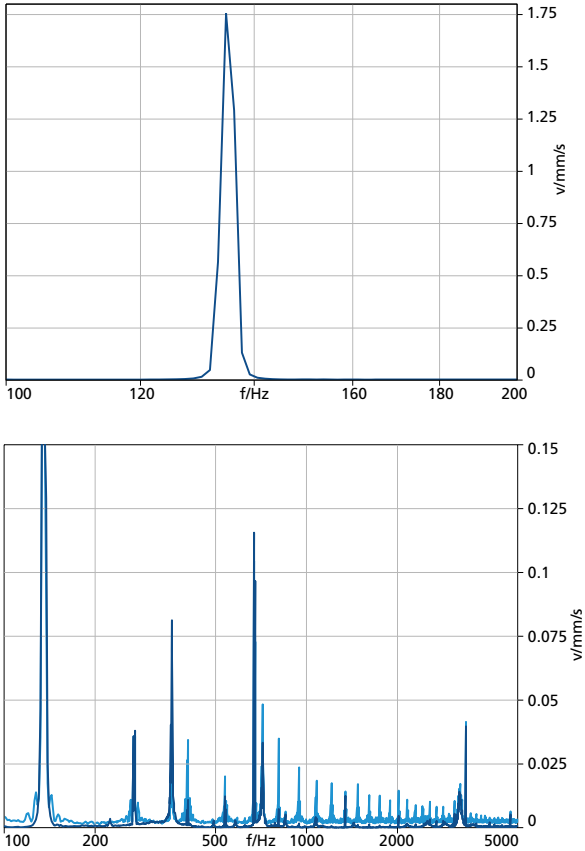


Fig. 6.13: Frequency spectra of freely suspended (dark blue) and clamped (light blue) motor

The two segments shown in Figure 6.14 each show the same five measurements in different frequency ranges. The measurements show that the error caused by the fixture remains reproducible, despite the same motor being installed repeatedly. Therefore, this solution is acceptable for actual practice. The definition of reproducibility is discussed in greater detail in chapter 8.3 (p. 134 ff.).

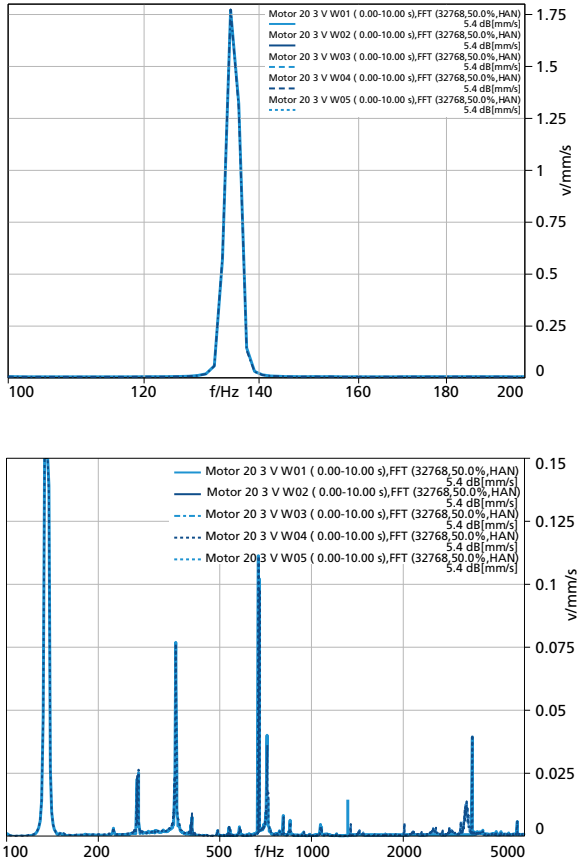


Fig. 6.14: Frequency spectra of a clamped motor (repeat measurement)

6.4 Measurement rooms

6.4.1 Anechoic chamber

Anechoic chambers (Fig. 6.15) are also frequently referred to as "acoustically dead" or "echo-free" chambers. Of course, such terms would demand that the degree of absorption at the walls be 100%. This is impossible to achieve technically, so we use the term anechoic chamber in a relative sense to mean a chamber that is extremely low in sound reflection.

Fig. 6.15: Anechoic chamber (Technical University of Dresden)

Anechoic chambers are completely or partially lined with sound-absorbing material (walls and ceiling: semi-anechoic chamber). The frequency-dependent degree of sound absorption can be adjusted by means of the design of the pyramid-shaped absorbent material. In this way, absorption of 99.9% can be achieved.

In addition to the ability to replicate free-field conditions in the chamber, the chambers must also have good damping of interference noises. This is accomplished by isolating the entire chamber using springs between the foundation and the measurement chamber. This type of design is often very elaborate and expensive, however it does keep interfering noises from the outside environment from entering the chamber. Such chambers meet the requirements of DIN 45635 for the measurement of defined sound variables.

6.4.2 Reverberation chamber

Reverberation chambers (Fig. 6.16), in contrast to anechoic chambers, achieve the requirements needed for a diffuse field. These requirements are achieved

Fig. 6.16: Reverberation chamber (Technical University of Dresden)

by designing the walls to have a reflectance of >98% and by distributing various diffusers freely in the chamber. In this way, a homogeneous distribution of energy results in such chambers.

Reverberation chambers are used, for example to determine the absorbance of various materials and to measure sound power. Since these measurement chambers are less relevant to analyzing noise, we shall not discuss them further.

6.4.3 Other measurement chambers and booths

Even when it is not necessary to obtain recordings and measurements that are compliant with international standards, it is nonetheless necessary to create

Fig. 6.17: Studiobox

an environment for reproducible noise measurement. Soundproofed booths that are completely suitable for such measurements are now available.

Figure 6.17 shows an example of a "Studiobox" isolation booth. Even though this modular type of booth was developed to permit hearing free of interfering noise, it is also suitable for recording airborne noise. However, compromises must be made in the borderline ranges of audible sound, and in doubtful cases it may be necessary to fall back upon anechoic chambers since the damping properties of soundproof booths are not always sufficient.

6.4.4 Production environment

Measurements or test locations in manufacturing environments frequently present particular acoustic challenges. Because of the interfering noises from various production processes and the vibrations that are induced, one of the

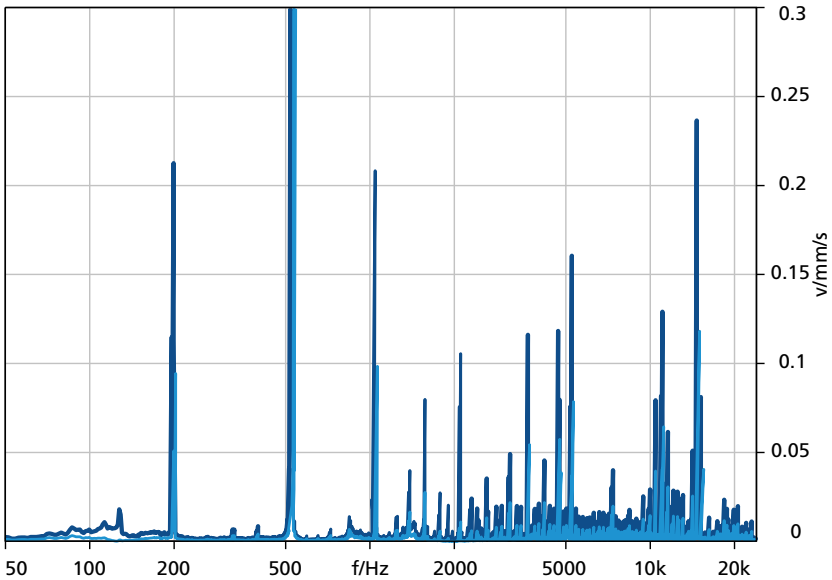


Fig. 6.18: Frequency spectrum before (dark blue) and after (light blue) optimization

main challenges faced in production is to isolate the test setup from the environment. Depending on the measurement task, this may also mean that isolation is only required when certain critical frequency ranges need to be measured. Measurement becomes particularly difficult when the noise that is emitted is not constant but changes over time.

The system used to hold the device under test presents a further challenge. With high-volume production and/or short cycle times, the mechanical characteristics of the device holding system can be extremely demanding. In addition to providing the isolation described above, the test setup must ensure that the device under test is held in such a way that measurements are reproducible. This in turn means that the retention forces, positions, and controls must be reproducible. Sometimes, though, it is more practical not to completely eliminate all external factors, but rather to keep the factors constant or only eliminate certain factors selectively at certain frequencies.

Figure 6.18 shows how the effects of a measuring setup were optimized for a given frequency spectrum.

7 Analysis of noises and vibrations

Currently, measurement signals are almost always digitized. As a result, a significantly broader spectrum of evaluation methods is available. In the "analog age," acquiring frequency spectra was a laborious task achieved by switching out filters. Nowadays it is easy to obtain a wide range of frequency spectra if a sufficiently powerful computer is being used.

Despite the wide variety of analytical methods, it is important to understand the fundamental principles that the methods are based on and to evaluate the results correctly. The sections below will help you to understand these principles as well as the possibilities and limits of signal analysis.

7.1 Goals and objectives

After noise and vibrations are measured and recorded reproducibly, as described in the previous chapter, the next task is to analyze the results. The goal of such an analysis is to determine

- the composition of a noise or vibration (what components are contained in the signal?), and
- to identify the origin of these components (what signal component can be attributed to what equipment component?).

In this way the cause of a noise or vibration component can be explained so that it can either be eliminated or reduced. It does not matter whether the analysis is based on a structure-borne noise signal or an airborne noise signal. Given all of the analysis options that are available, it is always possible

to assign a signal component (i.e. of a noise or vibration characteristic) to a part or event. This only becomes difficult in the case of nonlinear phenomena (friction).

The results of an analysis can then be used to improve a product or the manufacturing process required for product or to develop a criterion for standard production testing. So it is important to be able to perform a noise and vibration analysis efficiently and arrive at a clear and correct result. The standard analysis methods are described in detail below to give you the knowledge you need to use them effectively.

7.2 Basic principles of signal analysis

It is absolutely necessary to have an understanding of some of the physical principles of signal analysis in order to develop a broad understanding of how it can be applied. So we shall now take a closer look at these principles.

7.2.1 Signal processing

Any signal $u(t)$ that is periodic over time can be represented as a sum of trigonometric series of sine and cosine oscillations. These oscillations are defined as

$$u(t) = A_0 + \sum_{n=1}^{\infty} A_n \sin n\omega_0 t + \sum_{n=1}^{\infty} B_n \cos n\omega_0 t \tag{7.1}$$

Where:

$$\omega_0 = 2\pi f_0 = \frac{2\pi}{T} \tag{7.2}$$

The purpose of an analysis is always to break down the signal that is obtained from a noise event into such a sum containing individual summands. These summands – each of which represents a frequency –

must then be assigned to a component or product characteristic. This is precisely why it is important for the analysis to carefully consider all of the mechanical and physical conditions that apply to the device under test. This means that all of the mechanical contact surfaces at which cyclical forces can occur as well as all radiating surfaces that are capable of oscillating must be evaluated individually and critically. With the help of this information many factors can then be identified, evaluated, and possibly even ruled out through a more detailed analysis of possible causes.

Among the most important theoretical principles are the sampling theorem and Heisenberg's uncertainty principle. Both are decisive in the correct selection of test parameters. These parameters include in particular the time intervals between two measured values ("resolution") and the entire measurement duration, something that must already be taken into account when the testing or measurement is planned. In this case Heisenberg's uncertainty principle states that the maximum frequency resolution corresponds to the reciprocal of the measurement duration. This means, for example, that a measurement duration of > 1 s is required in order to separate two frequency lines at an interval of 1 Hz.

The Nyquist and Shannon sampling theorem moreover states that the scanning frequency of a measurement must be greater than twice the desired resolution.

$$f_{scanning} > 2 \cdot f_{\max} \tag{7.3}$$

It is also necessary to take into account aliasing effects. When analog signals are digitized, these aliasing effects reproduce low-frequency components that are greater than the maximum frequency resolution determined per Nyquist and Shannon. These effects must be suppressed by placing high-pass filters ahead of the circuitry used to digitize the measured data.

7.2.2 Filters

The use of filters, regardless of whether they are installed in the circuitry or are digitally calculated, helps to eliminate interference signals or to isolate certain components of the measurement signal better.

Weighting, which was described earlier in chapter 5.3.1 (see p.72 ff.), is used to bring measurement closer to the actual human auditory impression. In this way the physiological characteristics of human hearing are emulated. These filters also attenuate or amplify particular frequency components.

When filters are used, it is important to consider the effect that they have on the absolute level of the measurement.

7.2.3 Windowing

The Fourier frequency analysis assumes that the sound being analyzed is infinitely long and **continuous**. Since this cannot be achieved in reality, so-called window functions are used to obtain an artificial periodization of the measurement. If we did not do this, leakage effects (appearance of side bands in the frequency spectrum) would occur in the frequency analysis.

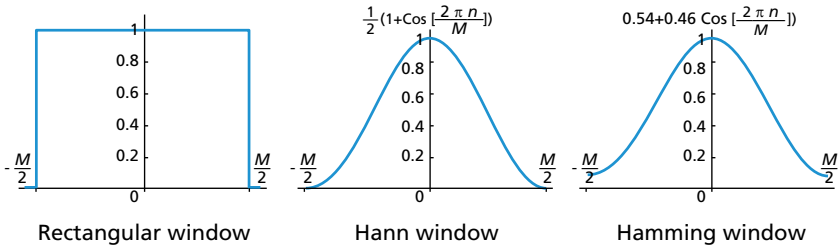


Fig. 7.1: Examples of window functions

Simply stated, windowing corresponds to a "fade-in" at the beginning of the measurement signal and a "fade-out" at the end. The most well-known window functions are the rectangular window, the Hann window, and the

Hamming window (Fig. 7.1). These functions are also the ones that are used most often in noise and vibration technology.

7.3 Basic methods of signal analysis

This section provides an overview of the methods than can be used to break a measurement signal down into its individual components and to describe each component with its corresponding frequency and amplitude. It is then possible, using the technical characteristics of the product, to arrive at conclusions regarding the sources of defects or undesirable noise. The components of these analyses can also be used as characteristics that precisely describe a specific noise event or record it as measurement data.

7.3.1 Fast Fourier transform

The infinite series of trigonometric functions described in Equation 7.1 is called a "Fourier series." The development of a function into its Fourier series is referred to as "harmonic analysis." In practical applications, the process is often stopped after a finite number of terms, and this way the function is approximated by a trigonometric polynomial. The calculation of the individual coefficients of each individual frequency component is very complex. Therefore, in real-world cases the individual coefficients are calculated recursively beginning with a known final value, thus reducing the computational work. This method is referred to as a fast Fourier transform (FFT). In an FFT, a given time signal ($a(t)$, $v(t)$, $x(t)$) is converted from the time range into the frequency range, so that one obtains the frequencies that are contained in this signal with the corresponding coefficients (amplitude factors) (Fig. 7.2). We shall not delve into the theoretical basis of the FFT here. Details may be found, for example, in [7].

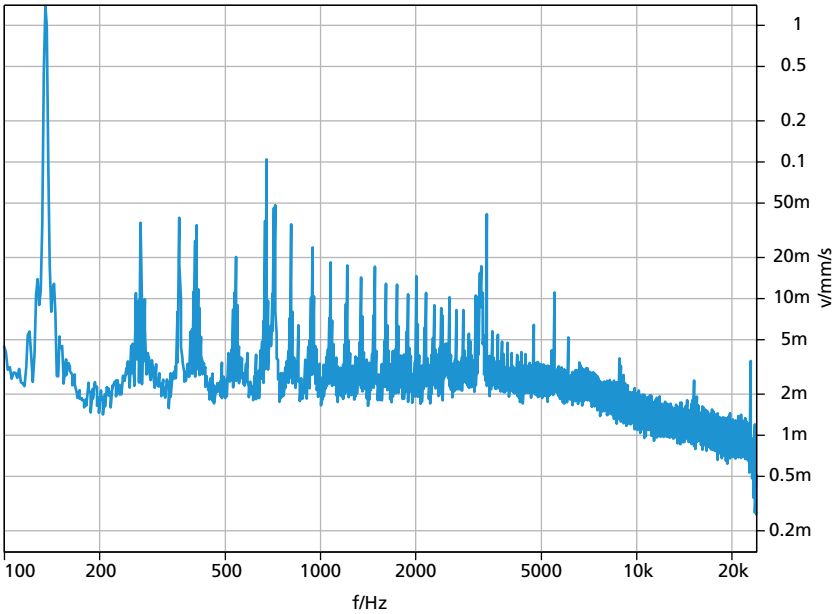


Fig. 7.2: FFT of a structure-borne noise measurement

7.3.2 Fast Fourier transform vs. short-time Fourier transform

The Fourier transform over time, which is also called a short-time Fourier transform (STFT), uses variable time windows to analyze the frequencies in the time signal. This makes it possible to also represent frequency changes over time in a diagram. Resolution is limited in this approach. Using this procedure, only one of the two parameters (time, frequency) can be increased at will. An increase in the resolution of one parameter always necessitates a decrease in the resolution of the other parameter.

7.3.3 Octave / one-third-octave analysis

The octave band and the one-third-octave band are among the narrow-band spectra. In the octave or one-third-octave analyses, we do not rep-

resent the individual frequency components according to their maximum resolution, but rather we consolidate specific frequency ranges into bands. An octave always corresponds to the frequency ratio 1:2. A one-third octave (1/3 octave) analysis subdivides this range into three further bands, and therefore corresponds to a ratio of 3:4. The analyzed bands, however, are not equivalent to musical octaves. In the analysis, the fundamental tone 'C' always corresponds to the central frequency of the corresponding frequency band.

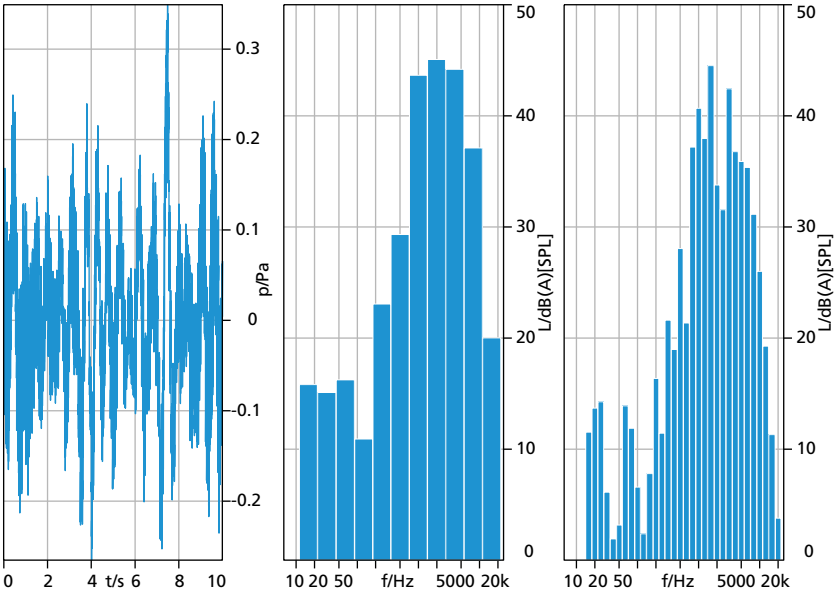


Fig. 7.3: Time signal (left), octave analysis (center), one-third-octave analysis (right)

Most analytical tools now offer additional subcategories of octave analysis, extending down to 1/96 octave analysis. Octave analysis is used in areas in which frequency ranges whose amplitudes must not be exceeded have been defined. However, this analysis plays a lesser role when one is analyzing the

causes of a noise. Figure 7.3 shows an example of an octave analysis and a one-third-octave analysis of an airborne noise signal.

7.3.4 Order analysis

Order analysis is a special type of Fourier transform. In this analytical procedure the frequency axis is related to the rotational speed of the motor and is normalized. This means that the sections on the abscissa no longer correspond to the frequency in hertz but rather, for example, to multiples of the motor speed or the shaft output speed in the case of gear drives. A basic requirement for such an analysis is that the motor speed must be recorded simultaneously. The resolution that is used when scanning

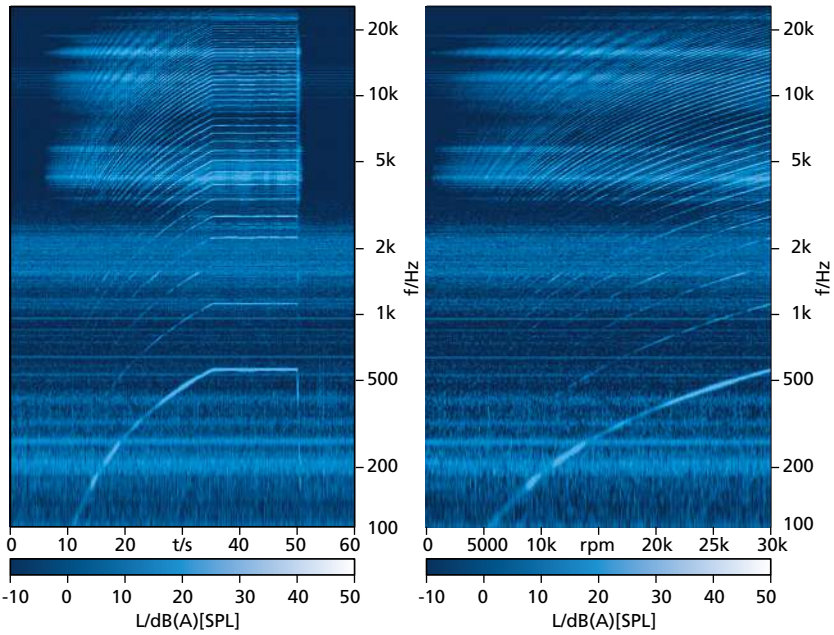


Fig. 7.4: Comparison of FFT vs. time (left) – FFT vs. rotational speed (right)

the motor speed must correspond to the application. Once again, in order to achieve a high resolution in the order axis, the scanning and resolution of the rotational speed signal must be correspondingly high.

This method is especially advantageous when variations in rpm occur and one wishes to compensate for them in the analysis or when two operating points having different rotational speeds are to be compared to each other directly. Order analysis can also be used when recording a run-up in order to make it easier to assign the individual frequency components. Figure 7.4 shows the differences between an order analysis (Fig. 7.4 right) and a frequency analysis over time (Fig. 7.4 left).

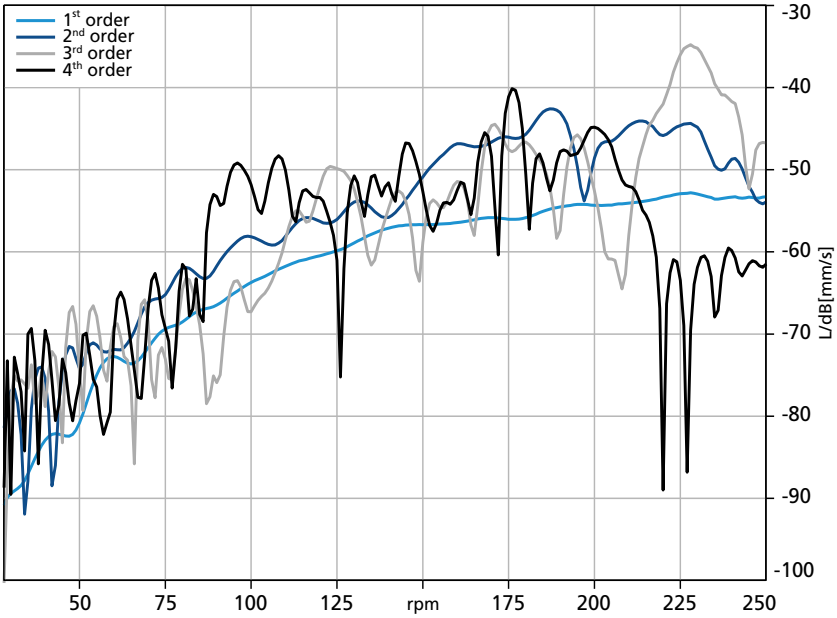


Fig. 7.5: Order level vs. rpm (1st–4th order)

Similarly, with the aid of the rotational speed information it is possible, for example, to represent the level of a particular order relative to the rotational speed in order to show resonance increases caused by imbalance, as in this example (Fig. 7.5). In addition to the effect of the imbalance, which correlates with the rotational frequency, additional harmonics of the rotational frequency are shown, so that other effects can also be analyzed.

7.3.5 Wavelet transformation

In wavelet transformation, in comparison with FFT, so-called wavelets are used. These wavelets are functions that are limited in terms of time and frequency. The individual diagrams shown in Figure 7.6 provide some familiar examples of wavelet functions.

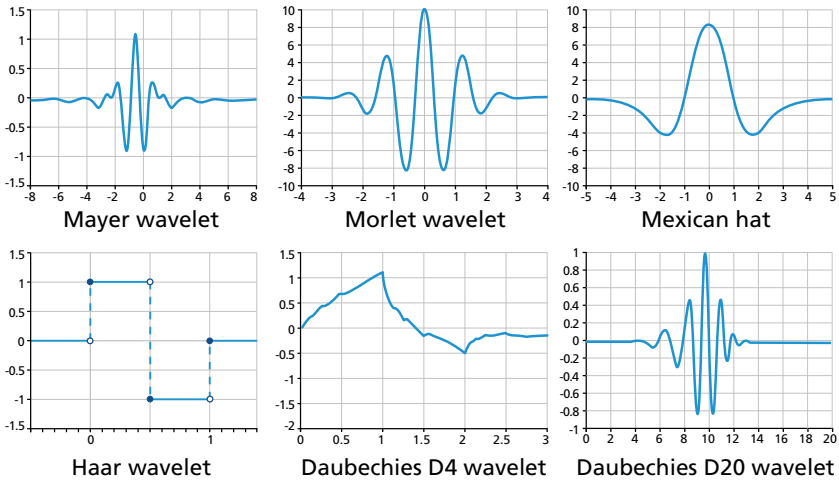


Fig. 7.6: Wavelet functions

Using the location of the functions, it is possible when scanning the measurement signals to scale both time and frequency and in this way achieve a

very high time resolution in all frequency ranges (Fig. 7.7). One of the advantages of this localization is that no windowing needs to be performed in the calculation.

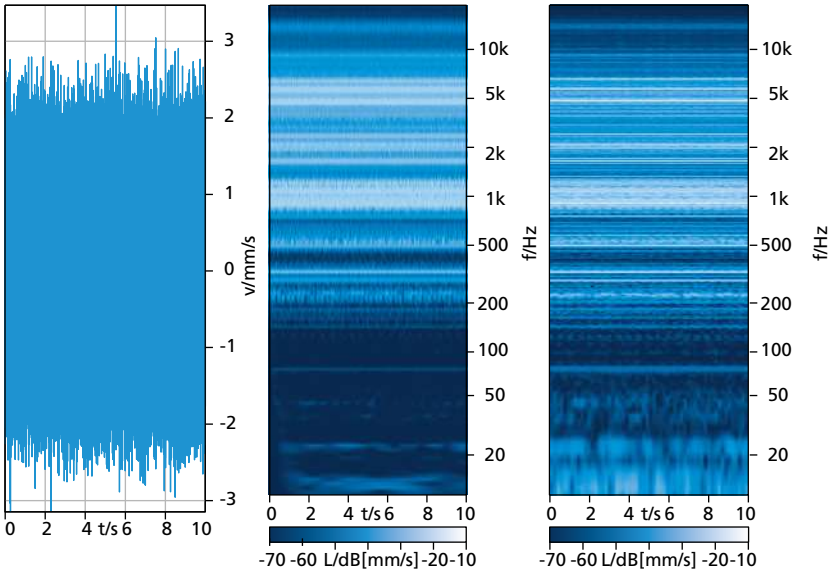


Fig. 7.7: Time signal (left), STFT (center), wavelet analysis (right)

7.3.6 Envelope analysis

Envelope analysis (Fig. 7.8) has its origins in communications technology. Here, low-frequency useful signals are communicated with the aid of modulated high-frequency signals. The receiver demodulates the signals again in order to be able to utilize the useful signal. The same principle is used here in the analysis of vibration signals. In the case of impactive excitations of components occurring as a result of damage in roller bearings or gear components, high-frequency structure resonances are excited. However, the resonance frequency is not of interest in interpreting the noise, but rath-

er the frequency with which the excitation is repeated and the amplitude of the excitation. This repetition frequency can be analyzed by means of demodulation.

Fig. 7.8: Envelope analysis process

When the modulated frequency range is analyzed – in other words, the resonance frequencies that are excited by an impact – the subsequent bandpass filtering system can be set in various ways. There are three different procedures, which will be mentioned here but not explained further:

- The filter range corresponds to the structure resonance range.
- The filter range lies outside of the structure resonance range.
- The filter range is well above the mean frequency range.

Figure 7.9 shows the effect of the envelope analysis on a signal modulated on a low frequency (upper left). While the low-frequency modulations cannot be seen in a direct Fourier analysis of the measurement signal (top right), they are readily apparent after the envelope curve is generated (bottom left). We can see that generating the envelope curve eliminates the harmonic components (for example: the motor rotation frequency at 150 Hz) and the modulated components and impactive excitations can be identified (bottom right). Generating the envelope curve allows us to see an impactive excitation in the frequency spectrum at 21 Hz. If we know the ball pass frequencies, this excitation can be attributed to a particular component in the gear drive.

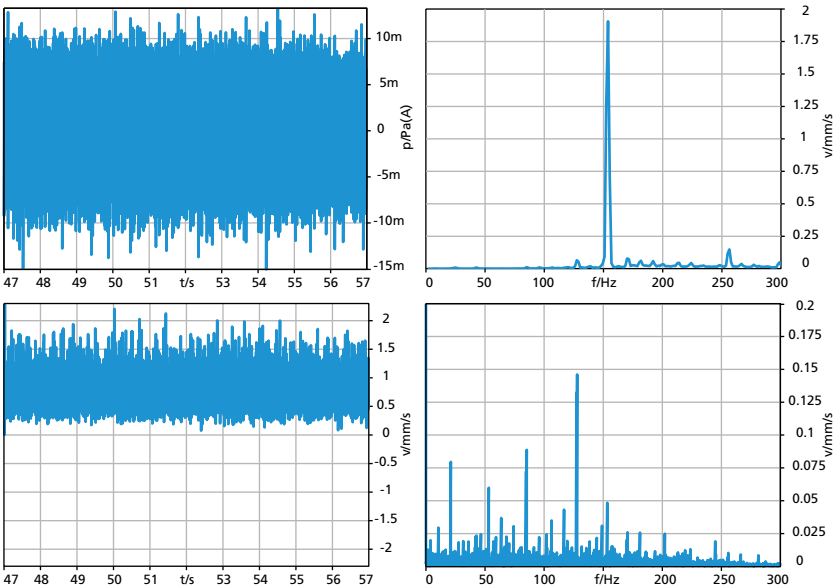


Fig. 7.9: Envelope analysis

7.4 Methods for noise and vibration analysis

The following pages describe useful ways to utilize the basic methods of signal analysis described above to investigate noise and vibration events. They explain which procedures can be used to obtain particular results and which methods can be used to solve a given problem effectively.

7.4.1 Investigation of the stationary operating point

An investigation of the stationary operating condition is easily performed and is suitable for revealing product changes or defects. In the case of motors, for example, the following defect patterns can be analyzed.

- Damage to ball bearings
- Commutation deviations (mechanical and electronic)
- Brush defects
- Gear damage (upon meshing)
- Imbalance
- Alignment errors
- Modulations (FM and AM), and much more

If the drive can be recorded over a "long" period, one obtains a very good average of the spectrum and very high frequency resolution. In this way, the individual frequency components can be easily distinguished from each other and attributed to the various causes. However, the natural frequencies of the motor are difficult to analyze with this method. If the natural frequencies are not previously known, it is almost impossible to recognize such frequency lines in the stationary operating point.

7.4.2 Run-up analysis

The vibrations and noises produced by rotating electric motors have three fundamentally different components in their frequency spectra:

- Components proportional to rotational speed
- Components independent of rotational speed
- Resonance increases caused by excitations of component natural frequencies

If one determines the frequency spectra of an airborne or structure-borne noise signal as a function of the rotational speed – for example during run-up or run-down – these three different components can be visualized very easily (Fig. 7.10) and distinguished from each other.

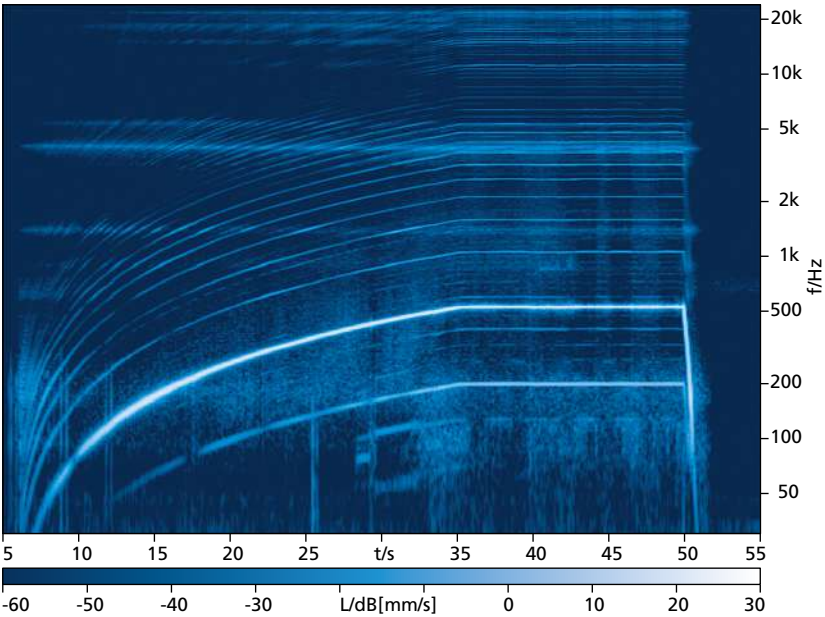


Fig. 7.10: Frequency spectrum vs. time (run-up)

Measuring a motor under different conditions (such as rotational speed acceleration, changing loads) provides more accurate information on its oscillatory behavior in the actual installed condition because motors are seldom operated at constant rpm, or they must in any event pass through a particular rotational speed range in order to reach their operating speed. In this way, potentially critical ranges containing resonances can be avoided or even eliminated. A run-up analysis is used with severe rpm- or load-dependent noise behavior. The natural frequencies or critical operating points can be identified by means of an analysis of frequency over time. If it is possible to record the rotational speed of the motor, the capabilities offered by an order spectrum should be utilized. Here too, it is easy to analyze the same defect patterns as at a stationary operating point. To achieve a level of resolution that can be evaluated, the variable component must be changed very

slowly. For this reason, run-up measurements can take several minutes to complete.

In addition, a run-up or a series of run-ups can be used to obtain detailed information on the operating behavior of the motor. For example, the operating points at which the motor runs as quietly as possible or at which certain frequencies do not occur can be determined. Figure 7.11 shows the airborne noise level of a motor-gear combination relative to rotational speed and load. To accomplish this, a number of run-ups were recorded under constant load and depicted in a diagram. A graph of this type allows one to identify the operating points at which the motor can be operated as ideally as possible. At the same time, operating ranges that should be avoided can be recognized.

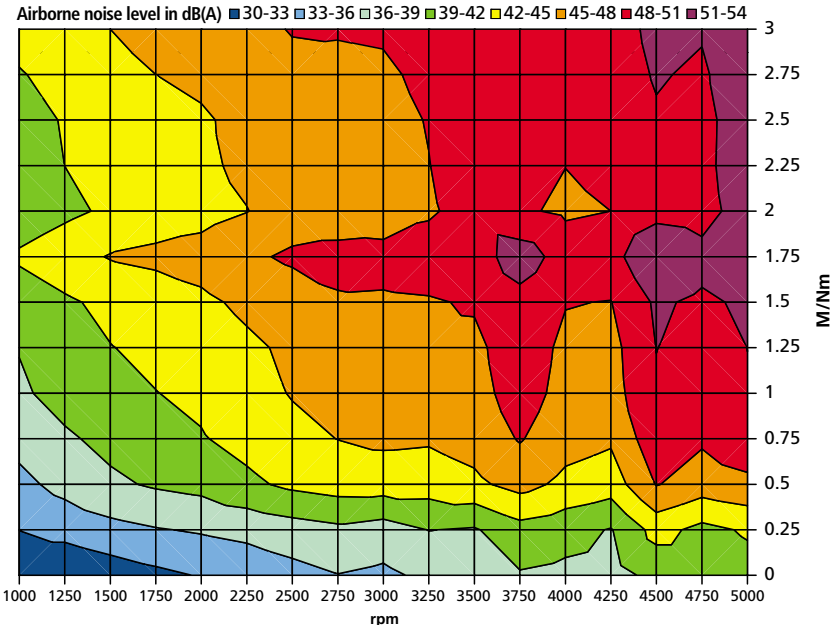


Fig. 7.11: Airborne noise level over rpm and torque

7.4.3 Natural frequency analysis

Depending on the objectives one has, various types of natural frequency analyses can be performed. Basically, there are two ways to do this: observing the motor's behavior in operation or observing its behavior when it is excited from the outside. In the first case, the natural frequencies that are excited as the motor accelerates during a run-up can be recorded and graphed (see Fig. 7.10, p. 120). The advantage of this method is that it is very easy to implement and that only those natural frequencies that are relevant to actual operation are determined. At the same time, the natural frequencies can be assessed critically through a subjective evaluation. One disadvantage of natural frequency analysis is that the evaluation range across which the natural frequencies are recorded is limited. If there are changes in operating behavior, an analysis of this type must be repeated, and the new operating points must be analyzed.

In the event of excitation from the outside, the motor can be excited via an external source with an appropriate random noise, sweep, or pulse, and the frequencies that occur in response to this excitation can be measured. This external excitation can occur in various ways:

- Via airborne noise (loudspeaker)
- Mechanically, using a piezo actuator or a pulse hammer
- By supplying an alternating current having an adjustable frequency to the motor winding(s)

The decision as to which excitation type is selected depends on the expected natural frequencies and the size of the component. One advantage of excitation by means of loudspeakers is that the excitation is free of feedback. However, the frequency response curve of the loudspeaker must be known. When exciting the motor with variable frequencies, it is important to be aware that a great deal of energy is required for excitation at high frequencies. Therefore, this type of excitation should only be used for relatively large surfaces. When an impulse hammer is used to produce the excitation, individual components can be systematically excited at individual locations, and this can even be done in various directions. When

excitation is systematically applied from the outside, in comparison with excitation in operation, all possible natural frequencies can be determined, and it is easier to estimate the effects that will occur upon changes in operating behavior.

7.4.4 Transfer path analysis

Transfer path analysis (TPA) is used to determine the impedances of transfer paths from the point at which the force is applied all the way to the radiating surface, or from a given point of transfer to a different component. Based on this information and the results that are obtained, the transfer function can be determined between two points. Optimization measures can then be simulated with the aid of this transfer function, and the consequences for the component can be estimated. Time-consuming and expensive sample test setups are no longer needed, and the effectiveness of measures taken can be verified in theory.

7.4.5 Operating vibration shape analysis

Operating vibration shape analysis can be used to determine the frequencies and vibration shapes (vibration modes) of a motor while it is operating. The advantage is that the locations and causes of problems in troublesome products can be determined quickly, and possible corrective measures can be identified. An analysis of this type is generally performed when the motor is operating at a constant speed. Large speed changes result in motor frequency errors and, therefore, errors in the vibration shapes. The measurements used to perform the operating vibration shape analysis can be carried out using accelerometers or a scanning vibrometer. The advantage of using accelerometers is that a number of sensors can be placed on the object under test in order to permit simultaneous measurement. Here, it must be remembered that attaching the sensors can have an effect on natural frequencies and amplitudes. When a scanning vibrometer is used to obtain measurements, the mass of the device does not affect the test, and significantly smaller measuring points can be achieved than would be possible if accelerometers were used. However, the use of scanning vibrometers rarely makes it possible to measure a

number of points at the same time. Instead, consecutive measurements must be performed. The required measurement time therefore increases rather substantially. During the measurement period, steps must be taken to ensure that the rotational speed and test conditions (e.g. temperature, viscosities, etc.) in the motor do not change. The number of test points and therefore the resolution of the operating vibration shapes depend on the frequencies that occur and the associated wavelengths. Performing a simple calculation before making any measurements makes it easier to assess the amount of work that will be required and the sensors or measuring time that will be needed. Figure 7.12 shows an operating vibration shape that was recorded with the aid of a scanning vibrometer.

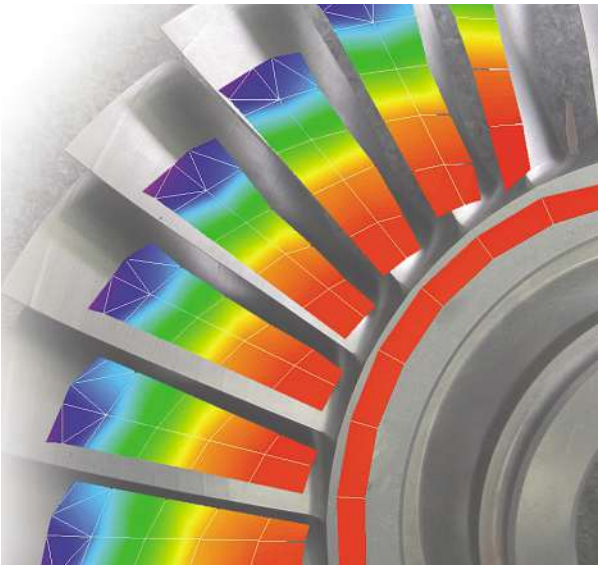


Fig. 7.12: Operating vibration shape of impeller

Because of their geometric shape, small electrical motors often behave like cylinders. The operating vibration shapes can be determined relatively easily on such cylindrical objects. To do this, one measures the radiated ra-

dial noise or vibration components around the motor using a microphone or a scanning vibrometer. Based on the number of vibration nodes and antinodes that are identified, one obtains the modal order number of the vibration and can conclude that a standing wave is present around the circumference. If one does not observe any pronounced vibration nodes and vibration antinodes, the vibration is most probably a circumferential radial sound wave. Two signal sensors are needed in this case to be able to determine the number of nodes and antinodes. One of the two signal sensors remains in a fixed position at a given point on the circumference of the motor; the second sensor is moved around the motor. Both sensors measure the same signal, however, the phase of the signal is shifted with respect to time. If one generates the difference signal between the two sensors, one obtains the vibration nodes and vibration antinodes as the movable sensor is traveling around the motor, just as a standing wave would be obtained if there were only one sensor.

7.4.6 Overview of analysis methods

As already noted, the purpose of noise analysis is to determine the composition, spatial distribution, and the origin of the components of a noise event. Noises come in an infinite variety of forms. The individual methods used to analyze noise are as varied as the noises themselves when it comes to their ability to detect characteristic components of a noise.

If one has a number of analytical methods to select from, the key question in each case is which method can be used to the best advantage. Generally, there is no clear unambiguous answer to this question. However, Table 8.1 (see p. 145) may be helpful in reaching this decision. Figure 7.13 shows the curve of a noise event over time, for example the run-up of an electrical machine after it is been turned on. All of the analysis methods stated in this chapter can be used to analyze the signal. However, the validity of the individual methods varies greatly. The right analysis method will depend on the question that must be answered in each individual case.

Some of the analysis methods are only effective when the noise is stationary. The earlier example (see Fig. 7.10, p. 120) shows a time segment taken from the stationary operation of the motor following run-up after the

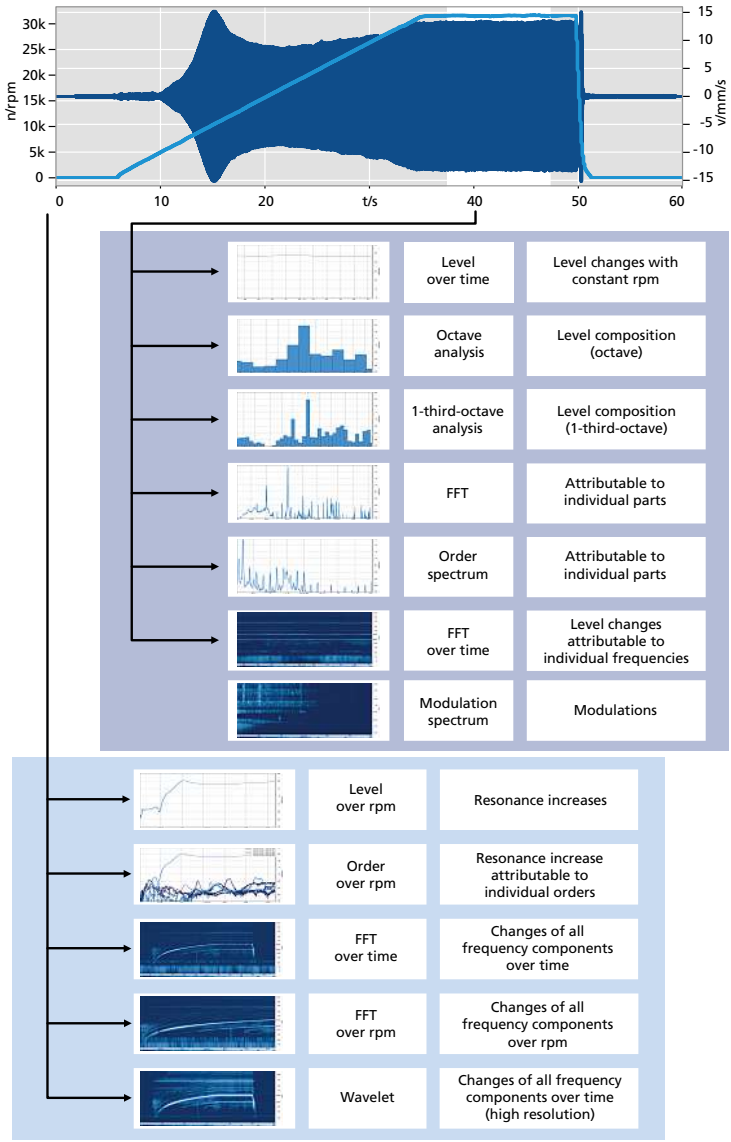


Fig. 7.13: Results of various analysis methods on a measurement signal

stationary final rpm has been reached. In such a case, useful results are obtained from various level values (sound level, acceleration level, etc.) octave analysis and one-third-octave analysis as well as spectral analyses with FFT and also rotational speed orders.

If we look at the nonstationary case – for example the entire motor run-up process – we quickly obtain relationships between rotational speed changes and changes in the noise being radiated. The analysis method often allows natural frequencies and noise components that vary according to motor rpm to be assigned to the components, and instances of increased resonance can be identified. Analysis methods that are particularly suitable are analyses of levels, modal order numbers as multiples of motor rotational speed, and FFT as a function of time or rotational speed.

The great variety of noises that is encountered does not make it possible to describe a noise event adequately using a scale that merely ranges from "loud" to "soft." For this reason, researchers often use a "quality" to describe noise with the assumption that this quality is generally recognized. This includes terms like "squeaking," "rattling," or "wobbling." Table 8.1 lists a number of noise patterns that occur frequently and are usually identified by means of such associations with familiar noises. It also contains information on which analysis method is most suitable for recording the noise patterns based on actual experience. Table 8.1 shows that there usually are a number of possibilities for analyzing a specific noise event. Frequently, the method that is chosen depends on the measurement equipment that is available.

Separate measuring equipment is not required for each individual analysis method. It is often possible to use a single measurement system to perform a number of analysis methods. In actual practice, a lot can be accomplished with just a few analysis methods. Using FFT, FFT versus time, or FFT versus rotational speed, it is possible to identify the components that are responsible for numerous undesirable noises.

Table 8.1 also provides an overview of the analyses and operating conditions that are needed to successfully identify and quantitatively characterize the noise properties described in the table using measurement equipment. The purpose of the table is to point the investigator in the right direction. However, there are also certainly other noise events for which other analysis methods are more suitable.

8 Testing vibrations and noises

8.1 General principles of vibrations and noises

The purpose of analyzing vibration and noise phenomena as described in the previous chapter is to determine the cause of objectionable noises. Once the cause is known, it can ideally be corrected by design modifications. If an effective corrective measure is not possible, so that basically the defect cannot be prevented, then the quality status of a product must be determined by means of testing.

Basically, there are two possible ways to classify products as either "good" or "bad." If the cause of noise problems is a defective part, this part can either be tested in standard production before it is installed in the device, or, optionally, noise testing can be performed on the part in the final installed condition. The decision to use one method and not the other usually depends on economic factors. It is generally easier, more economical and more sensible to check geometric parameters on an individual part than it is to perform sophisticated noise tests. If the bad part is detected early in the process, no further resources are wasted on the part. An additional advantage of testing on an individual component is that reworking is avoided.

If one nevertheless decides to perform noise testing, it is important to remember that this kind of testing differs in many ways from other tests, in particular standardized tests. The main differences lie in the variables being measured, measurement accuracy, calibration, and in the test setup.

While standardized tests (such as those for dimensions, geometry, and electrical parameters) and frequently test setups are clearly described and specified in industrial standards, this is not the case with noise tests. Here, the variables (airborne noise, oscillation velocity, oscillation acceleration, etc.) and the characteristics (level, weighted level, etc.) are not definitively

specified, and they must be selected on a case-by-case basis. Likewise, with standardized measurement procedures, the measurement accuracy and permissible tolerances are specified, but with noise testing they are not uniquely defined because the upper limits that are specified are usually merely subjective. Moreover, with noise testing the question of calibration presents problems that generally cannot be solved. Individual measurement instruments can of course be calibrated, but there is no test standard for the test itself that can be used to check and calibrate the entire noise test beginning with the mounting of the devices under test and ending with a good/bad assessment.

So it is virtually impossible to base noise testing on published standards. Instead, an entire test procedure must be developed for each individual case. The elements that go into this are:

- Determination of repeatability
- Determination of reproducibility
- Limit samples that can be used as the test standard
- A standard procedure for noise testing
- A test stand including mechanical systems for mounting the devices under test, possibly devices to apply loads, sensors, and evaluation circuitry

These elements are described in greater detail in some of the following chapters (8.3.1, 8.3.2, 8.4, 8.5, 8.8).

8.2 Noise testing in standard production

A noise testing procedure that is used in standard production must meet a wide array of requirements in order to actually achieve the goal of clearly distinguishing between "good" and "bad." These requirements include:

- Testing within the production line cycle time: Within one production line cycle time (generally 5 to 10 seconds), the device under test must be placed in the test fixture and subjected to the loads that one wishes to apply, the operating point must be reached, a measurement and evaluation must be

made, and the device must be removed from the fixture and transferred to the next station.

- Testing under the conditions that are present in the manufacturing environment: In manufacturing areas, ambient noise and vibrations caused, for example, by fork loaders or other machines and production equipment are present. Such noise and vibrations must not affect the test equipment and results.
- The difference between "good" and "bad" must be easy to recognize: This assessment must not depend on the qualifications or "experience" of the production personnel because this would unavoidably result in erroneous judgments and increased rejection rates or customer complaints.
- Results must not be dependent on the condition and mood of the tester on a given day: The test result must not change depending on the time of day, possible tester fatigue or distraction; different testers must always come to the same result.
- The "good" correlation or the "bad" correlation should be 100%: In other words, all products that are actually good must be recognized and evaluated as being "good," and all that are actually bad must be recognized and evaluated as being "bad."
- Reproducibility: Repeated testing always produces the same result.
- Calibratability: The identical result is in fact always achieved in two identical test devices.

These apparently obvious requirements are often very difficult to achieve in actual practice. Therefore, they must be verified explicitly before the test method is released for standard production. If the specified criteria cannot be met with absolute certainty, then at the very least the limits and risks of the test method must be known.

Given these problems, it is always best to first try to identify the causes that are being addressed in the noise testing and to try to replace the noise testing by the testing/inspection of geometric parameters. If this is not possible because geometric deviations do not occur until the manufacturing process is performed (for example, when plastic gears are damaged or deformed in the process of assembling the gears into drives), then noise testing must be used.

8.2.1 Subjective testing

If problems with noise occur in drive components in standard production, the most obvious thing to do is to listen to these components and evaluate them subjectively based on one's own experience. If the noises are caused by an abnormal condition in the component (for example, insulation parts rubbing on the stator), a subjective evaluation may be all that is needed. In such cases, however, the root cause must be considered, and the defect that is identified as "insulation dimension exceeded" must be checked. This example, however, does not actually involve a noise test, but rather a quality test that is performed with the aid of a noise that either occurs ("bad") or does not occur ("good"). On the other hand, a "true" noise test must be able to determine a value with a particular degree of accuracy, something that is extremely difficult to achieve in a purely subjective evaluation.

Very few individuals have the gift of perfect pitch. Such individuals are able to evaluate sounds and frequencies correctly, but even they cannot correctly "measure" amplitude. For everyone else, this is even more difficult. It is extremely difficult for humans to "store" a threshold between "good" and "bad" because this threshold tends to shift for numerous reasons. The individual's judgment is affected by his hearing ability as well as his physical condition on a given day, mood, fatigue, and fluctuating level of distraction and concentration. Such problems are exacerbated by fluctuations in the frequency and amplitude of the noise that is to be evaluated and possibly by the presence of a number of different defect characteristics that must be evaluated and are present to varying degrees (for example: gear noise and bearing noise occurring in combination with rubbing noises).

The same criteria that apply to final testing also apply to receiving testing conducted by the customer. Here too – for example due to the change in test personnel – parts that test "good" in final testing by the motor manufacturer are occasionally judged to be "bad" by the customer, resulting in a customer complaint.

8.2.2 Objective testing

Given the difficulties associated with subjective testing, we naturally often want to implement objective testing. We want to be able to pop our device into some sort of test apparatus, press a button, and get a "true" result. But the bottom line is that it is not possible to achieve objective noise testing because there are no objective requirements for "good" and "bad." Our specification is always derived from a subjectively evaluated random sample. This means that a subjectively evaluated random sample – which serves as our reference – needs to be evaluated objectively! In order to at least come close to "objectivizing" this random sample, we at least need to perform multiple evaluations with a number of testers, as described in chapter 5.4 (see p. 78 ff.).

In order to perform the ideal bench testing described above, **all** of the individual faults that occur must be described and converted into test criteria. However, our test apparatus can only test properties that have been established as test criteria. If a new defect that has not occurred previously appears, our test apparatus cannot detect this defect. On the other hand, with subjective testing it is at least possible that the new defect will be noticed by the tester and that the tester will be able to respond accordingly. This inherent problem with objective tests – namely their inability to recognize previously unknown defects – can be reduced somewhat by using a test apparatus that is equipped with a sophisticated teach-in evaluation system. Here, it is possible to place limit samples in such a test apparatus and "teach" the apparatus what is "good." Therefore, all devices that deviate from this condition are initially rejected as being "bad."

The test apparatus must identify the defects in such a way that either the good correlation or the bad correlation is 100%. This means that either all good products must be evaluated as being "good" (bad products may be evaluated incorrectly) or all bad products must be evaluated as being "bad" (good products may be evaluated incorrectly). However, in the real world it is not possible to achieve 100% good and 100% bad correlation simultaneously.

8.2.3 Objectively supported subjective testing

Subjective testing in which the tester himself can call up a reference signal has proved to work well in actual practice. This reference signal is obtained from a device under test that was previously approved for use as a limit sample. With the aid of this reference signal it is relatively easy to reach a decision as to whether a part is "good" or "bad" by alternately listening to the reference signal and the specimen signal. A comparative evaluation of this type is easy for human beings, because human hearing functions as a relative measuring instrument, and not as an instrument that measures absolute values. This test avoids the disadvantages of purely subjective testing as well as those of purely objective testing and therefore offers the following advantages:

- It is largely independent of the condition and mood of the tester.
- It is largely independent of who the tester is.
- New defects that did not occur previously can be recognized.
- It can be implemented quickly, easily, and economically.

In the objectively supported subjective test, it is important to be sure that the reference signal does not change. Therefore, a limit sample must not be used directly for the comparative testing, since this sample changes due to aging and frequent operation. Likewise, factors such as the distance to the tester's ear, handling, etc. must be ruled out. This is accomplished by recording the noise from the limit sample in a test recording facility and storing it in digitized form (Fig. 8.1). The data that are preserved in this way are played back to the tester as a reference signal using an amplifier and loudspeaker. In order for the comparison to be auditorily correct, the signal from the device under test must also be recorded in the same manner and presented to the tester

Fig. 8.1: Supported subjective noise test

using the same amplifier and loudspeaker. Therefore, the noise that is directly radiated by the device under test is not used in the evaluation. In this way, human hearing can also be used to compare and evaluate structure-borne noise signals.

8.3 Capability of noise tests

As in other processes in manufacturing technology, indicator scores can also be used for statistical evaluation purposes in noise tests. The purpose of these indicators is to state the reliability with which the goals set forth in the specification (in this case a standard testing procedure) are met.

8.3.1 Reproducibility

Reproducibility means that a test result ("good" or "bad") can be achieved repeatedly with a test procedure and the appropriate test equipment. If the same device is repeatedly placed in the test apparatus and tested, the result should always be the same. This sounds simple and obvious, but is often very hard to achieve in actual practice. The reasons are:

- The device under test changes during repeated testing (due to heating, for example).
- The position of the device under test changes (contact points in the recording device).
- The interface with a load (such as a brake) changes.
- The acquisition of the signal changes (with piezo sensors in particular as a result of the mechanical mounting system).

Standardized procedures are used in quality standards to determine reproducibility. These procedures include in particular measuring system analyses (MSAs), for which three different procedures exist. In MSA procedure 1, the repeatability of the measuring system is determined. One reference part must be measured 50 times for each characteristic and each measurement range.

MSA procedures 2 and 3 are used to evaluate the measuring process. Procedure 2 relates to measuring processes that can be influenced by the operator. Procedure 3 relates to those that are not affected by the operator. The number of devices under test and the number of repeat measurements are determined with the specified measurement tolerances.

With noise tests, this MSA procedure may only be used to a limited degree in this form because a normal value that does not change over time generally is not available. Therefore, for practical reasons five to ten devices under test are used. Of them, a number are clearly classified as being "good" and the others are clearly classified as being "bad." The value of the test characteristic that is to be considered should lie in the borderline region between "good" and "bad." The devices under test are each inserted in the test apparatus and tested 10 to 20 times. The reproducibility R (or also the repeatability) is then obtained from the ratio

$$R = \frac{\text{Number of correct test results}}{\text{Number of tests}} \cdot 100 \quad [\%]. \tag{8.1}$$

Using the results of the tests for reproducibility, the so-called "good correlation" K_G and the "bad correlation" K_S can be determined. These correlations are:

$$K_G = \frac{\text{Number of correct "good tests"}}{\text{Number of tests with "good parts"}} \cdot 100 \quad [\%] \tag{8.2}$$

$$K_S = \frac{\text{Number of correct "bad tests"}}{\text{Number of tests with "bad parts"}} \cdot 100 \quad [\%] \tag{8.3}$$

The good and bad correlations generally deviate from each other (measurement tolerance!), and a value of 100% is rarely achieved. For testing in standard production, the maximum value achieved and the corresponding proce-

dure are selected; in other words, we test for "good" or "bad" depending on which method is most likely to yield a reliable result.

A widely held but mistaken belief is that changing the test limit will improve the method. In fact, making the test procedure "stricter" by shifting the good/bad limit to lower values in no way helps to ensure that more devices under test are detected correctly. The only result is that more parts must be scrapped because more parts that are actually good are rejected.

8.3.2 Calibratability

Calibratability is also referred to as reproducibility. This describes how well the test results that were achieved on a first test stand compare with those achieved on a second test stand. Reproducibility and good and bad correlations must be determined for both test stands. These three values from test stand 1 should agree as closely as possible with those from test stand 2. If agreement is not close, then the test method must be improved with regard to the device mounting system, load coupling, the measuring point, or the stability of the device under test; or the test must be completely called into question.

8.3.3 Example of validation of capability

The following example shows an MSA that was performed per method 1 and an MSA that was performed per method 2. Since this statistical evaluation was not developed for logarithmic values, the measurement data for the sound pressure level must first be converted to sound pressure. Statistical software (MINITAB) is used to evaluate and represent test results.

Name of tester	Measurement Data Documentation	FAULHABER
Fuchs, Thomas	**Method 1 – Repeatability**	
Date of analysis	Attachment to Form DFF-FO-157	**Location of analysis**
04.03.2010		**EMC Room**

Airborne noise level in dB(A)

1510.K0002

Procedure: Apply a standard (reference) 50 times, measure, and remove device under test.

Test equipment			Standard/device under test			Property		
Test equip. name	Airborne sound level meter		Name	Reference part		Specified	---	dB(A)
Test equip. ID no.	00868		Number	SNR 100604469614		UTL	50.0	dB(A)
Resolution	0.1	dB(A)	Actual mean	46.9	dB(A)	LTL	---	dB(A)
Test location name	SPC airborne noise level measurement TC module					Room temperature: °C		

No	Value
1	48.1
2	48.6
3	48.5
4	48.8
5	48.5
6	48.4
7	48.7
8	48.4
9	48.5
10	48.0
11	48.3
12	47.9
13	47.6
14	47.8
15	47.7
16	47.7
17	47.5
18	46.6
19	47.2
20	47.0
21	46.2
22	47.1
23	46.9
24	46.9
25	46.6
26	46.9
27	47.0
28	46.9
29	46.0
30	46.6
31	46.9
32	46.4
33	46.2
34	46.2
35	46.3
36	45.3
37	45.1
38	45.0
39	45.8
40	46.3
41	46.6
42	45.9
43	46.4
44	46.2
45	46.1
46	45.4
47	45.4
48	45.6
49	46.2
50	45.9

Evaluation	
Mean	46.922
Std. dev.	1.058
MIN	45.000
MAX	48.800
MAX-MIN	3.800
T	50.000
Min. res.	2.500
Actual res.	0.100

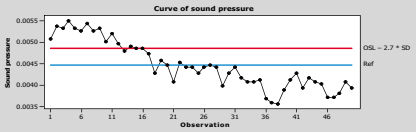
Cg	**N.D.**
Cgko	
Cgku	
Cgk	**N.D.**
p-Value	**N.D.**
Tmin	

The results are calculated in QMD!

MSA type 1: Airborne sound level measurement – calculated sound pressure

Instrument name: Airborne sound level meter
Test date: 4 March 2010

Report author: Dr. N. Bayat
OSL: 0.0063246
Other: ID no.: 00868



Statistical measures		Syst. measurement dev.	
Reference	0.00447	Syst. meas. dev.	-0.000000004
Mean	0.004469986	t	0.0000473370
SD	0.0005495682	p value	1.000
4 * SD (SU)	0.0021982729	(Test system mes.	
USL	0.0063246	dev. = 0)	

Remarks (procedure, annotations, noteworthy items)

A reference part that was measured 50 times was used for the procedure.
A measured value trend results from run-in effects. (Thomas Fuchs)

The calculation of dB data with MSA is not directly comparable.
Therefore, for the evaluation (see chart) the data were converted to sound pressure. *The characteristic was limited on the high-end (customer request).*
The run-in time must be provided with MSA type 2 since repeat measurements of the parts under test affect the result and run-in effects are present (see chart). (Dr. N. Bayat).

FOR INFO

The requirements established in DFF-PA0066 apply.
This evaluation sheet must be attached to Form FO-157 and therefore must be identified as also being valid and applicable.

Fig. 8.2: Example of proof of capability per method 1

Name of tester Fuchs, Thomas	Measurement Data Documentation Method 2 (expanded) – “With Operator Influence”	FAULHABER
Date of analysis 04.03.2010		Location of analysis EMC Room

Airborne noise level in dB(A)

1510.K0002

Procedure: Measure 25 parts, three times each. The data are evaluated by employees in QMD.

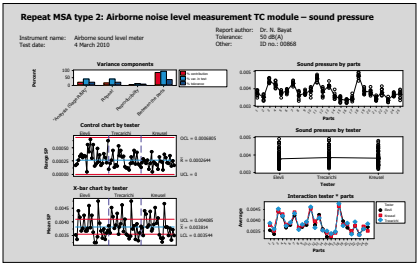
The testers must not be able to inspect the results obtained by the other testers.

Test equipment			Property			Environmental condition
Test equip. name	Airborne sound level meter		Specification		dB(A)	Room temperature: °C
Test equipment ID no.	00868		UTL	50.000	dB(A)	**Condition of test equipment**
Resolution	0.100	dB(A)	LTL		dB(A)	present
Name of test location	SPC airborne noise level measurement TC module					

Tester 1:	Ms. Elevli		
Date:	04.03.2010		
	Run		
Part	1	2	3
1	45.1	44.7	45
2	44.3	44.4	44.8
3	47.2	46.7	46.7
4	46.4	46	46.7
5	45.6	44.7	45.4
6	45.9	45.4	45.2
7	46.2	46.4	46.9
8	44.9	45.7	45.1
9	45.3	45.2	45.3
10	46	45.7	44.7
11	47.4	47.5	46.7
12	46.2	45.6	44.7
13	46.4	46.3	46.9
14	45.9	44.6	44.8
15	44.4	44	44.1
16	44.5	44.1	44
17	44.2	45.4	44.5
18	47.8	47.4	47.4
19	45.4	46.1	46.2
20	45	45.3	45.4
21	44.4	44.5	44.9
22	45.4	45	45.9
23	44.1	44.5	44.9
24	45	45.4	44.9
25	45.5	44.4	45.9

Tester 2:	Ms. Trecarichi		
Date:	04.03.2010		
	Run		
Part	1	2	3
1	45.7	45.6	45.8
2	45.1	44.8	45.3
3	46.8	47.2	47.3
4	47	46.2	46.1
5	45.5	45.3	45.6
6	45.7	46	45.6
7	46.7	46.5	46.8
8	44.3	45.6	45
9	45.9	45.1	45.7
10	45.2	46.1	46.2
11	47.6	47	47.2
12	44.9	45.7	45
13	46.2	46.4	46.1
14	45.1	45	44.8
15	44.9	45	44.7
16	44.6	44.4	44.6
17	44.8	44.5	45
18	47.2	47.6	47.8
19	45.2	45.7	45.4
20	46.2	45.7	45.2
21	44.4	44.8	44.3
22	45.9	45.4	46.3
23	44.4	44.1	44.6
24	45.1	44.9	45.3
25	45.8	45.6	45.5

Tester 3:	Ms. Kreusel		
Date:	04.03.2010		
	Run		
Part	1	2	3
1	45.5	45.3	45.8
2	44.5	44.9	45.2
3	47.1	46.6	46.8
4	46.9	46.5	46.5
5	45.8	45.2	45.8
6	45.8	45.6	45.8
7	47	46.8	46.6
8	44.6	45.5	45.6
9	45.5	45.4	45.4
10	45.7	46.1	45.6
11	47.2	47.2	46.8
12	45.7	46	46.1
13	46.2	46.4	46.4
14	45.4	44.7	44.6
15	44.2	44.8	44.8
16	44.2	44	44.6
17	44.6	44.7	44.5
18	47.6	47.5	46.9
19	45.7	45.2	45.9
20	45.3	45.8	45.7
21	44.8	45.1	44.6
22	45.4	45.9	45.1
23	44.6	44.4	44.2
24	45	45.5	45.6
25	44.7	44.7	45.1



Repeatability and reproducibility	
Total Gage %R&R	16.25

Remarks and notes

The units were run in for 2 min. before starting the MSA. (Thomas Fuchs)

The calculation of dB data with MSA is not directly comparable. ***Therefore, for the evaluation (see chart) the data were converted to sound pressure.*** *The test location is “OK” with regard to the maximum dB limit. (Dr. N. Bayat)*

FOR INFO

The requirements established in DFF-PA0066 apply.
This evaluation sheet must be attached to Form FO-157 and therefore must be identified as also being valid and applicable.

Fig. 8.3: Example of proof of capability per method 2

The example involves a test that occurs during production and that is used to monitor the airborne noise level. The customer wishes to have 20 motors tested per day. Figure 8.2 shows the evaluation of the repeatability determination in which the test specimen was measured 50 times.

Figure 8.3 shows the results of process 2, which is used to determine operator dependency. Twenty-five measurements were performed on the same part by the same person, and the results were compared.

8.4 Limit samples and test standards for noise tests

To be able to determine reproducibility and calibratability, it would generally be desirable to have a suitable test standard that could be used to generate clear and distinct values for the selected test characteristic in the test apparatus. However, such a universal test standard does not exist.

If one wishes to take this approach of using a standard for the test procedure, then this standard would need to be developed specifically for each individual case. The effort and expense that would be required would be rather huge, and it is questionable that the goal could even be achieved. Of course one can work with shakers, oscillation coils, piezo actuators and even dummy motors containing elements like shakers, oscillation coils, or piezo actuators. But since such standards do not match the device under test in all regards, one often does not achieve the desired ends. This is apparent when one sees how much variation is found in results from reproducibility tests – and in them the same device under test is being used again and again!

Limit samples offer an alternative to test standards. These are specially selected specimens that mark the boundary between "good" and "bad." But the question remains: How is this boundary determined?

First, the noises from a representative number of devices must be evaluated and classified. Selecting suitable devices to test is often difficult, though. It is nevertheless absolutely necessary to gather together a set of devices and classify them clearly as "good," "bad," and "borderline" with regard to the property being considered. This classification must be made separately for

each individual characteristic (e.g. gear noise, brush whine). Since this classification must be based on subjective judgments, it should never be done by a single person. At least three but no more than five persons should be involved. If a product is to be tested because the customer is dissatisfied with its acoustic quality, then the customer's decision-makers must be involved in this product classification team.

It is very important to work with the customer to develop well-defined terminology so that all of the parties are speaking about one and the same problem. Often this does not happen in actual practice: When the parties are dealing with a noise problem, they usually only refer to this problem in a vague way, and the staff member who is working on the problem is free to associate his own experience with what he perceives the problem to be. This vague way of referring to the problem can become part of company jargon. Such terms, even if they are only used in a single instance, take on a life of their own in the company. Persons outside of the company frequently need additional explanations in order to understand them. Therefore, it is important to describe a noise event in the most specific terms possible (see Table 8.1, p. 145).

Once noise limits have been defined, they must be properly preserved so that they can be used for a relatively long time. This means that such limit samples must be kept in storage, and it also means – and this is often much more important – that the noises produced by the limit samples must be recorded and preserved in a suitable format. Here, binaural recordings (such as those made using a dummy head) offer a good solution because they preserve the human auditory impression and do not require any additional processing of the measured signals.

If noise limits relate to calculated limit amplitudes of physical parameters or to levels established under law (e.g. occupational health and safety regulations), then a method that differs from that described above may be used. In such cases, there are guidelines and standards in which procedures and limits are already precisely defined.

8.5 Standard test procedure for noise testing

Before any noise testing is performed, a standard procedure for this testing must be developed. When this document is being prepared, it will be necessary to come to terms with problems that otherwise would not be encountered until much later during standard production testing. In this way, one can avoid developing a test apparatus that ultimately might prove to be unsuitable. In order for noise testing to be effective, it will be necessary to come up with a standard test procedure that covers 10 points. These 10 points are:

- Description of the test (what is the purpose, what part is to be tested?)
- Definition of the test apparatus
- Specification of the test procedure
- Definition of the test conditions
- Specification of the test variables (structure-borne noise, airborne noise, acceleration, velocity, …)
- Specification of the test characteristics (weighted A level, total level, crest factor, …)
- Evaluation of the test characteristics: The test is passed if the test characteristic x < limit value y
- Definition of the calibration method
- Definition of the scope of testing (testing of selected parts, 100% testing, …)
- Documentation

The reproducibility of the measurements on a test stand must of course be documented, however, this is not part of the standard test procedure but rather is part of the test stand acceptance procedure.

8.6 Selection of characteristics to be tested

As in any test, it is necessary with noise testing to identify the characteristic to be tested. A "correct" characteristic to be tested is one whose values correlate

with the acoustic behavior of the device under test in the final product. Using this characteristic to be tested, a noise pattern can then be clearly identified and a judgment can be made as to whether the part is "good" or "bad." However, if the wrong characteristic is selected, then the good/bad correlation will not be adequate. This will result in a situation that will be unacceptable for production: In order to be certain that only good parts are supplied to customers, it will certainly also be necessary to declare many parts to be rejects that in fact are good but unfortunately are classified incorrectly. The following three figures (Figs. 8.4, 8.5, 8.6) illustrate such problems.

Noise characteristics (level, perceived loudness, sharpness, etc.) frequently occur as normal distributions (Fig. 8.4). Whenever a measurement is made, however, it is important to consider that this measurement can only be made with a particular degree of measurement accuracy. When reproducibility and calibratability are considered, we find that the measurement accuracy in noise tests is generally very low. If the boundary between "good" and "bad" is selected in such a way that it is located in a sharply ascending portion of a normal distribution, in order to classify as "good" only those products that are definitely good, it will also be necessary, because of measurement inaccuracy, to classify some products as "bad" that are in fact good. This is economically disadvantageous.

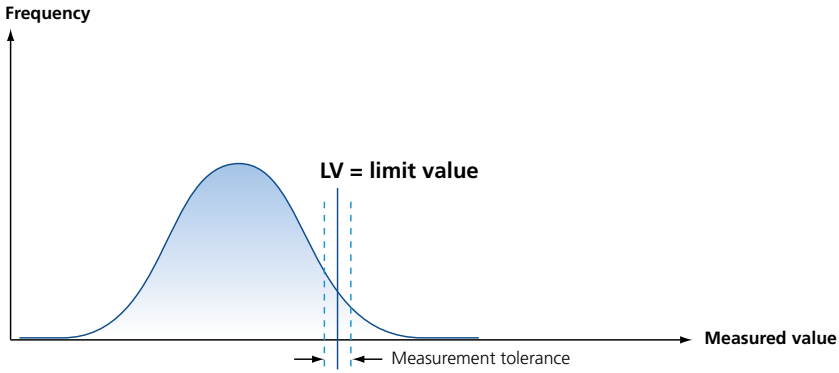


Fig. 8.4: Unsuitable characteristic in noise test

Therefore, in identifying defect patterns it is important to try to select those characteristics that ideally achieve a good or bad correlation of 100%. This means that the defect characteristics must not have normal distributions, but rather that they must be present with products that are classified as being bad (and are clearly measurable) and that they must be absent with products that are classified as being good. So we need to look for a bimodal distribution similar to that depicted for such a characteristic in Figure 8.5. The measurement

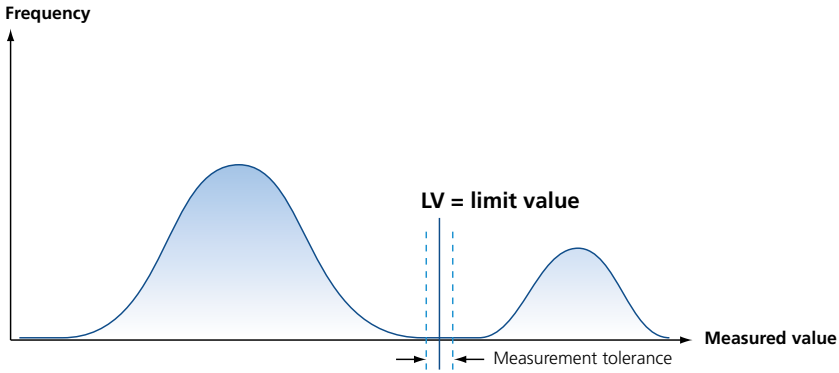


Fig. 8.5: Suitable choice of a characteristic to be tested

inaccuracy that is present in any test no longer has any effect when the characteristic is distributed in such a way. If such a bimodal distribution with complete separation of the characteristic to be tested cannot be achieved (for example, if the values of the individual distributions overlap, as shown in Figure 8.6), then a more suitable characteristic to be tested must be sought.

However, it is important to consider that a characteristic can appear to be unsuitable because of factors like individual part or batch variation, external factors (temperature, air pressure, etc.) as well as external vibrations that act on the part. If one is aware of these interfering variables, it is possible to make corrections and use a characteristic that initially appears to be unsuitable. This correlates with the reproducibility of the measurement. A characteristic should

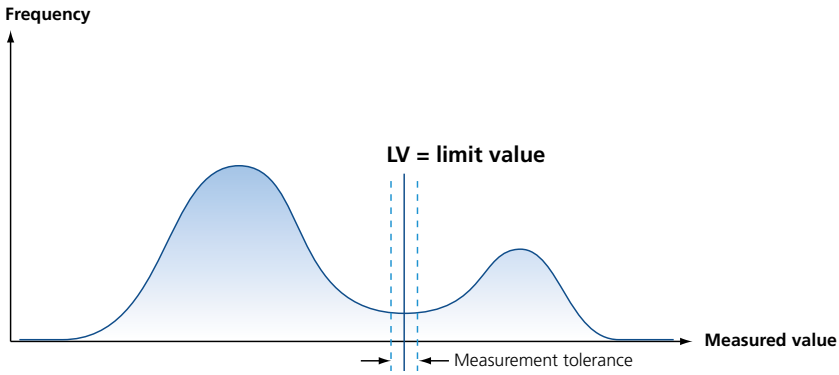


Fig. 8.6: Unsuitable choice of a characteristic to be tested

not be completely ruled out and a different characteristic or different solution sought until such efforts prove unsuccessful.

Depending on the nature of the noise problem, various characteristics may be suitable for achieving high good or bad correlation to varying degrees. It is therefore important in each individual case to verify that the best test results can be achieved. Table 8.1 should be used to guide one in identifying those characteristics that are promising for given auditory impressions.

When noise characteristics are evaluated, it is also possible to use one's own criteria that have been developed for the specific case. Even though it may not make any sense to do so in physical terms, it is sometimes helpful to generate a perceived loudness value of, say, a structure-borne noise signal. Psychoacoustic metrics sometimes can also function as helpful characteristics.

Table 8.1: Noise types and analysis methods

No.	Description of noise pattern							Analysis method															Description	
	Duration of noise					Modulations	Excited frequency range	Acoustic metrics															Description of feature	"Term" in local dialect
	Brief, one time	Brief, stochastic	Continuous, impactive	Continuous, modulating	Continuous, uniform	Mf		Sound pressure level	Sound pressure level over time	Perceived loudness	Articulation index	Degree of fluctuation	Modulation spectrum	Kurtosis/crest factor	Roughness	FFT	FFT over time	High-resolution spectrum Wavelet	Sharpness	1-third-octave spectrum				
1	●								○						●		●	●					FFT is only useful for a natural frequency analysis of a pulse, for example.	
2		●							○		○	○		○	○		●	●	●				Stochastic noises that occur for a brief time are often caused when a lubricating film is stripping off (in plain bearings, for example).	"Squeaking"
3				●		- 20 Hz			○		○	●	●				●						Low-frequency modulations can be caused by unroundness in gear drive elements, but also by beats that occur when two frequency components are close to each other.	"Droning"
4				●		20 Hz - 70 Hz			○		○		●		●		●							"Wowing"
5			●			- 15 Hz	- 5 kHz					○	●	●	○		●	●						"Ping"
6			●			15 Hz - 40 Hz	2 kHz - 6 kHz						●	●	○		●	●						"Rattle"
7			●			- 15 Hz	3 kHz - 12 kHz						●	●	○		●	●						"Chatter"
8			●			- 15 Hz	10 kHz -						●	●	○		●	●						"Clacking"
9					●			●	○	●	○					●		●	●				Severe excitation of a natural frequency or harmonic	"Squealing"
10					●		6 kHz - 20 kHz	●	○	●	○					●	○	●	●					"Whistling"
11				●		- 15 Hz	20 Hz - 300 Hz		●			○	●				○	●		●				"Whooshing"
12				●		- 20 Hz	1 kHz - 20 kHz		●			●	○			○	●	●						"Whining"

● Possible ○ Possible to some extent

8.7 Neuronal networks

Neuronal networks form the structure and informational architecture of the brain and nervous system in living creatures. Within a neuronal network, various inputs are combined to produce outputs in a manner similar to that which occurs in electronic circuitry. A neuronal network differs from a classic electronic circuit in that, because it is used more frequently, it can determine an output variable faster, or fewer input variables can generate the same output variable. Therefore, the system is **able to learn**.

This ability to learn can be put to use in recognizing and evaluating complex patterns without the need to explicitly know the rules for such an evaluation. Thus, neuronal systems do not follow the rule-based laws of artificial intelligence, but rather they follow the rules of what might be called "natural" intelligence. Of course, there are rules, laws, and logical behavior in neuronal systems, but they are derived from experience obtained through training and implicit learning.

Neuronal networks are also used in the acoustic test stands. Samples that have been judged to be "good" and "bad" are tested in this apparatus and the evaluation is communicated to the system. We refer to this as "teaching" the test stand. Training the neuronal system correctly is critical to successful learning and, therefore, corrective evaluation of devices under test in actual standard production testing. This requires a large number of teaching specimens, which must also include an adequate number of borderline samples. The biggest problem in using neuronal systems involves the stochastic evaluation of samples or devices under test that do not conform to the "learned" models.

8.8 Implementing noise test stands

When a noise test stand is being designed and built, it is a good idea to proceed one step at a time. The necessary steps are:

- Designing the mechanical system for holding the device under test and obtaining a measurement signal

- Construction of an output device to listen to the measured signal
- Providing the ability to listen to reference signals to assist in subjective testing
- Integrating an evaluation system in order to achieve objective testing

The first step is the toughest: designing a mechanical system for holding the device under test in a precisely defined manner and acquiring the measurement signal. The second step should not be undertaken until we have verified that this mechanical mounting system delivers reproducibility and sufficient accuracy. In designing the system for holding the device under test, it is important to make sure that the device is held in such a way that no feedback is generated – in purely conceptual terms, it is as if we were hanging the device from rubber bands. Electrical contacting can be a difficult problem, as can applying a torque load. In addition, the wear behavior of the fixture and the available cycle time must be considered.

One approach might be to implement a noise test as a modular system. The four elements of such a modular system are:

- Signal acquisition (basic module 1): A fixture that can hold the device under test in a defined manner and that can also simultaneously acquire a signal by means of a microphone, piezo sensor, or laser vibrometer, and can preamplify this signal.
- Output unit (basic module 2): The signal is amplified and provided to the tester over a loudspeaker. In order to protect the tester's hearing, a professional noise level limiter (occupational health and safety approval!) is used.
- CD reference signals (option module 1): Signals from limit samples are saved to a CD and are available to the tester at the touch of a button.
- Objective testing (option module 2): The signal can also be subjected to an objective evaluation. This can augment the subjective evaluation or replace it.

The real challenge in noise testing is to implement a suitable system for recording signals (basic module 1) and to identify a suitable characteristic to be tested that correlates with the motor that is to be tested when the motor is

in the installed condition. On the other hand, there are numerous possibilities and methods that can often be used to evaluate signals without difficulties, provided that the first step was done correctly. Figure 8.7 shows the principles of the modular noise test system.

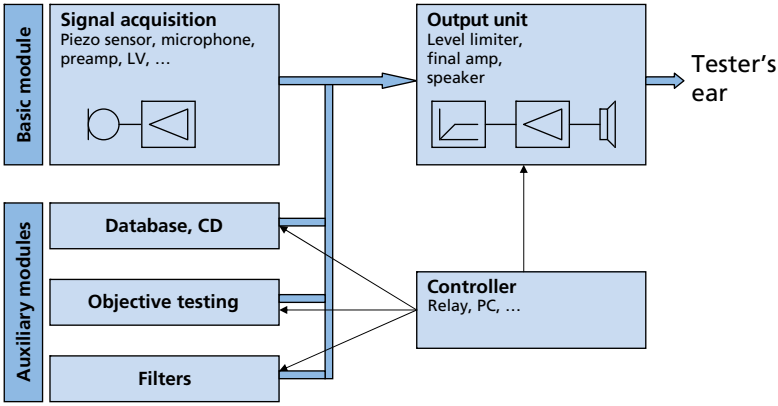


Fig. 8.7: Modular system for noise testing

To summarize:

- Noise test technology differs fundamentally from the technology used to measure physical variables.
- Vibration and noise test stands must be specifically designed for each individual case; there is no such thing as a "multipurpose test instrument" that actually works in the real world.
- The procedure can only be standardized through the development of a noise test procedure or test stand (using a modular system, for example).
- Because of the subjective nature of auditory impressions and sensation, there is no such thing as an objective noise measuring method.
- In order to arrive at an effective noise measuring procedure, it is necessary to determine how well a measured variable correlates with the noise in the final product.

9 Examples of actual systems

The implementation examples stated below provide a cross section of the methods, measurement systems, and analyses that have been introduced in this book. The possibilities offered by airborne and structure-borne noise measuring technology are readily apparent, as is the effort and expense that is required to obtain a capable measuring system and a meaningful analysis.

9.1 Analysis of a drive unit

This example involves a drive unit for a portable diagnostic device, which is referred to for short as a motor module with gear drive. The motor module consists of a mounting plate, a brush-type motor, and a 5-stage plastic gear drive (Fig. 9.1). The motor module is operated at speeds between 6000 and 10,000 rpm. The gear drive produces a reduction ratio of 450.91:1; the coil is of a three-part design. The precise gear calculation is shown in Table 9.1. Using the calculated rotational and engagement frequencies, it is possible later to link the frequency components in the spectrum of the measured airborne noise and structure-borne vibrations to specific individual components.

To make it easier to interpret the figure, the components of the individual gear drive stages and the columns in Table 9.1 containing the data for the respective stage are shown in the same colors.

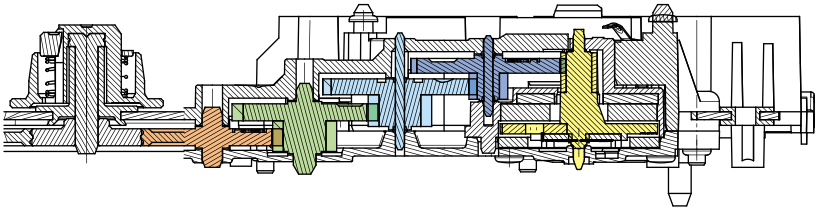


Fig. 9.1: Motor module (gear drive)

	Stage 1:	Stage 2:	Stage 3:	Stage 4:	Stage 5:
Speed in rpm:	6000	1125.0	204.5	72.6	27.5
Rotational frequency in s^{-1}:	100.0	18.8	3.4	1.2	0.5
Number of teeth, z1:	12	8	11	11	29
Number of teeth, z2:	64	44	31	29	60
Shaft output speed in s^{-1}:	18.750	3.409	1.210	0.459	0.222
Output order:	0.1875	0.0341	0.0121	0.0046	0.0022
Tooth meshing frequency in s^{-1}:	1200.0	150.0	37.5	13.3	13.3
Tooth meshing order:	12.00	1.50	0.38	0.13	0.13
LCM:	481.0	777.0	481.0	546.0	624.0
Tooth engagement repetition freq. in s^{-1}:	2.5	0.2	0.1	0.0	0.0
Gear ratio:	5.3	5.5	2.8	2.6	2.1
Overall ratio:	450.91				

Table 9.1: Gear drive data motor module

Figure 9.2 shows the frequency spectrum of the motor module at a speed of 6000 rpm. The spectrum is determined over a measurement period of 10 seconds. It is based on an airborne noise measurement using a dummy head whose two channels are each averaged. One can see that the greatest sound power is radiated in the range between 2 and 5 kHz. The additional modulation spectrum (Fig. 9.3) shows that precisely this frequency range is modulated at 100 Hz. This corresponds to the motor rpm, so that we can conclude that the cause of the high sound radiation can be sought somewhere in the area of the motor. For example, the cause may be the bearing system, the commutator, or rotor imbalance.

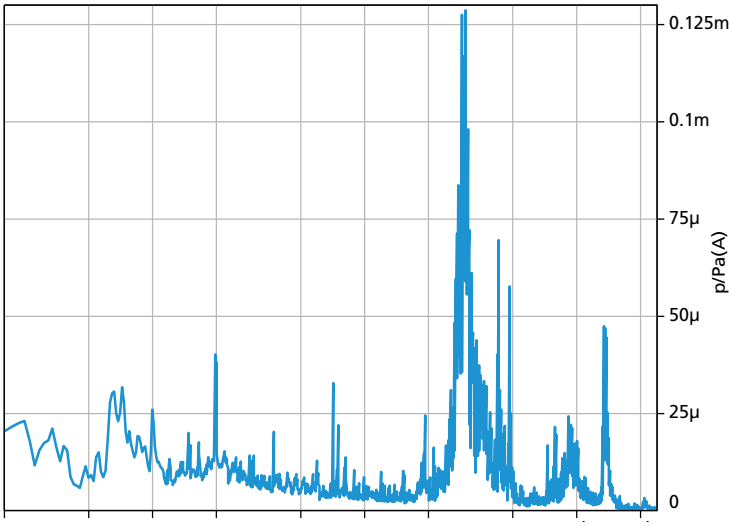


Fig. 9.2: FFT airborne noise measurement at a motor speed of 6000 rpm

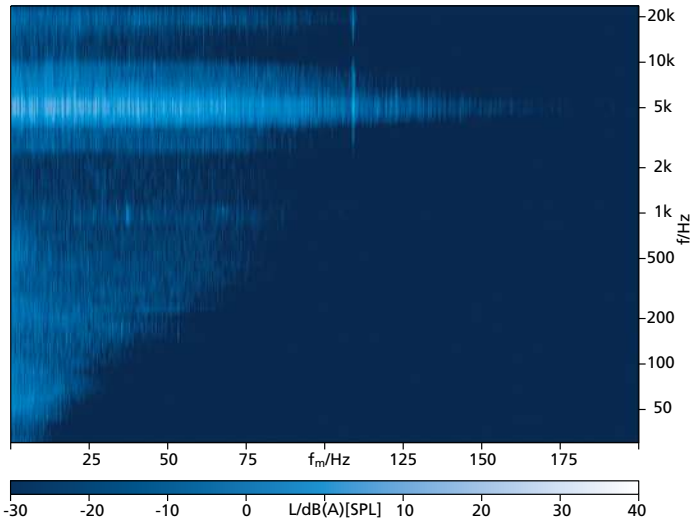


Fig. 9.3: Modulation spectrum from airborne noise measurement at a motor speed of 6000 rpm

In parallel, the structure-borne noise is recorded with the aid of a laser vibrometer. Figure 9.4 shows the spectrum of these oscillations. The spectrum has a sharp peak at 305 Hz. This corresponds precisely to the third harmonic of the motor rotational frequency. Because the fact that the coil is designed as three parts, we can conclude that the coil has a significant impact on the noise produced by the motor.

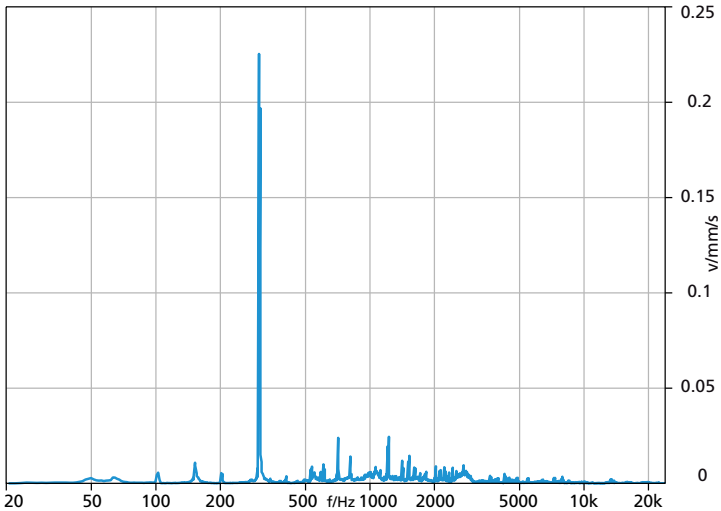


Fig. 9.4: FFT structure-borne noise measurement at a motor speed of 6000 rpm

The diagram of the modulation spectrum (Fig. 9.5) shows the pronounced amplitudes at the motor rotational frequency as well as noticeably weaker components at about 19, 38, 57 Hz, …. These components correspond to the rotational frequency of the first intermediate drive stage (see Table 9.1) as well as its harmonics. From this we can conclude that the first drive stage affects the noise generated by the motor.

If we take a closer look, we can recognize the components of the other drive stages in the spectrum. However, these amplitudes are so low that they play a lesser role in influencing noise.

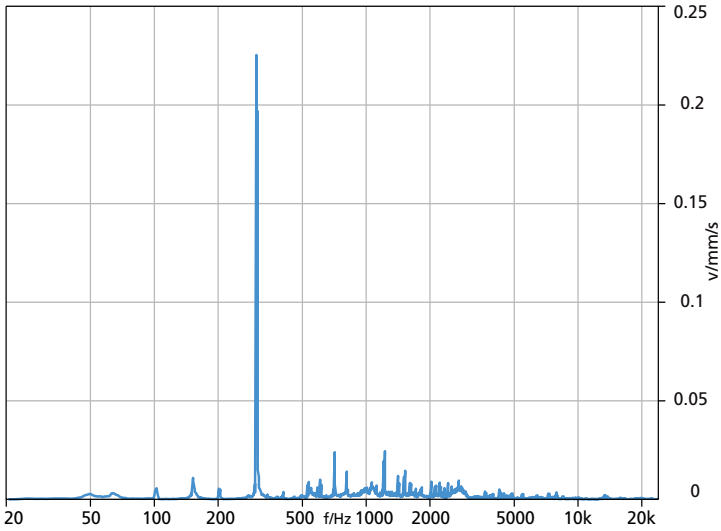


Fig. 9.5: Modulation spectrum from structure-borne noise at a motor speed of 6000 rpm

9.2 Evaluating imbalance

Four measuring points (MP1 to MP4) are needed to correctly evaluate and interpret imbalance in a motor. Of these four points, at least two must be measured in sync (MP1 + MP2 and MP3 + MP4) so that the phase relationships that exist between them can be evaluated. Two radial measuring points (MP1, MP2) and two axial measuring points (MP3, MP4) must be defined in order to interpret the data. The position of measuring points MP2 and MP3, as shown in Figure 9.6, must be taken into account in the phase relationship.

In addition to these oscillation levels, it is necessary to record a speed signal with a zero pulse (reference position for the phase relationship). The following defects can be distinguished using a setup of this type:

- Static imbalance
- Torque imbalance

Fig. 9.6: Measuring points for imbalance

- Dynamic imbalance
- Position of the imbalance (relative to the zero pulse)
- Bent shafts

The following analyses are required in order to interpret the measurement data correctly:

- Amplitude of the rotational frequency of MP1
- Amplitude of the rotational frequency of MP2
- Phase relationship of the rotational frequency of MP1 relative to the zero pulse
- Phase relationship of the rotational frequency of MP2 relative to the zero pulse
- Phase relationship of the rotational frequency of MP3 relative to MP4

The following analyses show the results for a typical motor:

The amplitude distribution of the two measuring points can be represented and interpreted nicely using a frequency analysis (Fig. 9.7). Given that the amplitudes of the two radial components are not equal, the problem may be a dynamic imbalance or a dynamic imbalance combined with a bent shaft. The representation of the phase relationship between the axial components (Fig. 9.8) helps us to gain a clear understanding of what is occurring.

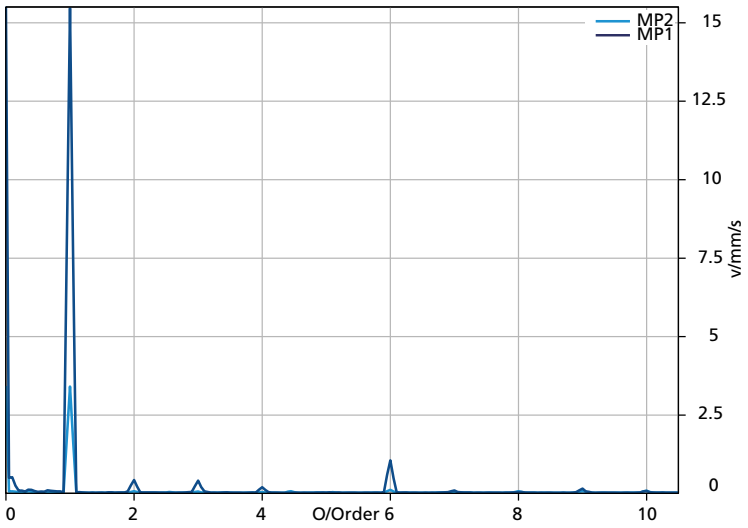


Fig. 9.7: MP1 and MP2 frequency spectrum

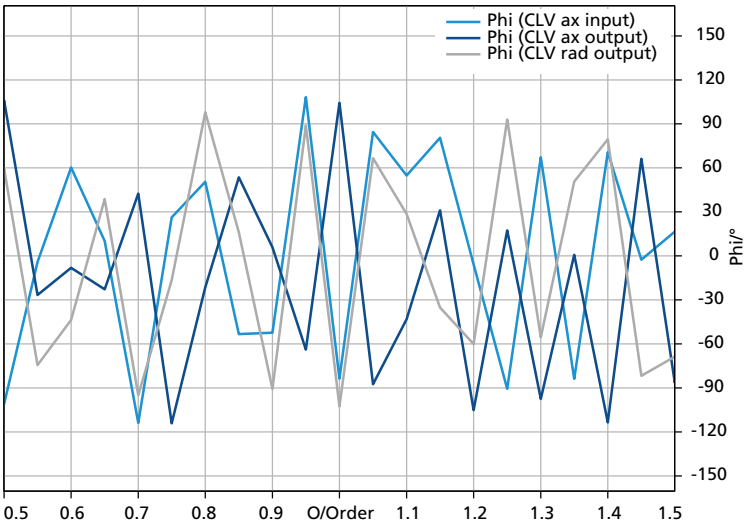


Fig. 9.8: MP3 and MP4 phase relationship

In order to be able to analyze the defect "bent shaft," we must first determine which oscillation components and which corresponding phase relationship are generated by a bent shaft. Figure 9.9 clearly shows that the oscillation components of the first order are in phase-opposition at measuring

Fig. 9.9: Response of axial measuring points to a bent shaft

points 3 and 4. However, if one takes the measurement direction at the two points into account, a phase angle of 0° would have to be obtained with a bent shaft. However, the fact that the phase angle between the two measured axial components is 180° eliminates the possibility of a bent shaft (see Fig. 9.8).

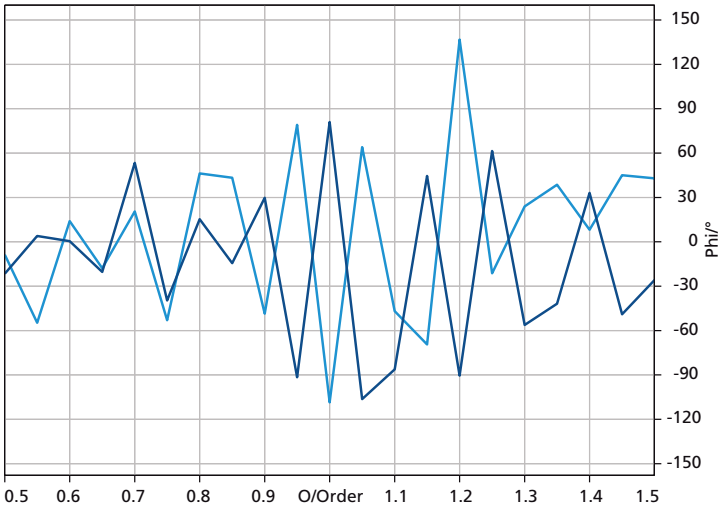


Fig. 9.10: MP3 and MP4 phase relationship relative to zero pulse

Furthermore, it was possible to determine the position of the out-of-balance masses by representing the phase relationship between the rotational frequency of the two measuring points 1 and 2 relative to the zero pulse (Fig. 9.10). As a result of this investigation, it is clear that the motor has a dynamic imbalance with a phase angle of 75° (MP1) and –110° (MP2). Other defects, such as a bent shaft, can be ruled out.

In order to interpret the amplitudes further, measurements with defined balancing masses are necessary. The balancing mass that is needed can be calculated precisely from them.

9.3 Testing a gear drive by checking the shaft output speed curve

This chapter describes an approach that avoids the need for an elaborate vibrational analysis using sensors. With gear drive combinations, it is definitely possible to use synchronization measurements on the output side to determine the quality of the gearing and therefore reach conclusions as to the vibrational characteristics of the gear drive. The theoretical basis for this approach

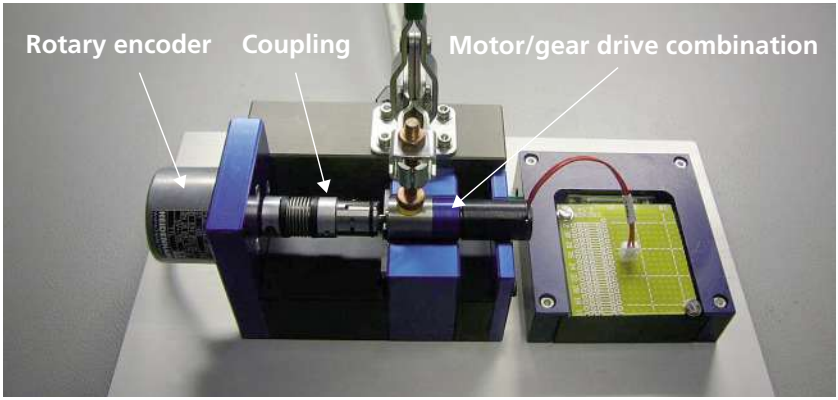


Fig. 9.11: Test setup for constant velocity test

is given in section 2.2.4 (see p. 29 f.). The torque fluctuations described there are also reflected by output speed variations. To record these variations it is merely necessary to analyze and evaluate the shaft output speed, which is recorded with a pulse sensor, for example, using the same signal analysis methods. Figure 9.11 provides an example of how a setup for a constant velocity test might look.

Figure 9.12 shows how signals can be measured using this recording method. One can clearly see the problem area in the gear drive, which occurs at 0.011-second intervals. The gear stage that is involved can be identified with the aid of FFT, or by means of a simple calculation taking the motor speed into account.

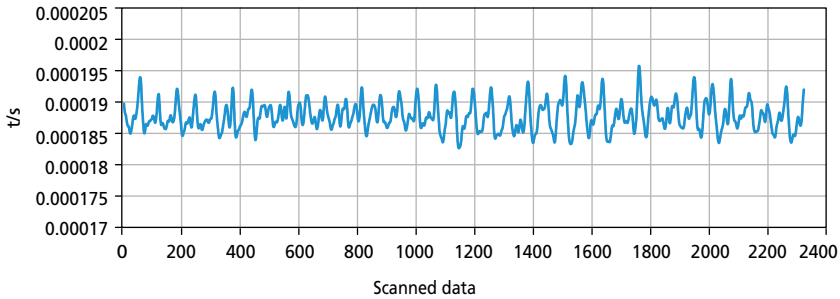
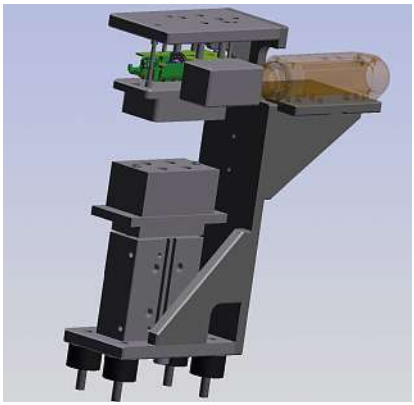


Fig. 9.12: Rpm variations at outputs of motor / gear drive unit

9.4 Noise test stand for checking gear noise in large-scale production

The next section provides an example that illustrates the use of the procedures described in sections 6–8 for standard production testing. The case: The objective was to introduce noise testing in actual production because damage on intermediate drives was repeatedly causing an unacceptable noise pattern ("chatter"). The damage was only apparent in the form of noise, and other technical characteristics such as rpm, speed



consistency, motor current were within specifications. Because of the short cycle time (4.5 seconds) and the corresponding high production rates, it was not possible, for example, to inspect the individual parts visually. Therefore, a vibration test had to be performed on the finished product.

The product consists of two motor/gear drive units that are controlled separately from each other. At

Fig. 9.13: Fully automatic test stations

their working point, the two motors are operated at a rotational frequency of 125 Hz. The rotational frequency of the critical intermediate drive is 34 Hz for the first unit and 32 Hz for the second unit. This means that to implement the test, one or more characteristics must be found that allow this damage to be classified at the repetition frequency of 34 Hz (32 Hz).

Since the production line is fully automatic (cycle time of 4.5 seconds), automatic 100% testing is also required. The key characteristics of the test apparatus were mechanical isolation of the test cell from the rest of the apparatus, isolation of the test locations (Fig. 9.13) from the test cell, and reproducible mounting of the device under test.

The transfer system in which the devices under test are transported in workpiece holders provides the connection between the cells. In order to isolate the test cells, all connections to this transfer system were removed. This means that while the transfer system is routed through the cell, the locating of the belt is accomplished using mounts located outside of the test cell.

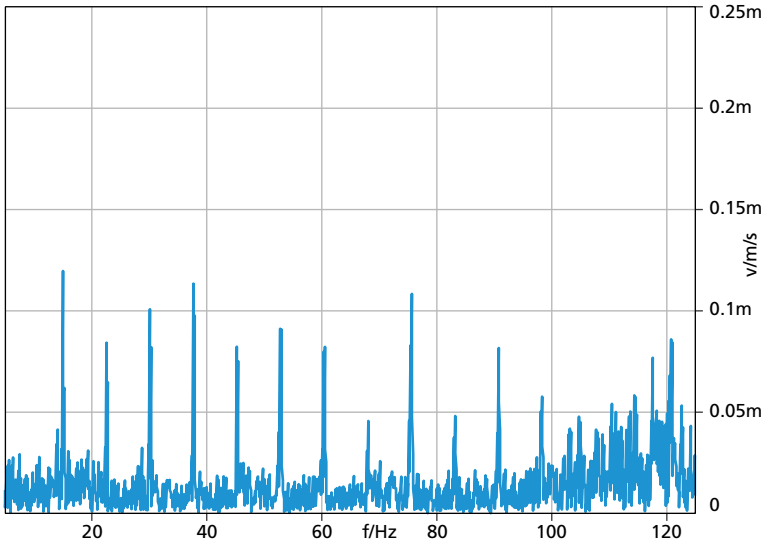


Fig. 9.14: Envelope spectra of unacceptable samples

In order to isolate the device under test from the test cell, the device under test together with all workpiece holders was lifted out of the rotary indexing table and placed in its measuring position. The measuring device and lifting mechanism were isolated from the test cell base plate by means of rubber elements.

On the basis of previously performed tests, the laser vibrometer was determined to be the optimal measuring system. Measurements using the laser vibrometer produced the best results because they were free of feedback and because of the advantages resulting from adaptation.

In order to arrive at the test limits, it was necessary to evaluate a sufficient number of limit samples subjectively. Here, it was important to have the customer participate in the evaluation. Joint participation was the only way to develop a test that would be accepted without reservation by both parties.

Using the limit samples, various characteristics were calculated, and their suitability for the classification was evaluated. As described in the previous sections, a characteristic cannot be selected unless it permits reproducible and repeatable classification. This evaluation made it possible to rule out the suitability of time-related characteristics, for example. Further analysis of the product and assessment of the envelope analysis produced optimal results with respect to the ability to separate individual classifications from each other. Finally, defined frequency bands within the envelope curve spectrum were used for the test. Figure 9.14 shows the envelope curve spectrum of a BAD device under test together with the selected frequency bands.

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