Environmental Leadership from Cradle to Grave

Diplomarbeit/Studienarbeit

Fundamentals and Applications of Active Noise Control

(Grundlagen und Anwendungen des
„Aktiven Schallschutzes“)

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1 Introduction

Noise is unwanted sound and is more and more perceived as an environmental pollutant. This shift in attitude is reflected in stricter legislation and has wide ranging economic consequences. Traditionally, noise was reduced by passive means like damping plates, sound absorbing materials, double-glazing windows, noise barriers etc. All these passive means are mostly suitable for high frequencies and have various disadvantages. In the last decade, active control of sound and vibration (at audio frequencies) has emerged as a viable technology to fill the low-frequency technology gap.

The challenge has been and still is to make active noise control a cost-effective solution to noise problems. This challenge has been far more difficult than many people expected. When a technology is more expensive than its benefits are generally worth, only a small market exists for the technology.

New technologies are usually expensive to start out. Early users of a technology pay a higher price for being first, with a competitive advantage as the benefit. Technology developers gain valuable experience working with early customers. This experience and know-how yields product improvements that provide more economical solutions for future customers. In this way, the technology spirals into wider and wider use, as the cost for implementation falls.

So, what happened to active noise control? Why don't we see active noise control technology in our daily lives? The answer lies in the fact that most consumers don’t pay more than 100 to 2000€ for noise reduction solutions. This cost range has been difficult to achieve with effective active solutions. The notable exceptions are noise canceling headphones and some dishwashers.

The result is that other applications of active noise control have been in industrial situations, where the benefit for active noise control is driven by hearing safety concerns. In some cases, improving speech communication has been important enough to use active noise control . Even in industrial markets, active noise control technology spirals have been rare.

So, where has active noise control been successfully applied? A partial list:

- Exhaust mufflers for internal combustion engines
- Headphones for aircraft use
- Ambulances
- Dishwashers
- Vibration isolators for engines
- Vibration isolation to protect electronics from vibration and shock
- Aircraft cabin noise reduction
- Furnaces and boiler

These products are successful because they have taken active noise control technology and reduced it to practice. Another common element is that with the exception of cabin noise reduction, the solution is 'simple'. The solution does not require an extensive number of microphones, sensors etc. In many cases the controller that produces anti-sound commands can be built with analog circuits, leading to lower costs and portability.

So where should we go in ANC technology development? In my opinion continued development of simple solutions is best done through education. Present commercial efforts are blocked in patent issues and return on investment questions. Through education, engineers can be trained in the use of active noise control. When these engineers are confronted with design problems, they will be equipped to find niche solutions using active noise control.

Researchers in this field should focus on two approaches. The first is the development of integrated active control solutions for complex noise problems (industrial noise) and the second are automobile audio systems, with integrated personal communications systems. Active noise control technology could improve our environment as well as driving enjoyment and safety.

2 Aims and scope

This study describes the fundamentals and development of active noise and vibration control as well as some new technical applications which took place in recent years. In Austria ANC is not widely known, though several applications could help solving problems caused by noise and/or vibration.

This work is not a practical scientific study, the aim is to show the worldwide state of the art in active noise technology, especially under the environmental point of view. Which opportunities gives us ANC? Where is it used today commercially? Where does this technology find its new applications? Where can I find more information about ANC? These are some of the questions answered in the following chapters.

3 Historical Review

3.1 Overview

Active Control is an old concept, but it is mainly in the past 10-15 years that developments in digital signal processing have made it a commercial reality. Some milestones in the development of active control are [2]:

Early 1930's:

Coanda (France) and Lueg (Germany) publish the first patents on active control.

Mid 1950's:

Olsen (USA) gives a wide range of proposed applications and the results of experiments. Conover (USA) applies active control to transformer noise. Active headsets developed.

1960's

Jessel (France) and Kido (Japan) develop the theoretical basis of active control and carry out experiments.

1970's:

Leventhall, Swinbanks and Chaplin, all in the UK, develop practical systems. Chaplin makes early use of microprocessors.

1980's:

Eriksson (USA) develops commercial digital-based systems. First DSP chips become available. Several companies are formed to provide active control systems. First commercial installations.

1990's:

Steady progress and growth of interest. Many new participants come to the the topic. Rapid development in availability of hardware. Considerable drop in cost. active control becomes a product for OEMs.

3.2 Development of ANC

The first patent on an ANC system (interference and absorption) got the German Lueg in 1936. He was the first who mixed acoustic waves with result of constructive or destructive interference, which cause intensification and weakening of the sound field. Lueg tried to manipulate the principle of superposition (see chapter 7.1) so that the destructive interference of sound waves could be used to cancel unwanted noise. He introduced the concept of active attenuation of sound by using artificially generated acoustic waves mixed with the unwanted sound so that the waves were in anti–phase and destructive interference resulted by design. In Figure 3.1 Lueg shows the problem of cancelling sound in a duct.

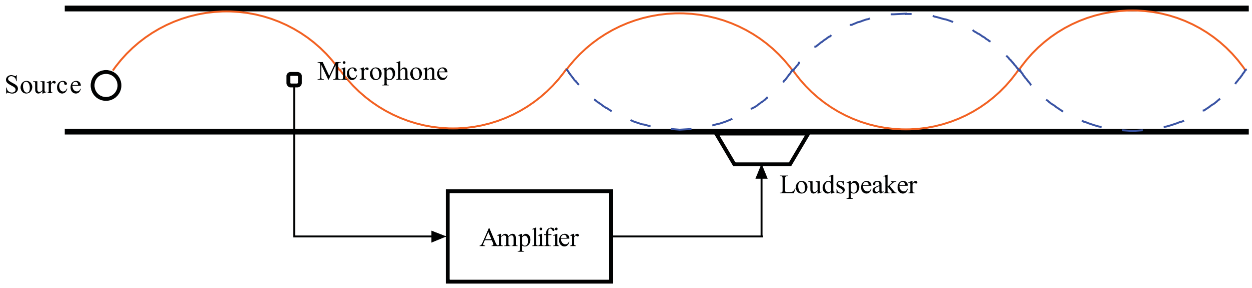


Fig. 3.1: Cancelling sound in a duct (Lueg)

Here acoustic noise propagates along the duct. The microphone detects the sound and converts it to an electrical signal. The electrical signal passes through an amplifier and then to the loudspeaker. Lueg shows a single frequency in the duct, the phase reversal of which is accomplished by considering the electronic system as a transmission line with a given time delay. The length of the line is adjusted to give the neccassary time delay that results in 180° phase shift relative to the sound wave detected at the microphone so that cancellation of that particular wave (frequency) results.

In Figure 3.1 Lueg illustrates the basic physical phenomenon which provides the possibility of ANC for this case[3]:

'An acoustic wave with a specific frequency has a relatively much lower speed than an electrical signal of the same frequency. This implies that while a sound wave is travelling from a point where it is to be attenuated, there is enough time available within the electronic circuit to process the signal and activate the control elements, to a greater or lesser degree, depending on the frequency, type of noise, and physical extent of the system'.

Lueg found out this basic phenomenon and is also shown in the structure of Figure 3.1. The relative distance of the detector from the primary source is less than that of the secondary source relative to the primary source. In applying the above mentioned physical phenomenon, Lueg shows that within the time interval required for the passage of an acoustic wave from the detection point (microphone) to the controler point (loudspeaker) sound can be detected by the microphone, passed through the amplifier (controller) and fed to the loudspeaker. In general, to achieve good noise cancellation has to have the required phase correction as well as amplitude correction characteristics. Unfortunately, the electronic technology of the 1930s was not sufficiently advanced to meet the requirements of ANC systems and practical results were not realized.

Interests in ANC disappeared for about twenty years until Olsen introduced his ‘electronic sound absorber’ (Olsen and May 1953). In several papers Olsen proposed localized sound reducers for occupants of vehicles and for machine operators, machinery noise control, noise reducing headsets and duct noise reduction. Olsons absorber is shown in Figure 3.2. The unwanted noise is detected by the microphone and passed through the amplifier. The amplifier drives the loudspeaker so that the soundpressure at the microphone location is reduced. This effectively creates a zone of cancellation in front of the absorber. Olson’s device shows considerable sophistication for the technology in the 1950s. The problem of the backwave is solved by using the absorbing material in the cabinet. The phase problem is minimized by locating the microphone (detector) close to the loudspeaker and ensuring linearity in the electronics. It was possible to control the phase with a good degree of accuracy over a broader frequency range. The basic feedback structure of the system with other system errors limits the utility of broadband noise cancellation and the physical extent of cancellation.

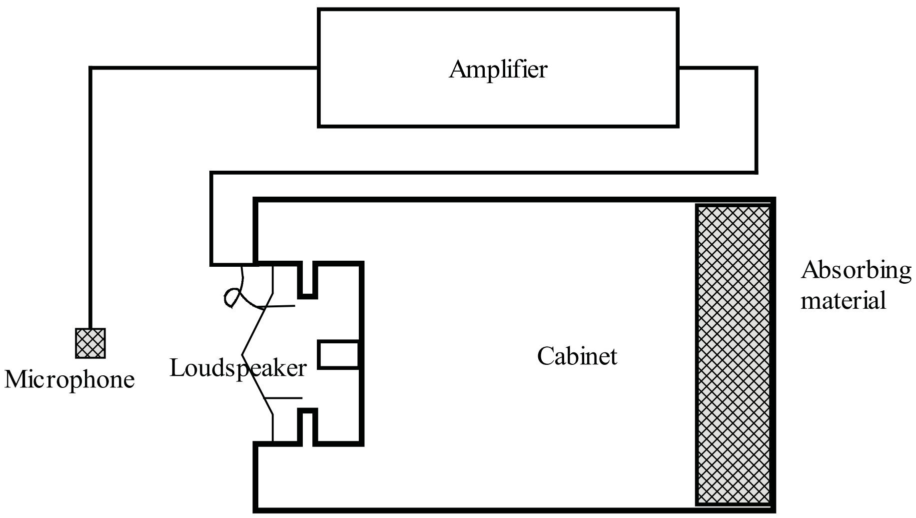


Fig. 3.2: Olsons absorber

Simshauser and Hawley proposed the development of an ‘active ear defender’ in the mid-1950s. A device to reduce ambient noise at the ear by using a headset to generate a sound pressure equal in magnitude and opposite in phase to the noise. In a noisy environment where the interference from the ambient noise especially at low frequencies, could be at a significant level, conventional ear defenders which use passive techniques of noise reduction are incapable of providing sufficient noise attenuation. Additional attenuation at low frequencies can be obtained by feeding an anti-phase signal to the ear defender. The device developed by Simshauser and Hawley is a two channel system consisting of two microphones mounted on either earphone of a conventional military headset. Each channel has an amplifier and a phase-shifting network. Noise is detected by the microphone, passed through the amplifier and phase-shifting network and applied to the earphone. Simshauser and Hawley tested the device in a pure tone field with proper amplitude and phase adjustments for minimum loudness. The result of their calculations, based on experimental measurements, indicated that such a device would provide an average reduction of 10dB more than is provided by the earcap alone, in the range of 100Hz to 1200Hz. This application is currently one of the more succesful one.

At about the same time Conover discussed the active control of transformer noise. He worked with a relatively large 15MVA tranformer installation. Unlike Simshauser and Hawley he did not confine his work to the laboratory but was rather succesful in that he developed his work in the field. Conover's scheme was to place loudspeakers near the tranformers surface and cancel the pressure radiation in the near field. He argued that sound radiates from the transformer due to vibrations of the core (caused by magnetostriction) and is transmitted through the core mountings and the fluid cooling medium to the tank. The tank couples these vibrations to the surrounding air. This produces a periodic spectrum with harmonics at even multiples of 100Hz. Furthermore an important characteristic of transformer noise is that the first few harmonics are usually the most important contributors to the sound level. A substantial reduction of one or more of the first three components (i.e. 100/200/300Hz) is usually sufficient to achieve a reduction in the noise.

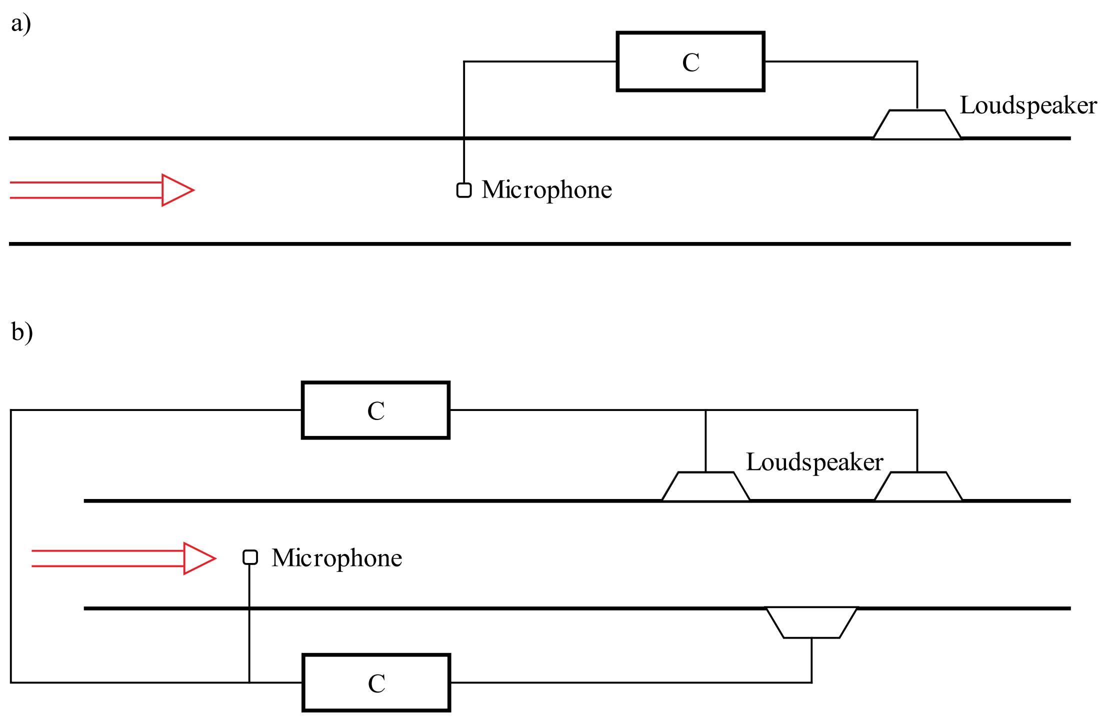
After adjusting each component in amplitude and phase the signals were recombined, amplified and fed to the speaker placed at the centre of one of the flat faces of the transformer. Measurements at 30m radial distance from the transformer showed a reduction of more than 6dB within an angular zone of about 11.5° on either side. Conover was the first to attempt the attenuation of transformer noise by active methods. Since then this has become a classic problem investigated directly or indirectly by Kido (1975), Hesselman (1978), Ross (1978), Jessel and Angevine (1980).

It is now clear that the noise of a physical source may be reduced everywhere in the far field, as long as the source is much smaller than the wavelength of the maximum frequency component of the radiated sound, and the cancelling source and radiating source are located less than one third or one quarter of the maximum wavelength apart (Hesselman 1978, Warnaka 1982). When the size of the source is much less than the wavelength of the radiated sound , the vibration of the surface is generally in phase. In this case the combined acoustic pressures of the source and the out-of-phase cancelling signal tend to cancel each other near the source. On the other hand, if the physical size of the source is large compared to the wavelength of the radiated sound the surface no longer movers in a simple uniform phase relationship. The movement is rather in a more complicated pattern since the phase varies over different regions of the source. As a consequence, the radiation pattern of the source is causing local reduction of the sound pressure without having much effect on the total radiated sound energy. Certain zones are created with reduced sound level whereas in other locations the sound is intensified.

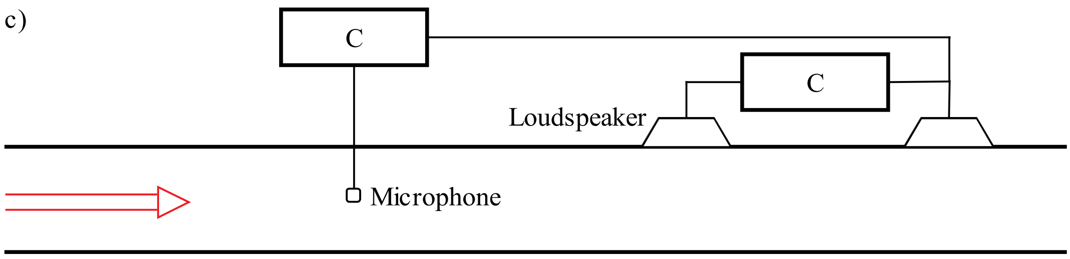
The accelerating interest in ANC began in the late 1960s with the publications of Jessel and his co-workers in France, and Kido in Japan. Jessel's work has been mainly concerned with duct noise. He and his co-workers have also made significant contributions to the theory of active attenuators and general consideration of the field. Jessel and Kido both realized early that the primary advantage of ANC systems is in their ability to attenuate low-frequency noise. This is an area of great interest because of the pervasiveness of low-frequency sources and the high cost and relative inefficiency of current passive hardwear in low-frequency applications. An advantage in the control of the one-dimensional propagation of duct noise lies in the fact that active noise silencers produce no back pressure.

Jessel and others discovered some of the problems associated with reducing duct noise. Longitudinal duct modes leading to acoustic feedback, caused by reflected components, tend to confuse the controllers as to the exact level of noise itself, since the detector microphone cannot distinguish between the noise and the reflected components. This leads to system instability and/or no noise reduction in some bands of frequency. In order to solve the longitudinal mode problem, so that the detector microphone detects the unwanted noise only, loudspeaker arrays can be used. Two or three loudspeakers used to form an acoustic dipole or an acoustic tripole can be phased to produce acoustic waves travelling in one direction.

The acoustic monopole system was originally considered by Lueg. In this system a standing wave is produced upstream which interferes with the detection of the unwanted noise. This makes the position of the microphone extremely sensitive to noise reduction. The acoustic tripole and acoustic dipole were developed by Jessel and his co-workers (Jessel and Mangiante 1972) and Swinbanks (1973). They attempt to provide a cancelling signal in the duct that propagates only in the downstream direction. The three types of active attenuators for duct noise are shown in Figure 3.3.



c)



Types of active attenuators for duct noise: a) monopole (Lueg), b) tripole (Jessel), c) dipole (Swinbanks)

Fig. 3.3: Types of active attenuators for duct noise: a) monopole (Lueg)
b) tripole (Jessel)
c) dipole (Swinbanks)

Multiple loudspeaker systems (dipole and tripole) have essentially been attempted to overcome the problems associated with frequency dependent controller design. These systems try to isolate the detector microphone from secondary source radiation and/or produce undirectional radiation, thereby avoiding problems of acoustic feedback and instability in the system. Furthermore, they have been unable to cope with reflected waves and have geometry related limitations. The control problem is also much more complex in such systems. The third limitation of these systems is the so called 'tuning effect' due to the physical spacings of the microphone and loudspeakers relative to each other. By altering these spacings the system is tuned to a different centre frequency, with no significant improvement in the bandwidth of attenuation. For these reasons, the current trend is to use the capabilities of modern electronic systems to implement complex controllers and hence give preference to controller complexity over geometrical complexity. With proper control the monopole is capable of cancelling noise over a broad frequency range. The control problem is relatively simple with a single source. The task of large attenuation over a broad band primarily lies in the design of the controler. Required is a proper ANC system design based on sound analytical conditions for general phase cancellation.

A general trend since the late 80s is now seen towards the application of digital signal processing techniques to the active control of noise. The current advance in digital technology makes the implementation of digital controlers feasible at low cost.

4 Fundamentals

4.1 Fundamentals of acoustics/sound

Sound is the sensation produced at the ear by very small pressure fluctuations present in the surrounding medium (which we will assume to be air). This sensation is produced in response to the pressure fluctuation induced vibration of the ear drum. The fluctuations in the surrounding air constitute a sound field. The pressure fluctuations themselves are usually referred to as sound pressure or acoustic pressure.

Noise is unwanted sound. What constitutes noise is therefore subjective. One type of sound which is generally agreed upon as being noise is that which comes from industrial processes and, in particular, sound from machines, punches, saws, fans etc. Why is it important to do something to limit the unpleasantness of noise? In more modern times, noise has been recognized as a serious health hazard, with significant financial compensation paid to those whose hearing has been impaired by their job or circumstances. This recognition, and the resultant penalties for noncompliance with accepted standards, has provided the impetus for an entire area of commereial activity and research: noise control.

Being able to communicate a problem is a prerequisite to fixing it. We must therefore be able to quantify sound levels, to assign them a number which describes "how big," before we can determine an appropriate course of action. Sound is most often described in terms of pressure. Commonly, it is the amplitude of the pressure fluctuation which is of interest. Pressure is not the only quantity which could be used to describe sound, but it is the most convenient because it is simple to measure, and it is a scalar quantity, and does not require a direction to be associated with it. Possible alternatives such as velocity are not scalar quantities, and so to use them as the basis for quantifying a sound field would complicate matters.

Pressure P is defined as the force F acting on a given area A (P = F/A). The standard (SI) unit of measurement for pressure is the Pascal, abbreviated as Pa. The quantity of 1 Pa is defined to be equal to a force of 1 Newton applied over an area of 1 square meter. The human ear can detect an incredible range of pressure fluctuations in the ambient environment, from approximately 0.00002 Pa (threshold of hearing for a healthy young person) to 100 Pa or more (pain begins around 60 Pa). These pressure fluctuations sit on top of the atmospheric pressure of approximately 101 kilopascals (101,000 Pa). The ear is also incredibly sensitive to relative changes in the acoustic pressure fluctuations. For example, a change in level from 0.0001 Pa to 0.0005 Pa will be as noticeable as a change from 1 Pa to 5 Pa.

The range and sensitivity of the ear led workers at the Bell Laboratories to define a new unit to quantify acoustic pressure. The unit was originally ealled the "Bel" in honor of their founder (Alexander Graham Bell), but was later modified to become the decibel (deci = 10, so 10 decibel are equal to 1 Bel). The decibel is commonly abbreviated as "dB." The decibel scale is a logarithmic scale. This means that the units increment proportional to the logarithm of the quantity of interest. Sound pressure, denoted Lp, in decibels is defined by the relationship

$$Lp = 10 \log_{10} p^2/p^2_{ref} = 20 \log_{10} p - 20 \log_{10} p_{ref} \quad (dB)$$

In this expression, p is the amplitude of the pressure fluctuation in Pascals, and p_{ref}, is a reference amplitude. This latter quantity is defined as the threshold of hearing, which is 20 micropascals (0,00002 Pa). Substituting this value into the expression, a more convenient form is obtained:

$$Lp = 20 \log_{10} p + 94 \text{ (dB)}$$

Note that when expressed in the decibel scale, a sound pressure increment of 20 dB is actually an increase in pressure amplitude by a factor of 10. For example, a pressure amplitude of 1 Pa is 20 dB greater than a pressure amplitude of 0.1 Pa.

It is interesting to consider what the above results imply for the world of advertising. A company might say that their new and improved product, for example, a dishwasher, is so quiet because sound pressure has been reduced by 50%. Well, a 50% reduction in sound pressure is a drop of 6 dB. Indeed this a significant reduction.

4.1.1 Sound Waves

Sound pressure fluctuations are most commonly generated by something which is vibrating. An exampie of this is a loudspeaker, where the sound is "generated" by the vibrating loudspeaker cone. Vibrations are not the only way to generate the required pressure fluctuations. Let us consider the case of a vibrating surface in a bit more detail, as it is the simplest to visualize.

As shown in Figure 4.1, if a vibrating surface placed in air is at equilibrium the air next to it will be also at its equilibrium. Suppose now the vibrating surface moves in an outward direction. Initially the air immediately in front of it is not moving, and so is compressed somewhat. The number of air molecules in front of the vibrating surface is the same, but the volume available to them has decreased owing to the presence of the outward-curved surface. The result: a pressure fluctuation is born. As the vibrating surface reverses direction and heads inward, it passes once again through equilibrium and then beyond. In the process, the air is "stretched" (rarefaction, as the volume availabie to the air molecules is increased). Another pressure fluctuation is born, this time being a small reduction in the value. If we were to measure the pressure in front of the vibrating surface, say with a microphone, we would see a series of equilibrium/compression equilibrium/rarefaction levels. This looks very much like a wave. In fact, this type of plot is referred to as a waveform.

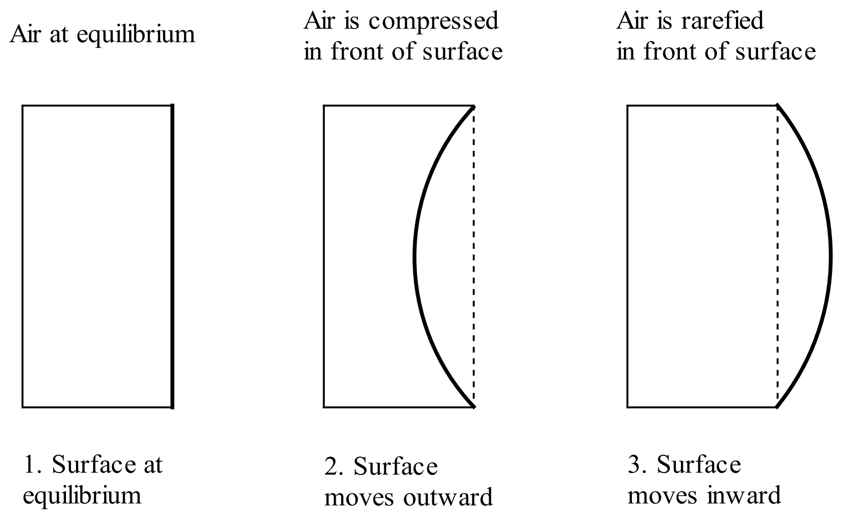


Fig. 4.1: Generation of pressure fluctuations by a vibrating surface.

4.1.2 Frequency Analysis

Frequency content is important for a whole host of reasons, ranging from how a human perceives a given sound pressure level, to what equipment is required to reproduce sound, and to what techniques can be used to control unwanted noise. It is therefore important to be able to qantify frequency, to assign it a number in the same way we did with sound pressure amplitude so that we an communicate and work with the notion of "low pitch/high pitch" in a scientific way. So what actually is frequency? Frequency is a measure of how many times an acoustic wave moves from compression to rarefaction and back again in a given period of time (the ωstandard time increment is 1 second).

This notion of measuring frequency by counting transitions between compression/rarefaction/compression works when the acoustie wave is "nice and smooth". It is rather unusual to encounter such a "pure" acoustic wave. Rather than talk about the "frequency" of the acoustic wave, we talk about the "frequency content." To expand upon this, it is necessary to discuss the concept of a sine wave and the work of Fourier.

4.1.3 Sine and Cosine Waves

The concept of a sine wave and the work of Fourier are absolute cornerstones of scientific and engineering thought, and indeed of technology as we know it. Practically any technical advancement, from the "simple" generation of electricity to the extraordinary feats of space probes, would lack description without them.

In classifying a sound pressure wave, we need to include mention of both its shape and frequency. For example, the sound pressure is a "sine wave at 100 Hz," or it is "tonal at 200 Hz." In the first of these, the general shape of the pressure (measurement) is a sine wave, and the number of cycles it undergoes per second is 100. In the second case, the general shape of the pressure measurement is again a sine wave (or better, sinusoidal), and the number of cycles it undergoes is 200. Two additional descriptors of a sine wave or waves which are of importance are amplitude and phase. The amplitude of a sine wave describes exactly "how large" it is. There are two measures of amplitude which are commonly cited: peak and root mean square, or RMS. Peak amplitude is simply the maximum (pressure) amplitude of the sine wave. It is the "top of the hump" in the plot. In a mathematical description of pressure field, it is the peak amplitude which is often used as the descriptor. The RMS amplitude provides a measure of the average sound pressure level over time. It is the measure which is used by noise control practitioners to assess the potential for hearing damage in a given environment. If the amplitude of the sound pressure measurement were simply averaged over time, the result would be zero. This is because for each "bit" of acoustic wave compression (positive amplitude) there is an equal, canceling "bit" of acoustic wave rarefaction (negative amplitude).

4.1.4 Fourier Analysis

The true value of the notion of a "sine wave" becomes apparent when considering the idea of a Fourier transform. Back in the nineteenth century, Joseph Fourier arrived at the conclusion that any steady-state waveform can be de-scribed as the sum of a number of sine waves with differing amplitudes and phases. The complex waveform shown in Figure 4.2 is actually the sum of independent sine and cosine waves. More complex waveforms will be the sum of even more waves. Even "random noise" can be described as the sum of sine and cosine waves.

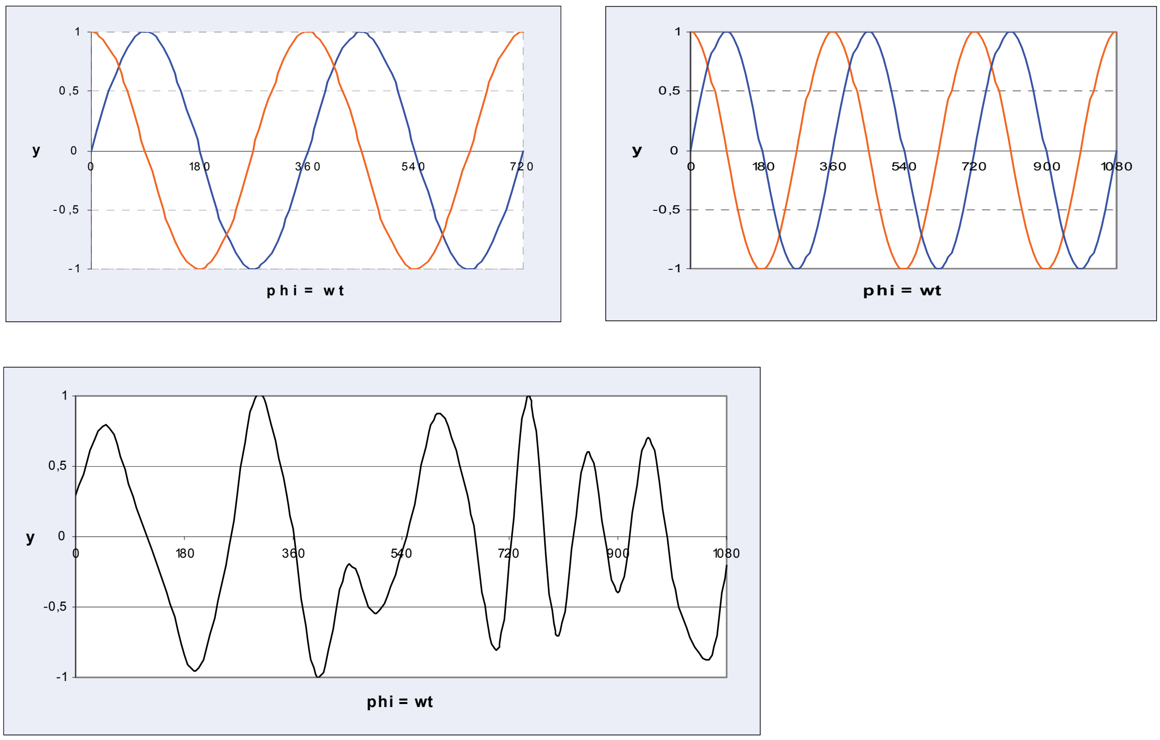


Fig. 4.2: The complex waveform will be the sum of independent sine and cosine waves.

The description of random noise is similar to that of white light, in that it is the sum of "all" sine waves in the frequeney range of interest. This notion that any waveform can be described as the sum of a group of sine waves is a powerful tool for studying and quantifying sound. It provides a mathematical way of explaining a variety of phenomena, ranging from why you can identify high- and low-pitched components in a complex sound field, to why a recording sounds "tinny" when played through poor quality loudspeakers, and to why active noise control works in some instances and not others. In fact, sound fields are most commonly described in terms of their spectrum, which is the variation in amplitude (and possibly phase) of the components of the waveform, ordered in terms of (sine and cosine wave) frequency. The analysis of a waveform in terms of its constituent sine waves is referred to as spectral analysis, frequency analysis, or Fourier analysis. The sine waves which make up a given waveform are referred to as frequency components of the signal. Fourier not only arrived at the amazing conclusion that all waveforms can be described by the sum of sine waves, but also developed a mathematical way of working out what the frequencies, amplitudes, and phases of the sine waves are. This technique is known as the Fourier Transform. There is also an opposite technique which is used to turn a group of sine waves into a waveform. This latter technique is called the Inverse Fourier Transform. While the Fourier transform has been a cornerstone technique in many areas of engineering and mathematical science since its inception, the "practical" use of spectral analysis has truly blossomed in the last 30 years as a result of the development of technique for fast calculation of the Fourier transform.

4.1.5 Harmonics

In the study of sound and vibration, and important concept is that of harmonics. Harmonics are frequency components which are integer multiples of some "fundamental" harmonic frequency. For example, if 100 Hz is the fundamental harmonic frequency, then 200 Hz is the second harmonic, 300 Hz is the third harmonic, 400 Hz is the fourth harmonic. If a waveform is periodic then the spectrum of the waveform will contain a series of harmonics. The simplest example is that of a sine wave, that contains a single harmonic (the fundamental). If the waveform is perfectly square and periodic, then it contains all odd-numbered harmonics (1, 3, 5, etc.). If the waveform is some other periodic shape, it will be constructed from some other combination of harinonics. Note that the number of times the pattem repeats itself every second will be the frequency of the fundamental harmonic.

The sources of a lange number of practical noise problems produce sound fields which are comprised of harmonics (that is, harmonic sound fields). Examples of these include the following.

1. Anything that Rotates

Consider the generation of a sound field by a fan. Every time a fan blade goes past a given point, a pressure pulse is created, resulting in sound generation. The base frequeney of the sound in Hertz is equal to the rotational speed of the fan (revolutions per second) multiplied by the number of blades. So if a fan rotates 60 times per second, and there are four blades, then there are 240 pressure pulses created each second. The fundamental frequeney is 240 Hz. While the sound pressure waves generated by the movement of the fan blades will be periodic, they will not in general be perfectly sinusoidal. Instead, they will be "distorted" sine waves, comprised of multiple harmonies (at 480 Hz, 720 Hz, etc.).

2. Many devices which use mains electricity

Mains electricity, the sort of electricity which comes from a wall socket, provides altemating electrical current. The electricity is delivered in the form of a wave, with a fixed frequency. Many appliances which plug into the wall "latch on" to the current frequency and, as a result, operate at this frequency. In turn, the appliances may generate sound fields which are periodic, with frequency components that are harmonics of the mains frequency. This is true for many low-cost electrical motors and fans. It is also true for a large number of nonrotating devices such as transformers.

Transformers vibrate in response to changes in their internal magnetic field. As this magnetic field is generated by the electricity entering the transformer, it is intuitively obvious that the vibrations must be related to the frequency of the electric current. They are, in fact, harmonically related. The strength of the magnetic field increases and decreases in re- sponse to the current "wave." It also changes direction, from "north" to "south" . To the metal components of the transformer, whether the field is north or south has no bearing upon the physical response. They only see "strong" and "weak." The result is that the fundamental frequency of vibration is twice the frequency of the current. We call the pattern of the resultant vibration rectified. Neither the vibration nor the resulting sound pressure field is perfectly sinusoidal (it is rectified). Therefore, it contains harmonics. Most people are familiar with the "buzz" caused by the higher frequency harmonics in the sound pressure field generated by a transformer and/or power supply in an appliance.

4.2 Fundamentals of vibration[1]

This chapter covers only the fundamental aspects of vibration theory, the principles have a much wider applications. This is because nearly all vibration phenomena that occur in practice follow these basic principles so that generally only a simple model is necassary to qualitively explain an event. For a fully quantitative simulation, much more detailed models are required, but even these often consist of an assembly of simple models. The size of a model is specified by the number of degrees of freedom it possesses, i.e. the number of independent co-ordinates required to uniquely define ist configuration at any time. This chapter only deals with systems having one degree of freedom, it can be considered as the basic building block for all sorts of vibrations.

Two forms of vibration are discussed here, namely transient and periodic. A vibration phenomenon is described as transient if it has a definite beginning and end. If the event repeats itself at regular intervals, it is termed periodic motion. A special form of periodic motion is where the displacement of a vibrating system varies sinusoidally with time. This is termed harmonic motion as is particularly important because all periodic motion can be decomposed into a series of sinusoids.

4.2.1 Simple freely vibrating spring-mass system

4.2.1.1 Without damping

The system considered consists of a spring with stiffness *k* fixed at one end and with a point mass *m* at the other end (Figure 4.3). If the mass is displaced downwards from ist equilibrium position and is the released, the spring force will accelerate the mass upwards.

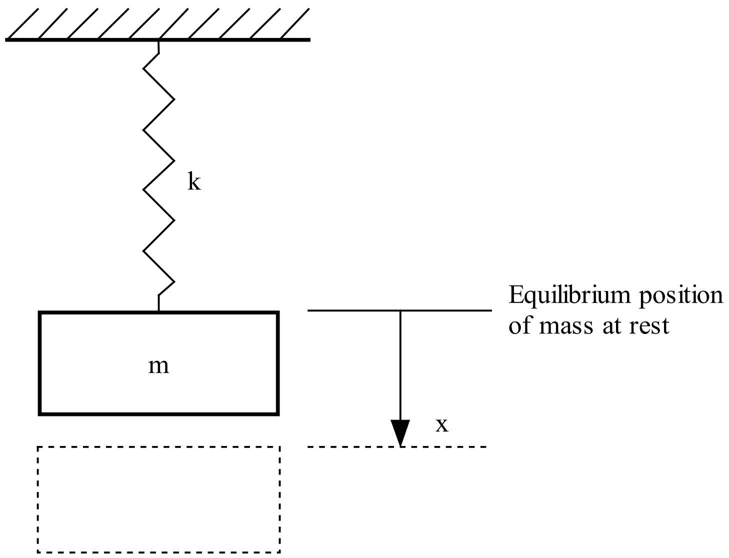


Fig. 4.3: Freely vibrating spring-mass system

At any instant of time following release, when the mass is a distance x from ist equilibrium position, the force acting on the mass is

$$-kx. \ [N/m]$$

k = spring stiffness (Force / unit extension) [N/m]

The sign is negative because the force acts in the opposite direction to the displacement. Applying Newton's Second Law of motion gives

$$mx = -kx$$

or

$$mx + kx = 0.$$

The most general solution of this differential equation is

$$x = A_1 \sin \omega_0 t + A_2 \cos \omega_0 t$$

where A_1, A_2 are constants.

$$\omega_0 = \sqrt{(k/m)}. \ [s^{-1}]$$

As a result the above analyses shows that, unless $B_1 = B_2 = 0$ (no vibration), the equation $\omega = \sqrt{(k/m)}$ is valid for all values of x. This is because the system is linear, which means that the force in the spring is directly proportional to ist extension, i.e. it obeys Hook's Law. In the following chapters we will always find that the natural frequencies and the solutions to equations of motions are independent of the amplitude of vibration and that this amplitude of vibration is proportional to the amplitude of the exciting force.

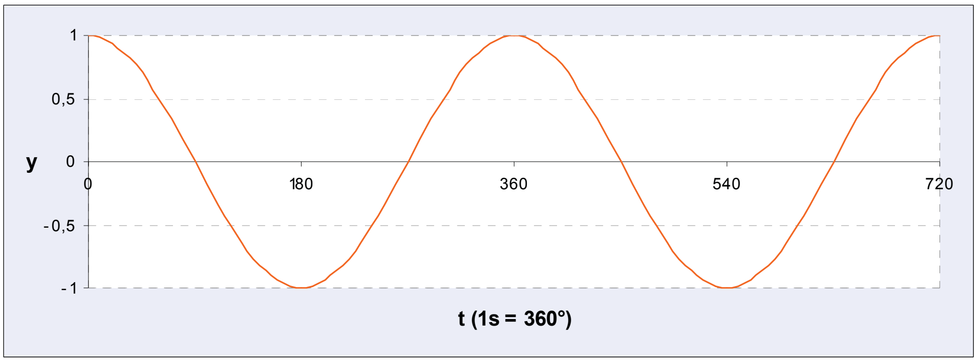


Fig. 4.4: The variation of x with time as determined by the equation $x = x_0\cos(\omega t)$

4.2.1.2 Viscous damping

If a viscous damper is placed in parallel with the spring, as shown in Figure 4.5, the differential equation of motion becomes

$$mx + cx + kx = 0$$

where c = viscous damping constant (force / unit velocity)

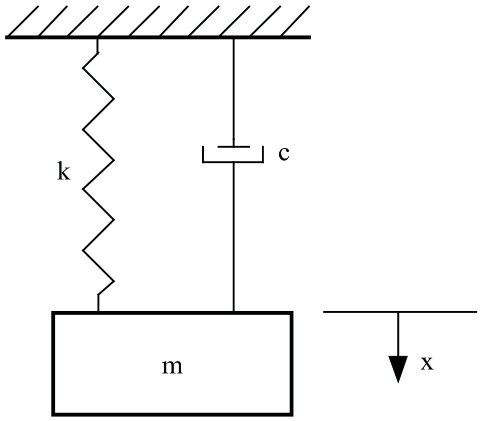


Fig. 4.5: Spring-mass system with viscous damping

Note that the damping force *cx*. Also opposes the motion of the mass and therefore has the same sign as the spring force *kx*. The most general solution for this equation is

$$x = B_1 e^{s1t} + B_2 e^{s2t}$$

where B_1 and B_2 are constants determined by the initial conditions. Each of these components of x are independent solutions. Therefore we can consider a typical solution as

$$x = Be^{st}$$

$$ms^2 + cs + k = 0.$$

Using the quadratic formula to solve for s we come to a solution with two basic forms. The first is where

$$(c/2m)^2 > k/m.$$

Secondly

$$k/m > (c/2m)^2.$$

When the former is the case, both s_1 and s_2 are real numbers and, on release, the mass will return exponentially to ist equilibrium position without performing any oscillation. The border line between oscillatory and non-oscillatory motion is when

$$k/m = (c/2m)^2$$

or

$$c = \text{critical damping} = c_c = 2\sqrt{(km)}.$$

Figure 4.6 shows the result of a damped sine wave. Strictly speaking the motion is no longer harmonic (in fact it is not even periodic but transient). Provided the damping is not too large, the motion can still be considered harmonic.

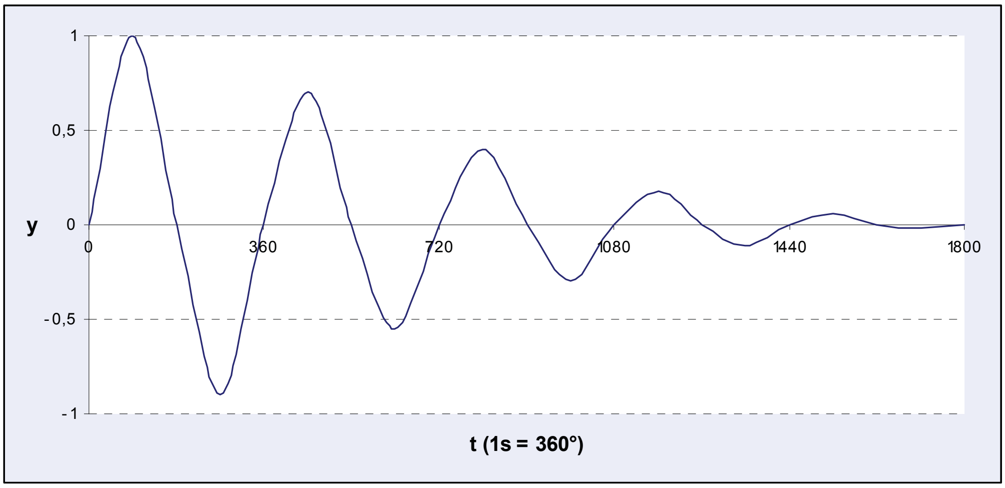


Fig. 4.6: Damped sine wave

4.3 Fundamentals of ANC

In the simplest form of ANC a control system drives a loudspeaker to produce a sound field which is an exact mirror-image the offending sound (the "disturbance"). The loudspeaker thus "cancels" the disturbance, and the net result is no sound at all. This phenomenon is called ‘destructive interference’ and is explained in chapter 3.3.1.2. In practice, of course, active control is somewhat more complicated.

The name differentiates "active control" from traditional "passive" methods for controlling unwanted sound and vibration. Passive noise control treatments include "insulation", silencers, vibration mounts, damping treatments, absorptive treatments such as ceiling tiles, and conventional mufflers like the ones used on today's automobiles. Passive techniques work best at middle and high frequencies, and are important to nearly all products in today's increasingly noise-sensitive world. But passive treatments can be bulky and heavy when used for low frequencies. The size and mass of passive treatments usually depend on the acoustic wavelength, making them thicker and more massive for lower frequencies. The light weight and small size of active systems can be a critically important benefit.

The four major parts of an active control system are:

- The **plant** is the physical system to be controlled; typical examples are a headphone and the air inside it, or air traveling through an air-conditioning duct.
- **Sensors** are the microphones, accelerometers, or other devices that sense the disturbance and monitor how well the control system is performing.
- **Actuators** are the devices that physically do the work of altering the plant response; usually they are electromechanical devices such as speakers or vibration generators.
- The **controller** is a signal processor (usually digital) that tells the actuators what to do; the controller bases its commands on sensor signals and, usually, on some knowledge of how the plant responds to the actuators.

There are two basic approaches for active noise control: active noise cancellation (ANC) and active structural-acoustic control (ASAC). In ANC, the actuators are acoustic sources (speakers) which produce an out-of-phase signal to "cancel" the disturbance. Most people think of ANC when they think of active noise control. On the other hand, if the noise is caused by the vibration of a flexible structure, then ASAC may be more appropriate than ANC. In ASAC, the actuators are vibration sources (shakers, piezo-ceramic patches, etc.) which can modify how the structure vibrates, thereby altering the way it radiates noise.

Active vibration control is a related technique that resembles active noise control. In either case, electromechanical actuators control the response of an elastic medium. In active noise control, the elastic medium is air or water through which sound waves are traveling. In active vibration control, the elastic medium is a flexible structure such a piece of vibrating machinery. The critical difference is that active vibration control seeks to reduce vibration without regard to acoustics. Although vibration and noise are closely related, reducing vibration does not necessarily reduce noise.

Active noise control usually occurs by one, or sometimes both, of two physical mechanisms: "destructive interference" and "impedance coupling". Here is how they work:

In some cases, an active control system can actually absorb acoustic energy from a system. Of course, the amount of energy absorbed by the system is usually tiny compared to mechanical losses or other losses in the system, but absorption is one possible mechanism for active systems.

Active noise control works best for sound fields that are spatially simple. The classic example is low-frequency sound waves traveling through a duct, an essentially one-dimensional problem. These systems are ideally suited for use in the frequency range below approximately 500Hz. The spatial character of a sound field depends on wavelength, and therefore on frequency. Active control works best when the wavelength is long compared to the dimensions of its surroundings, i.e. low frequencies. Fortunately, as mentioned above, passive methods tend to work best at high frequencies. Most active noise control systems combine passive and active techniques to cover a range of frequencies. For example, many active mufflers include a low-back-pressure "glass-pack" muffler for mid and high frequencies, with active control used only for the lowest frequencies.

Although higher frequency active control systems have been built, a number of technical difficulties, both structural/acoustic (for example, more complex vibration and radiated sound fields) and electronic (where higher sampling rates are required) limit their efficiency, so they are restricted to very special applications.

Controlling a spatially complicated sound field is beyond today's technology. It is somewhat easier to control noise in an enclosed space such as a vehicle cabin at low frequencies where the wavelength is similar to (or longer than) one or more of the cabin dimensions. Easier still is controlling low-frequency noise in a duct, where two dimensions of the enclosed space are small with respect to wavelength. The extreme case would be low-frequency noise in a small box, where the enclosed space appears small in all directions compared to the acoustic wavelength.

Often, reducing noise in specific localized regions has the unwanted side effect of amplifying noise elsewhere. The system reduces noise locally rather than globally. Generally, one obtains global reductions only for simple sound fields where the primary mechanism is impedance coupling. As the sound field becomes more complicated, more actuators are needed to obtain global reductions. As frequency increases, sound fields quickly become so complicated that tens or hundreds of actuators would be required for global control.

Directional cancellation, however, is possible even at fairly high frequencies if the actuators and control system can accurately match the phase of the disturbance.

Bandwidth is also important. Broadband noise, that is, noise that contains a wide range of frequencies, is significantly harder to control than narrowband (tonal or periodic) noise or a tone plus harmonics. For example, the broadband noise of wind flowing over an aircraft fuselage is much more difficult to control than the tonal noise caused by the propellers moving past the fuselage at constant rotational speed.

The many practical benefits of active control technology are not all obvious at first glance. The main payoff, of course, is low-frequency quieting that would be too expensive, inconvenient, impractical, or heavy by passive methods alone. For example, the lead-impregnated sheets used to reduce aircraft cabin propeller noise impose a severe weight penalty, but active control might perform as well with a much smaller weight penalty.

Other possible benefits reflect the wide range of problems on which active control can be applied. For instance, with conventional car mufflers the engine spends extra energy to push exhaust gasses through the restrictive muffler passages. On the other hand, an active control muffler can perform as well with less severe flow restrictions, thus improving performance and/or efficiency. Additional benefits include:

- increased material durability and fatigue life
- lower operating costs due to reduced facility time for installation and maintenance
- reduced operator fatigue and improved ergonomics

Of these, the potential for reduced maintenance and increased material fatigue life have received new emphasis in the last few years. In the long-term benefits may extend far beyond those mentioned above. The compact size and modularity of active systems can provide additional flexibility in product design, even to the point of a complete product redesign.

4.3.1 Structures of ANC systems[5]

4.3.1.1 Single channel ANC system

The basic structure of a single ANC sytem is shown in Fig 4.7. The noise propagates to the error microphone via an acoustic path. In addition the reference microphone picks up a version of the noise signal after it passes through acoustic path. The idea is to process this reference signal, using the controller, in such a way that when it emerges from the loudspeaker and arrives at the error microphone, it is the exact negative of the primary signal.

In theory, if the impulses responses of all the acoustic paths could be measured, then it should be possible to directly calculate the required controller parameters, and a fixed controller could then be used. This was tried by several researchers in the 1950s, and the results proved to be quite unsatisfactory. Reasonable attenuation could be achieved for narrowband noise, but for wideband noise the attenuation was poor. Therefore, most ANC systems make use of feedback of some kind (see chapter 4.3.3). This may be in the signal path with a fixed controller, or in the form of an adaptive controller in which the error signal is fed back to a control algorithm which modifies the controller, as in Figure 4.7. Of course, a combination involving signal feedback as well as adaptive control is also possible.

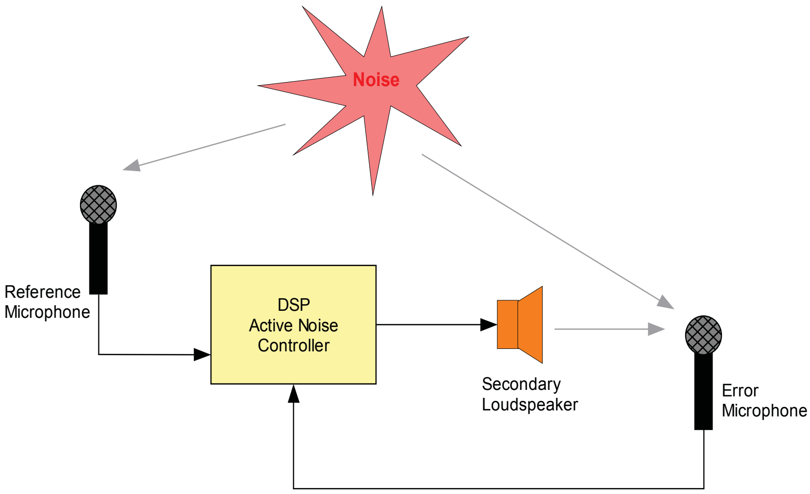


Fig. 4.7: Single channel ANC system

4.3.1.2 ANC with combined error/reference microphone

Figure 4.8 shows an ANC system in which only one microphone is used. This combiened error/reference microphone provides the error signal and the reference signal. Intuitively, this system can be thought of as taking the two microphone in Figure 4.7, and placing them so that they share the same physical location. Many ANC headsets use this type of architecture, with a fixed analoque controller. In this case two such systems are required, one for each ear.

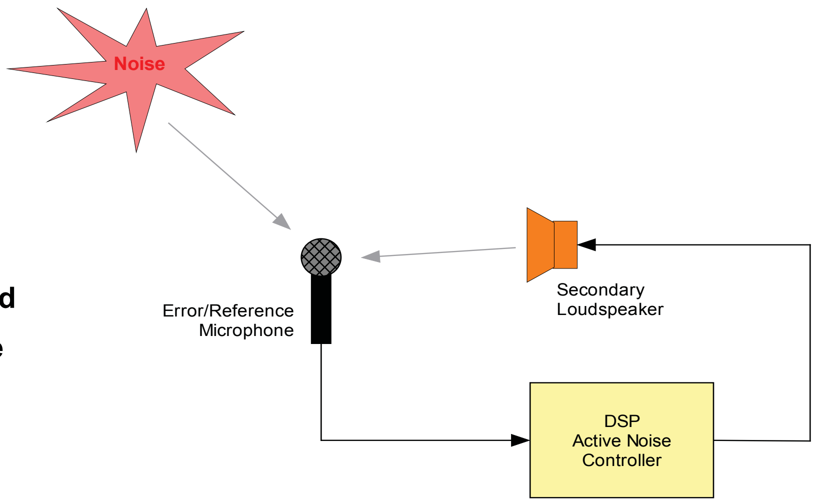


Fig. 4.8: ANC with combined error/reference microphone

4.3.1.3 Multichannel ANC

To create a larger zone of attenuation, several loudspeakers and error microphones are required, as shown in Figure 4.9. As can be seen, the number of secondary acoustic paths can be very large, and the computational complexity of the control algorithms tend to reflect this.

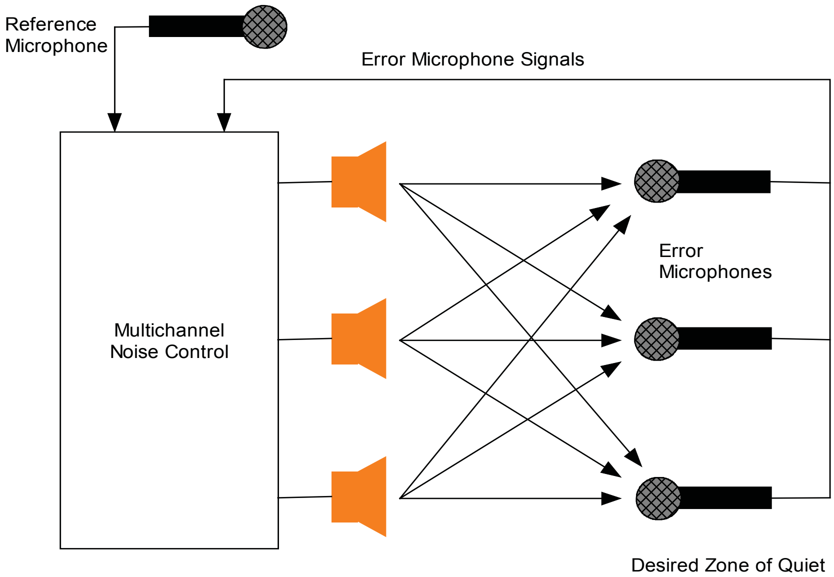


Fig. 4.9: Multichannel ANC

4.3.2 Interference

4.3.2.1 Constructive interference

Let's set up a situation: two speakers are situated at the exact same distance (3 meters) away from you; and each speaker is emitting the same sound. The wavelength of the sound is 1m. Finally, and most importantly, the speakers' diaphragms are vibrating synchronously (moving outward and inward together). Since the distance from the speakers to you is the same, the condensations of the wave coming from one speaker are always meeting the condensations from the other at the same time. As a result, the rarefactions are also always meeting rarefactions. One principle of sound is linear superposition (see chapter 7.1), which states that the combined pattern of the waves is the sum of the individual wave patterns. So, the pressure fluctuations where the two waves meet have twice the amplitude of the individual waves. An increase in amplitude results in a louder sound. When this situation occurs it is said to be "exactly in phase" and to exhibit "constructive interference", as shown in Figure 4.10.

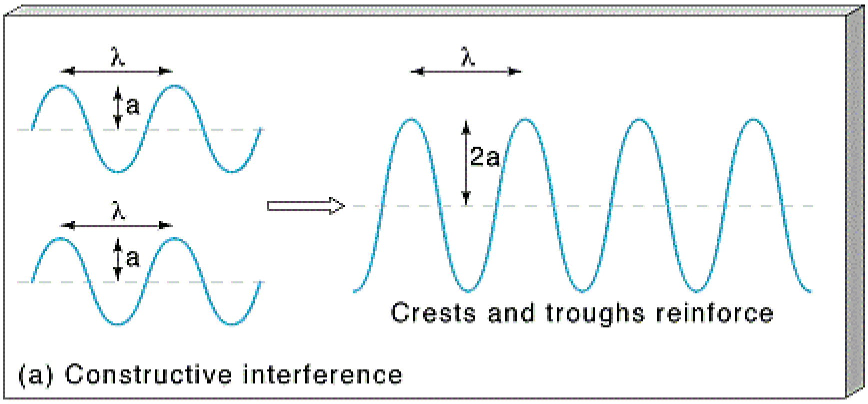


Fig. 4.10: Constructive Interference

4.3.2.2 Destructive interference

A sound wave is a moving series of compressions (high pressure) and rarefactions (low pressure). If the high-pressure part of one wave lines up with the low-pressure of another wave, the two waves interfere destructively and there is no more pressure fluctuation (no more sound). The matching must occur in both space and time. You can say that the control system creates an inverse or anti-noise field that cancels the disturbance sound field. The principle is called ‘destructive interference’, shown in Figure 4.11.

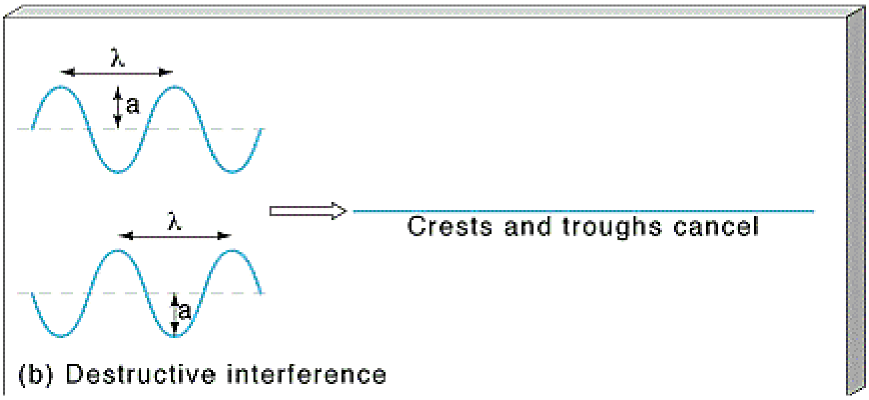


Fig. 4.11: Destructive inteference

When we move one of the speakers 0.5m (1/2 of the wavelength) further away. The volume on this speaker has to be turned up slightly so that the amplitude remains constant. This movement causes the condensations from one speaker to meet the rarefactions from the other sound wave and vice versa. Again referring to the principle of linear superposition, the result is a cancellation of the two waves. The rarefactions from one wave are offset by the condensation from the other wave producing constant air pressure. A constant air pressure means that you can hear no sound coming from the speakers. This is called ‘destructive interference’ where two waves are exactly out of phase.

4.3.2.3 Impedance coupling / acoustic impedance

Sound intensity is associated with the product of sound pressure and particle velocity and quantifies the amount of sound. The specific acoustic impedance is related with the ratio of both, and is a useful quantity to determine the reflection or absorption of sound by matter.

$Z = c\rho$ $\quad Z$ = acoustic impedance [kg/m^{2}s]

c = sound velocity [m/s]

ρ = density of transmitting medium [kg/m^3]

Sound energy quantifies how much energy is stored in an acoustic wave, sound intensity quantifies how much sound energy is transported and specific acoustic impedance quantifies the possibility for sound energy to be transported.

Sound pressure and particle velocity depend on the medium/material and the physical condition which shows that acoustic impedance is primarily a material constant. As a result only a small part of sound energy will be transported between two mediums with a different value for the acoustic impedance. The bigger the difference, the smaller the amount of transmission. Researchers throughout the world use this fact to develop barriers for noise and vibration transmission in many different ways.

In some cases, destructive interference and impedance coupling can be two sides of the same coin; in other cases destructive interference occurs without impedance coupling. The difference is related to whether the acoustic waves decay with distance traveled.

Table 4.1: Acoustic impedances

Material	Density [kg/m$^{3]}$	Acoustic velocity (longitudinal wave) [m/s]	Acoustic impedance (longitudinal wave) [10^6 kg/m^{2}s]
Steel	7800	5960	46,6
Concrete	2400	3000 - 4500	7,2 – 10, 8
Water (20°C)	1000	1480	1,48
Air (0°C)	1,3	331	0,0043
Air (20°C)	1,2	343	0,0041
Acrylic plastic	1180	2730	3,2

Sound from a speaker located in the middle of a stadium decays (is less loud) at a distance because of spherical spreading. As you get farther away, the sound energy is spread out over an increasingly large area. Go far enough away and the sound decays down to zero. On the other hand, sound in a "waveguide" such as a duct can travel long distances without significant decay. There are many situations in which walls, ducts, buildings, roadways, or other surfaces can act as waveguides for sound.

If a control system actuator is close to the disturbance source, destructive interference and impedance coupling can both occur. But what about when the actuator is far away from the disturbance, so far away that any wave it creates decays completely down to nothing before reaching the disturbance? There can still be destructive interference near the actuator, even though the actuator cannot possibly affect the impedance seen by the disturbance. Example: the tiny speaker in an active control headphone will not effect the impedance seen by a cannon firing a mile away, but it can create destructive interference within the headphone.

4.3.3 Feedforward and Feedback

An important property of modern active sound control systems (particularly feedforward systems) is that they are self-tuning so that they can adapt to small changes in the system. Usually the controller employs some sort of mathematical model of the plant dynamics, and possibly of the actuators and sensors. Unfortunately, the plant can change over time because of changes in temperature or other operating conditions. If the plant changes too much, controller performance suffers because the plant behaves differently from what the controller expects. Changes only need to be small to cause a non adaptive feedforward control system to become ineffective. An adaptive controller is one that monitors the plant and continually or periodically updates its internal model of the plant dynamics.

Non-adaptive controllers are generally confined to the feedback type in cases where slight changes in the environmental conditions will not be reflected in significant degredation in controller performance. One example of an effective non-adaptive feedback control application is in active ear protection and headsets where analog feedback control systems have been used successfully for some time. An interesting point is that an adaptive feedforward controller is effectively a closed loop implementation of a non-adaptive feedforward controller.

As integrated microprocessors dedicated to signal processing become cheaper and faster, potential active control applications increase in number. It should not be assumed that more processing power will extend the applications endlessly. There are some supposedly potential applications which will remain impractical, no matter how much processing power is available, because the limitations are a result of the structural/acoustic characteristics of the problem. Although more powerfull signal processing electronics help to alleviate higher frequencies and to more complex multichannel problems, the limitations mentioned remain. For the example mentioned above, a vast array of sensors and actuators would be required. As a result it would be cheaper to build a thicker wall.

It is usefull to start with an overview of single channel adaptive feedforward and feedback control. Non-adaptive feedforward systems are generally impractical for most industrial applications, because of the time variability in the physical system beeing controlled, and thus will not be considered further here. The simpliest example to consider for the illustration of the principle of feedforward and feedback control is the active control of plane wave sound in a duct and is shown in Fig.12 below.

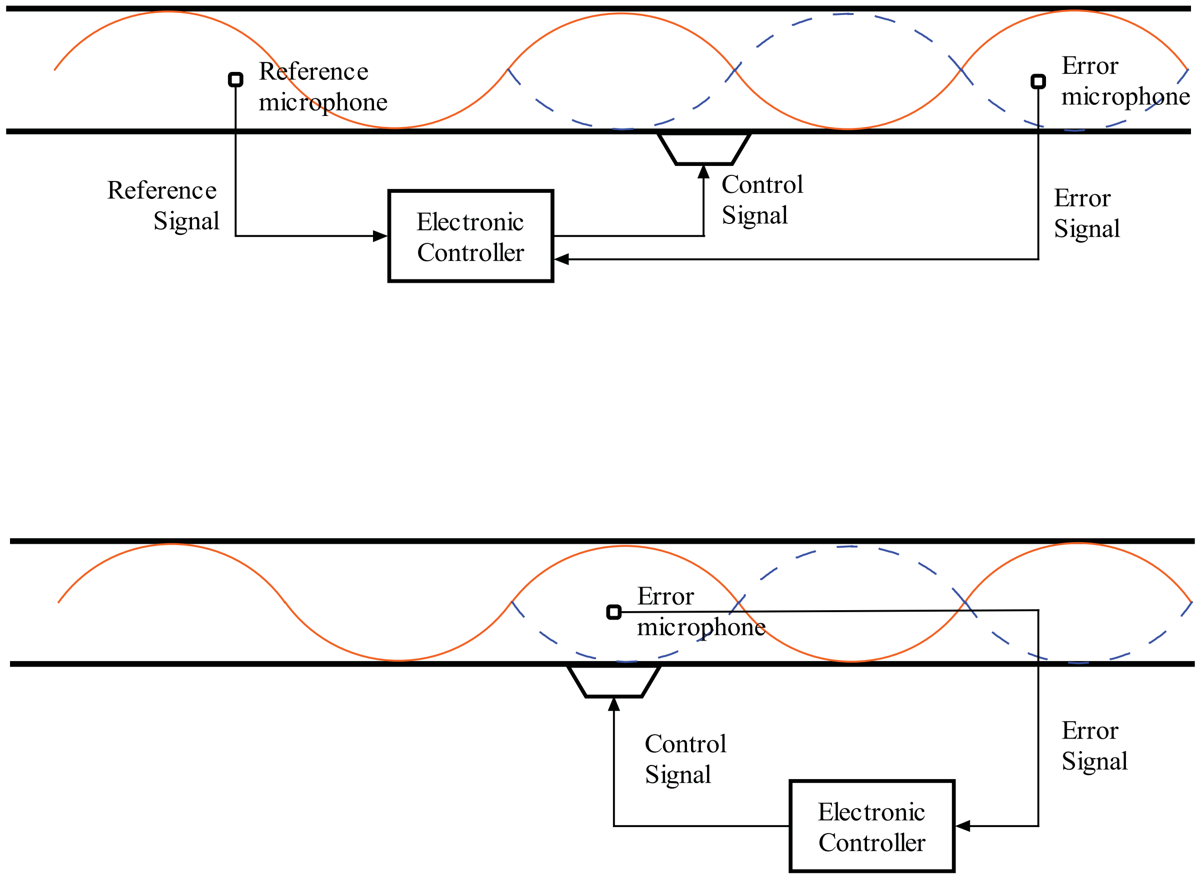


Figure 4.12: Principles of Feedforward (above) and Feedback [8]

4.3.3.1 Feedforward

When a signal correlated to the disturbance is available, feedforward adaptive filtering constitues an attractive alternative to feedback for disturbance rejection; it was originally developed for noise control, but it is very efficient for vibration control too. Its principle is explained in Figure 4.13.

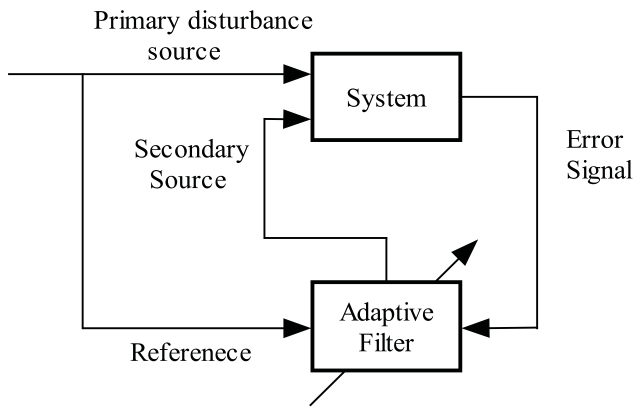


Fig. 4.13: Feedforward principle

The method relies on the availability of a reference signal correlated to the primary disturbance; this signal is passed through an adaptive filter, the output of which is applied to the system by secondary sources. The filter coefficients are adapted in such a way that the error signal at one or several critical points is minimized. The idea is to produce a secondary disturbance such that it cancels the effect of the primary disturbance at the location of the error sensor. Of course, there is no guarantee that the global response is also reduced at other locations and, unless the response is dominated by a single mode, there are places where the response can be amplified.

The method can therefore be considered as a local one, in contrast to feedback which is global. Unlike active damping which can only attenuate the disturbances near the resonances, feedforward works for any frequency and attempts to cancel the disturbance completely by generating a secondary signal of opposite phase (destructive interference).

The method does not need a model of the system, but the adaptation procedure relies on the measured impulse response. The approach works better for narrow-band disturbances, but wide-band applications have also been reported. Because it is less sensitive to phase lag than feedback, feedforward control can be used at higher frequency.

The main limitation of feedforward adaptive filtering is the availability of a reference signal correlated to the disturbance. There are many applications where such a signal can be readily available from a sensor located on the propagation path of the perturbation. For disturbances induced by rotating machinery, an impulse train generated by the rotation of the main shaft can be used as reference. Table below summarizes the main features of the two approaches.

4.3.3.2 Feedback

The principle of feedback is presented in Figure 4.14; the output y of the system is compared to the reference input r and the error signal, e = r - y, is passed into a compensator H and applied to the system G.

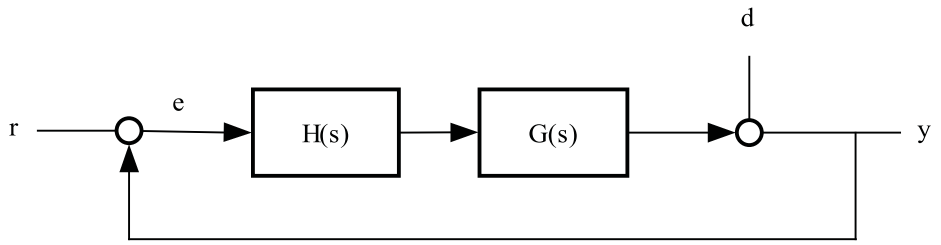


Fig. 4.14: Feedback principle

The design problem consists of finding the appropriate compensator H such that the closed-loop system is stable and behaves in the appropriate manner. In the control of lightly damped structures, feedback control is used for two distinct and somewhat complementary purposes: active damping and model-based feedback.

The objective of active damping is to reduce the resonant peaks of the closed-loop transfert function. Active damping can generally be achieved with moderate gains. Another nice property is that is can be achieved without a model of the structure and with guaranteed stability, provided that the actuator and sensor are collocated and have perfect dynamics. Of course actuators and sensors always have finite dynamics and any active damping system has a finite bandwidth.

The control objectives can be more ambitious and we may wish to keep a control variable (a position, or the pointing of an antenna) to a desired value in spite of external disturbances d in some frequency range. In general an elaborate strategy involving a mathematical model of the system which can only be a low-dimensional approximation of the actual system G is necessary.

When implemented digitally, the sampling frequency fs must always be two order of magnitude larger than fc to preserve reasonably the behavior of the continuous system. This puts some hardware restrictions on the bandwidth of the control system [9].

Table 4.2: Comparison of control strategies

Type of control	Advantages	Disadvantages
Feedback Active damping Model based	no model needed guaranteed stability global method attenuates all disturbance within ω_c	effective use only near resonances limited bandwidth disturbances outside ω_c are amplified spillover
Feedforward Adaptive filtering of reference	no model necessary wider bandwidth works better for narrow- band disturbance	reference needed local method large amount of real time computation

4.3.4 Active vibration control

4.3.4.1 Fundamentals

Vibration control is aimed at reducing or modifying the vibration level of a mechanical structure. Contrary to passive methods (dampers, shock mounts for machines, acoustic packing, various foams, etc.), active control is based on superimposing secondary noise or vibration sources on primary sources to obtain a minimum residual signal.

To reduce the acoustic level in structures with a more complex geometry or free-field noise, it was attempted to reduce the noise at the source, by modifying the vibration behavior of the structures where the noise originates. Commonly, active solutions supplement passive ones, especially in the low frequency domain where the passive systems are not as effective. This is known as active vibroacoustic control.

When applied to a structure, instead of creating an antinoise wave, the principle consists of locating vibration sensors on the structure or in the outside space and actuators capable of creating vibrations in the structure to achieve the smallest vibration. The sensors and actuators are coupled with an electronic control system, either an analog system or a digital computer, which calculates the signal and the phase shift to be applied to the actuators in real time.

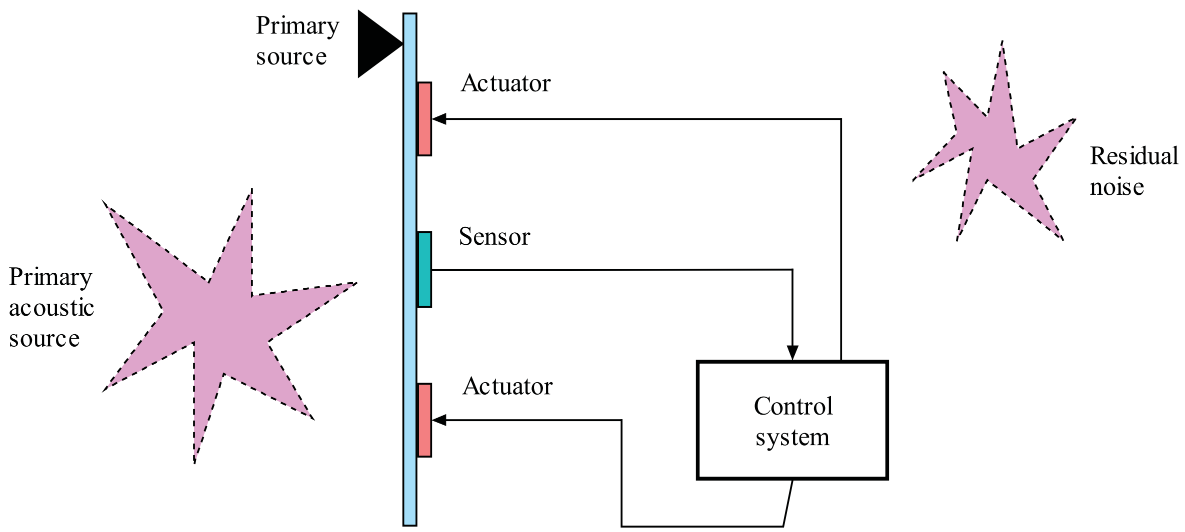


Fig. 4.15: Principle of active vibroacoustic control [7]

It may also be attempted to reduce the vibration level of a structure for the structure itself, not for an acoustic purpose, to improve comfort, increase structural fatigue strength or to protect sensitive equipment. This is the fundamental area of active vibration control applied to microvibrations.Active control of structures is a multidisciplinarian field involving the basic disciplines of structural dynamics, fluid-structure coupling, acoustics, automatic control, and materials research, since it is increasingly attempted to include the active control sensor and actuator functions in the material. This results in intelligent structures.

4.3.4.2 Smart materials and structures

An active structure consists of a structure provided with a set of actuators and sensors coupled by a controller. If the bandwidth of the controller includes some vibration modes of the structure, its dynamic response must be considered. If the set of actuators and sensors are located at discrete points of the structure, they can be treated separately. The distinctive feature of smart structures is that the actuators and sensors are often distributed and have a high degree of integration inside the structure, which makes a separate modelling impossible.

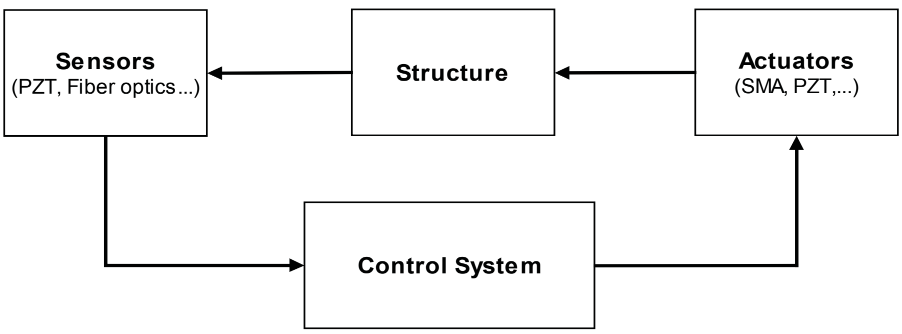


Fig. 4.16: Setup for the use of smart materials

In some applications like vibroacoustics, the behavior of the structure itself is highly coupled with the surrounding medium; this also requires a coupled modelling.

From a mechanical point of view, classical structural materials are entirely described by their elastic constants relating stress and strain, and their thermal expansion coefficient relating the strain to the temperature. Smart materials are materials where strain can also be generated by different mechanisms involving temperature, electric field or magnetic field, etc... as a result of some coupling in their constitutive equations. The most celebrated smart materials are briefly described below:

Shape Memory Alloys (SMA)

SMAs allow one to recover up to 5% strain from the phase change induced by temperature. Although two-way applications are possible after education, SMAs are best suited for one-way tasks such as deployment. In any case, they can be used only at low frequency and for low precision applications, mainly because of the difficulty of cooling. Fatigue under thermal cycling is also a problem. SMAs are little used in vibration control.

Piezoelectric materials

They have a recoverable strain of 0.1% under electric field; they can be used as actuators as well as sensors. They are two broad classes of piezoelectric materials used in vibration control: ceramics and polymers. The piezopolymers are used mostly as sensors, because they require high voltages and they have a limited control authority; the best known is the polyvinylidene fluoride (PVF2). Piezoceramics are used extensively as actuators and sensors, for a wide range of frequency including ultrasonic applications. They are well suited for high precision in the nanometer range. The best known piezoceramic is the Lead Zirconate Titanate (PZT).

Magnetostrictive materials

Magnetostrictive materials have a recoverable strain of 0.15% under magnetic field. The maximum response is obtained when the material is subjected to compressive loads. Magnetostricitive actuators can be used as load carrying elements (in compression alone) and they have a long life time. They can also be used in high precision applications.

The range of available devices to measure position, velocity, acceleration and strain is extremely wide, and there are more to come, particulary in optomechanics. Displacements can be measured with inductive, capacitive and optical means (laser interferometer). The latter two have a resolution in the nanometer range. Strain can be measured with strain gages, piezoceramics, piezopolymers and fiber optics.

5 Applications

5.1 Introduction

The most successful demonstrations of active noise control have been for controlling noise in enclosed spaces such as ducts, vehicle cabins, exhaust pipes, and headphones. That most demonstrations have not yet made the transition into successful commercial products.

One exception, active noise control headphones, has achieved widespread commercial success. Active headphones (see Figure 5.1) use destructive interference to cancel low-frequency noise while still allowing the wearer to hear mid- and high-frequency sounds such as conversation and warning sirens. They cancel noise automatically with an additional 15–18dB reduction in background noise for clearer voice transmissions. The system comprises a pair of earmuffs containing speakers and one or more small circuit boards. Some include a built-in battery pack, and many allow exterior signal inputs such as music or voice communications. Used extensively by pilots, active headphones are considered indispensable in helicopters and noisy propeller-driven aircraft. Prices have dropped in recent years. Passenger headsets, which lack the microphone boom found on pilots headsets, are even cheaper.

Fig. 5.1: An active Headset

Another application that has seen some commercial success is active mufflers for industrial engine exhaust stacks (see chapter 5.4). Active control mufflers have seen years of service on commercial compressors, generators, and so forth. As unit prices for active automobile mufflers have fallen in recent years, several automobile manufacturers are now considering active mufflers for future production cars (see chapter 5.3).

Large industrial fans have also benefited from active control. Speakers placed around the fan intake or outlet not only reduce low-frequency noise downstream and/or upstream, but they also improve efficiency to such an extent that they pay for themselves within a year or two.

The idea of canceling low-frequency noise inside vehicle cabins has received much attention. Most major aircraft manufacturers are developing such systems, especially for noisy propeller-driven aircraft. Speakers in the wall panels can reduce noise generated as the propeller tips pass by the aircraft fuselage. For instance, a system by Noise Cancellation Technologies now comes as standard equipment on the new Saab 2000 aircrafts. The key advantage is a dramatic weight savings compared to passive treatments alone.

Automobile manufacturers are considering active control for reducing low-frequency noise inside car interiors. The car stereo speakers superpose cancellation signals over the normal music signal to cancel muffler noise and other sounds. For example, Lotus produces such a system for sale to other automobile manufacturers. Unit cost is a major consideration for automobile use. While such systems are not at all common, at least one vehicle (currently offered only in Japan) includes such a system as a factory option.

In the following you can see a list of applications for active control of noise and vibration. The list includes topics which are currently being investigated by research groups throughout the world.

- Control of **aircraft interior noise** by use of lightweight vibration sources on the fuselage and acoustic sources inside the fuselage.
- Reduction of **helicopter cabin noise** by active vibration isolation of the rotor and gearbox from the cabin.
- Reduction of **noise radiated by ships and submarines** by active vibration isolation of interior mounted machinery (using active elements in parallel with passive elements) and active reduction of vibratory power transmission along the hull, using vibration actuators on the hull.
- Reduction of **internal combustion engine exhaust noise** by use of acoustic control sources at the exhaust outlet or by use of high intensity acoustic sources mounted on the exhaust pipe and radiating into the pipe at some distance from the exhaust outlet.
- Reduction of **low frequency noise** radiated by industrial noise sources such as vacuum pumps, forced air blowers, cooling towers and gas turbine exhausts, by use of acoustic control sources.
- **Lightweight machinery enclosures** with active control for low frequency noise reduction.
- Control of **tonal noise** radiated by turbo-machinery (including aircraft engines).
- Reduction of low frequency noise propagating in **air conditioning systems** by use of acoustic sources radiating into the duct airway.
- Reduction of **electrical transformer noise** either by using a secondary, perforated lightweight skin surrounding the transformer and driven by vibration sources or by attaching vibration sources directly to the transformer tank. Use of acoustic control sources for this purpose is also being investigated, but a large number of sources are required to obtain global control.
- Reduction of **noise inside automobiles** using acoustic sources inside the cabin and lightweight vibration actuators on the body panels.
- **Active headsets and earmuffs**

- Active control systems which suppress the noise appearing in **furnaces and boiler** rooms. The noise generated by the combustion chamber of the furnace has different injurious influences: in extreme situations it can destroy the equipment, but even at normal conditions the noise is harmful for the personnel working in the boiler room. The noise leaving the boiler room is a heavy environmental load. Conventional methods of suppressing this noise using sound absorbers generally do not work well at such circumstances. On the one hand, the power of this noise is concentrated in the low-frequency range (below 500 Hz), on the other hand traditional noise absorbers cannot be installed on furnaces. Active cancelation of the furnace-noise seems to be a promising solution.

5.2 Transformer noise reduction

5.2.1 Introduction

One promising alternative in contrast to traditional means of controlling transformer noise, which involves the construction of full enclosures, is to use active sound cancellation. Recent studies have shown, that the global control can be achieved by a completely surrounding of the transformer with loudspeakers (Hesselmann [12] and Angevine [13]). They found that the attenuation was dependent on the number of control sources and that this dependence was stronger at low frequencies. Angevine and Wright [14] demonstrated some success using multiple loudspeaker type control sources to minimize the transformer noise. Eight loudspeakers were arranged in two rows of four in front of a transformer tank and eight error microphones were loctated 10 meters away from the tank (at the same side of the transformer as the loudspeakers). The result showed that significant noise reduction over a wide area (15 to 20 dB over an angle of 35 to 40 degrees) could be achieved.

Other researchers (M. McLoughlin, K. Brungardt, J. Vierengel and K. Weissmann) reported an attempt to actively control transformer noise using Active Structural Acoustic Control (ASAC). In their control system, piezoelectric actuators were fixed to the transformer tank as vibration control sources and microphones were used to sense the error signals. The control system with multiple inputs/outputs adapted to changes in the system and the system environment. The typical sound reduction was about 15 to 20 dB at 120 Hz and 10 to 12 dB at 240 Hz.

Fig. 5.2: The test transformer in the anechoic room

In the following chapter an active control system of noise radiated from a small transformer (50 kV, dimensions 0,35m x 0,82m x 0,82m, see Figure 5.2) is described [15]. The physical system design for active control of noise radiated from structures deals with:

- The evaluation of error sensing strategies
- The selection of types of control actuators, i.e. what type of control source should be used: vibration or acoustic
- The optimization of the location and the number of the actuators and error sensors

The effects of the sensing strategies and the error sensor arrangement (location and number) on the control performance were numerically studied. A small transformer was located in an anechoic room for testing. The control system consists of eight control shakers mounted on the transformer, eight error microphones located in the near field and a ten channel controler. The control results were evaluated by measuring the sound field on an enclosed surface surrounding the transformer. An optimal search procedure was employed to optimize the error sensor layout based on transfer function measurement between the eight control sourcesand a large number of possible error sensor locations.

5.2.2 Measurement of sound field

For safety reasons the transformer was not energized. To generate noise, the transformer was excited by one internal type shaker at 100 Hz and 200 Hz. It was found that this excitation provided a sufficiently complex vibration pattern on the transformer tank to simulate the complexity of the pattern that occured in practice. For measuring the sound field the transformer was surounded with a frame for holding sound intensity probes around all four sides and over the top. The distance of the frame to the transformer tank was 0,8m. Six intensity probes were fixed on a beam for sensing thge sound field. Noise radiated by the transformer was measured at 492 sensing locations at 0,25cm intervals by sliding the beam along the frame during the data acquisition. Eight internal type shakers, acting as control sources, were mounted on the transformer tank.

The transfer function from each loudspeaker input to two pressure microphone outputs on each sound intensity probe were measured. Each shaker was driven through a power amplifier by a signal generator that produced sine waves at frequencies of 100 Hz and 200 Hz. For analysing the control mechanism, structural responses to shakers were measured at 224 sensing points as well. Vibration data were recorded to the data aquisition system. The block diagramm of the experimental setup is shown in Figure 5.3.

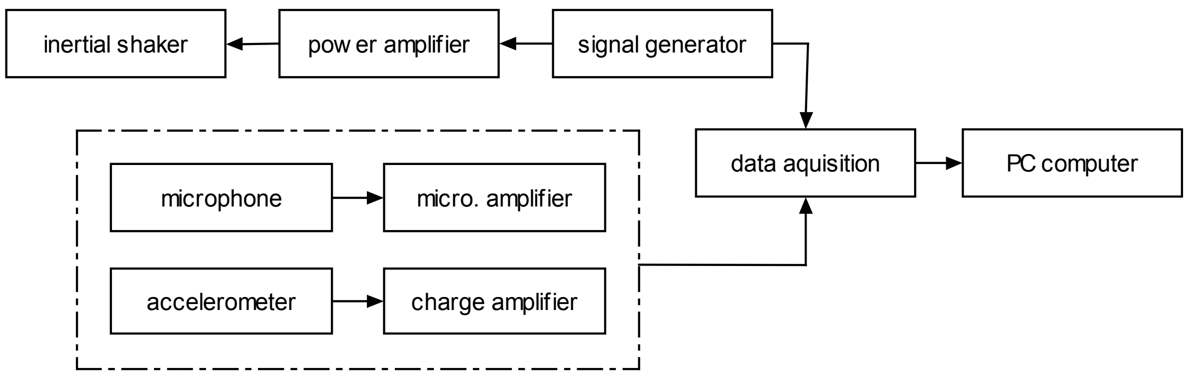
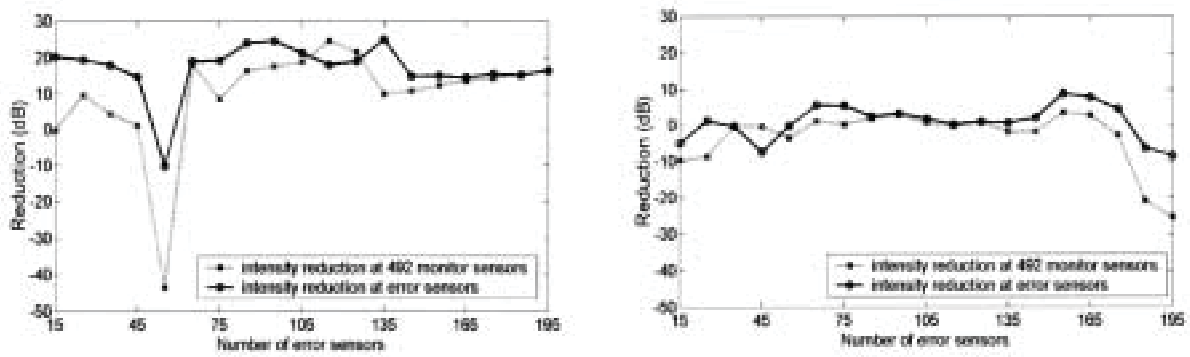


Fig. 5.3: The experimental setup

5.2.2.1 Control achieved by sound intensity minimization

To predict the average sound intensity level reduction achieved by intensity minimization using the setup given in Figure 5.3, the primary sound pressures at two microphones, on the sound intensity probe and the transfer functions between the control sources were measured. The optimal control forces resulting in the minimization of the cost function at the error sensors were then used to calculate the average sound intensity level reduction at 492 monitor sensors. Figure 3 shows predicted results associated with correctly spaced error sensors.

From Figure 5.4, one can see that the average sound intensity level reduction at 100 Hz achieved by sound intensity minimization using evenly spaced error sensors fluctuated wildly as a function of the number of error sensors unless a large number of error sensors were used. This is because the cost function is the sum of the sound intensities and the optimum result is an average of values measured at the 195 error sensors. It does not follow that the sound field reduction will increase as the number of error sensors increases when an insufficient number of error sensors are used to sense the sound field.



a) 100 Hz **b) 200 Hz**

Fig. 5.4: Variation of the average sound intensity level reduction with the number of error sensors

The transfer functions, from the control sources to the error sensors, depends on the relative positions between the control sources and the error sensors. The unexpected poor control performance at the error sensors may be avoided by optimizing the control source locations and/or the error sensor locations, to rearrange the transfer functions from the control sources to the error sensors. In this case, the function for optimizing the control source locations and the error sensor locations is the maximum average sound intensity level reduction at the error sensors, because the objective is to improve the control performance at the error sensors. Figure 5.5 shows the maximum average sound intensity level reduction at the error sensors as a function of the number of error sensors, when using the optimally located error sensors.

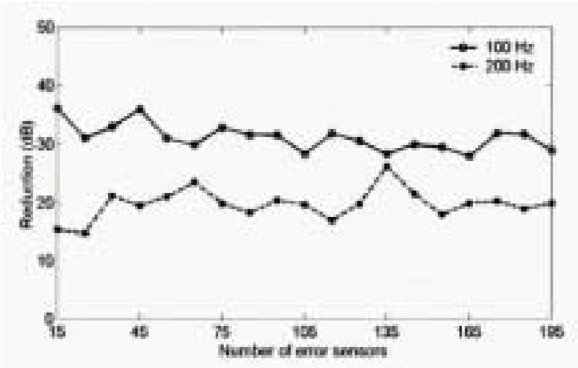
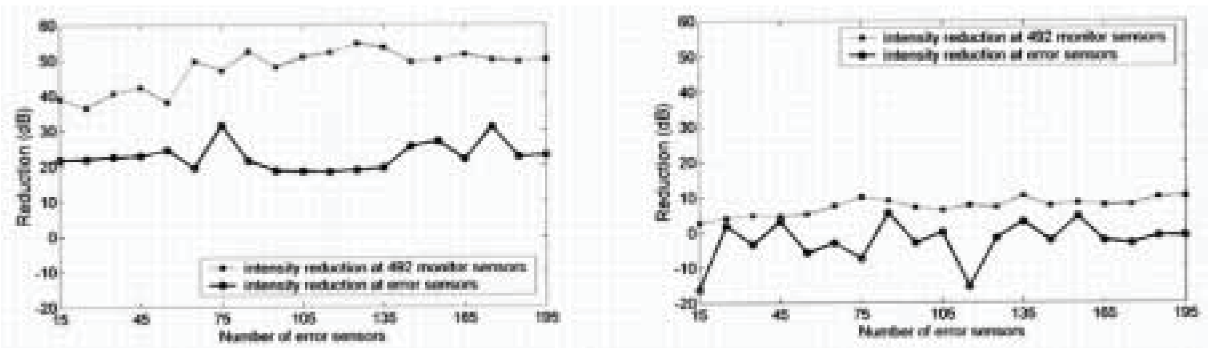


Fig. 5.5: Variation of the average sound intensity level reduction at the error sensors as a function of the number of error sensors, associated with the optimum error sensor locations that correspond to the maximum sound intensity reduction at the error sensors

Comparing the results in Figure 5.4 with the respective error sensor measurements in Figure 5.5, one can see that the average intensity reductions at the error sensors were significantly improved after optimizing the error sensor locations that correspond to the average maximum sound intensity level reduction at the error sensors. It should be emphasized that the maximum average sound intensity level reduction at a finite number of error sensors does not necessarily imply that a maximum global sound power reduction will be achieved. This is because generally, there is no inherent relation between the sound intensity level reduction at the error sensors in the near field and that at the monitor sensors used for evaluation of the control performance. In the work described here, the objective function for optimization of the error sensor locations is the maximum average sound intensity reduction at the monitor sensors, which represents the global control of noise radiated from the source. It should be noted that in general, the optimum error sensor locations depend on the primary source distribution and therefore the optimum locations will be changed if the excitation conditions of the transformer tank are changed.

In this study, due to the limitation of the potential locations of the control sources, an effort was made to improve the control results by optimizing the error sensor locations based on the fixed control source arrangement. The average sound intensity reductions at the monitor sensors obtained by sound intensity minimization using the optimal error sensor locations are shown in Figure 5.6.



a) 100 Hz **b) 200 Hz**

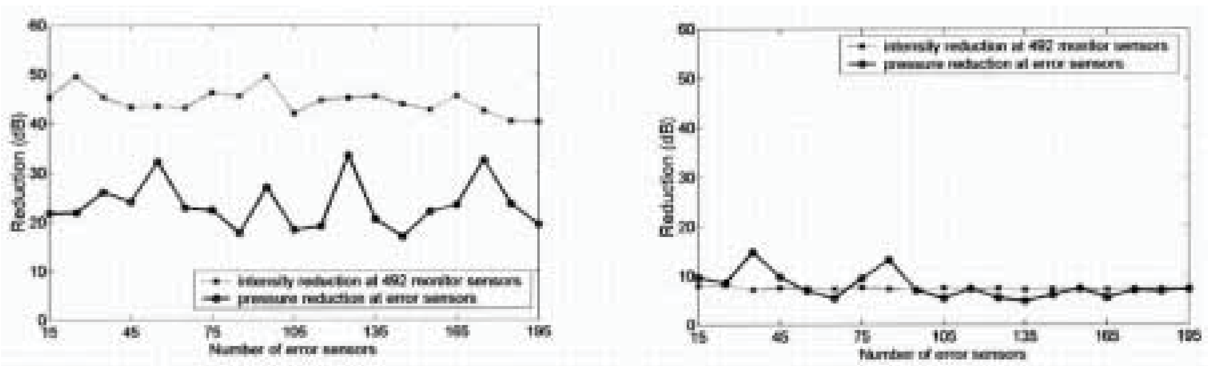
Fig. 5.6: Variation of the average sound intensity level reduction as a function of the number of error sensors, associated with the optimum error sensor locations that correspond to the maximum sound intensity reduction at the monitor sensors (intensity minimization)

From Figure 5.6 it can be seen that for 100 Hz excitation, the average sound intensity level reduction at 492 monitor sensors achieved using the optimally located error sensors was significant, even for the case of a small number of error sensors. The figure shows that the average intensity reduction at the error sensors is still negative (i.e. a sound power gain) for some of the 200 Hz control cases (for instance 15 error sensors at 200 Hz).

Figure 5.6 also shows that it is very hard to achieve a significant global sound intensity reduction at 200 Hz even though error sensors were optimally located. The optimization illustrates how the control mechanisms work for an active control System. More details about the control mechanisms are discussed later.

5.2.2.2 Control achieved by squared pressure minimization

In this section, the aim is to demonstrate the extent of control performance at monitor sensors achieved by minimizing the sum of the squared sound pressures at the optimum error sensors. To do this, the same prediction procedure as was used in last section for sound intensity minimization was employed and Figure 5.7 shows the average sound intensity reduction associated with the optimal error sensor locations, which corresponded to the maximum average noise reduction at monitor sensors.



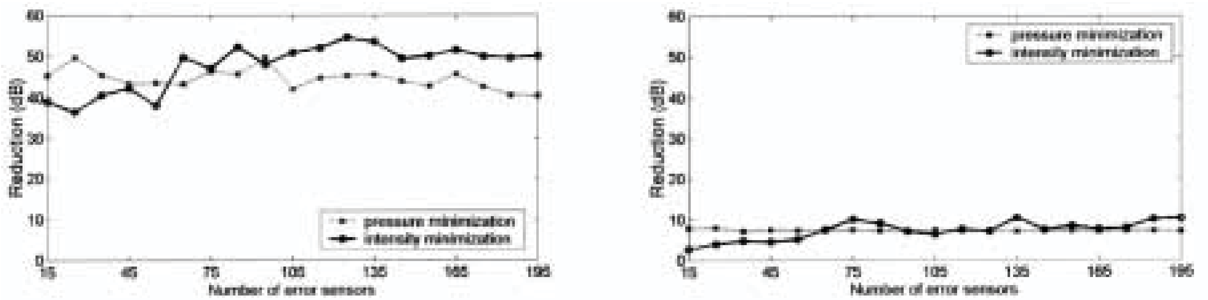
a) **100 Hz** b) **200 Hz**

Fig. 5.7: Variation of sound field reduction as a function of the number of error sensors, associated with the optimum error sensor locations that correspond to the maximum sound intensity reduction at the monitor sensors (sqaured pressure minimization)

Similar to the case of intensity minimization, a significant average sound intensity reduction at the monitor sensors was achieved using the optimum error sensor locations at 100 Hz, but poor control results were obtained at 200 Hz.

5.2.2.3 Effect of error sensing strategies on the control performance

The comparison of the maximum average sound intensity reduction at the monitor sensors achieved by intensity minimization with that achieved by squared pressure minimization is shown in Figure 5.8. The results demonstrate that, for the experimental configuration tested, better control performance can be achieved by minimizing the sum of the squared sound pressures than can be achieved by minimizing the sum of the sound intensity when a small number of error sensors are used. This is because more error sensing points are required to accurately describe the intensity field than are required to describe the pressure field. This conclusion agrees with that reached in earlier work [20].



a) 100 Hz **b) 200 Hz**

Fig 5.8: Comparison of the sound intensity level reduction achieved by intensity minimization with that achieved by sqared pressure minimization at the monitor sensors (associated with optimum error sensor locations)

In Figure 5.8(a) one can see that if a large number of error sensors are used, the noise reduction achieved by intensity minimization at 100 Hz is approximately 10 dB higher than that obtained by squared pressure minimization. The reason is that in the case of this physical system configuration, the sound field at 100 Hz is characterized as the transition region (between the hydrodynamic near field and the geometric near field). In this region of the sound field, due to the complexity of the sound field, the measurement of the sound pressure may not give an indication of the sound power radiated by the source. In other words, rather than squared pressure minimization, a good control result in this region may be achieved by sound intensity minimization provided that a large number of error sensors are used.

5.2.3 Experimental results

5.2.3.1 Evaluation of control performance at the monitor sensors

For the verification of the predicted results, experiments were carried out with a ten channel control system which was specially developed for transformer noise control. The control results at 100 Hz associated with 8 error sensors and 8 vibration control sources were evaluated at 526 monitor sensing points that were on a frame surrounding the transformer. The distance of the frame to the transformer was 0.8 m. Figure 5.9 shows the coordinate system for defining the monitor sensor locations. An inertial shaker was used to excite the transformer and to generate the primary sound field. Eight inertial type shakers were used as the control sources. Signals from the error microphones were input to the controller through electret microphone amplifiers and the output signals from the controller were used to drive the control shakers through the power amplifiers.

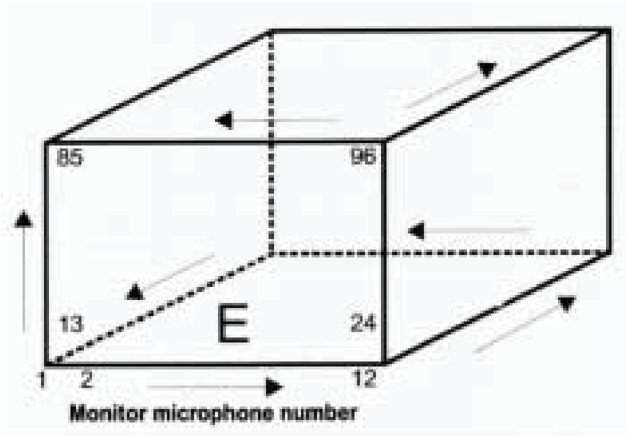


Fig 5.9: Coordinate system for monitor microphones (sensors)

A Graphic User Interface (GUI) on the controller was used to communicate with the controller and to monitor the error signals while the control was on so that it was known if a maximum level of sound field reduction was reached. Once the minimum error signal, which represented the sum of the squared sound pressures at the error sensors was reached, the control adaptation was set to off. Then the control shakers were constantly driven by the Controller during the measurement of the sound field at the 526 monitor locations. The experimental set-up is shown in Figure 5.10.

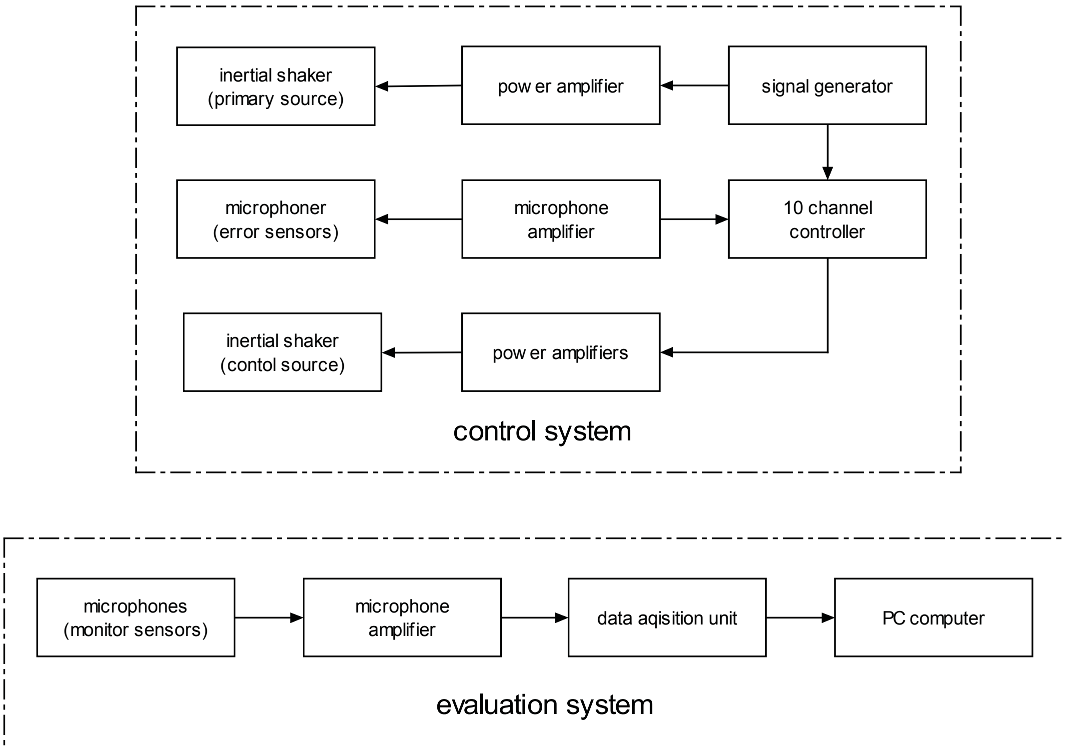


Fig 5.10: Experimental set-up for evaluating global control performance

Figure 5.11 shows the sound pressure reduction levels at the monitor sensing locations. From the figure it can be seen that noise was significantly reduced at most monitor sensing locations even though noise was increased at 3 sensing points. The largest overall sound pressure reduction level was approximately 32.7 dB and the largest overall increase of the sound pressure level was 2.7 dB. The average sound pressure level reduction at the monitor sensor locations was 15.8 dB. This result is similar to the predicted result of 18.7 dB.

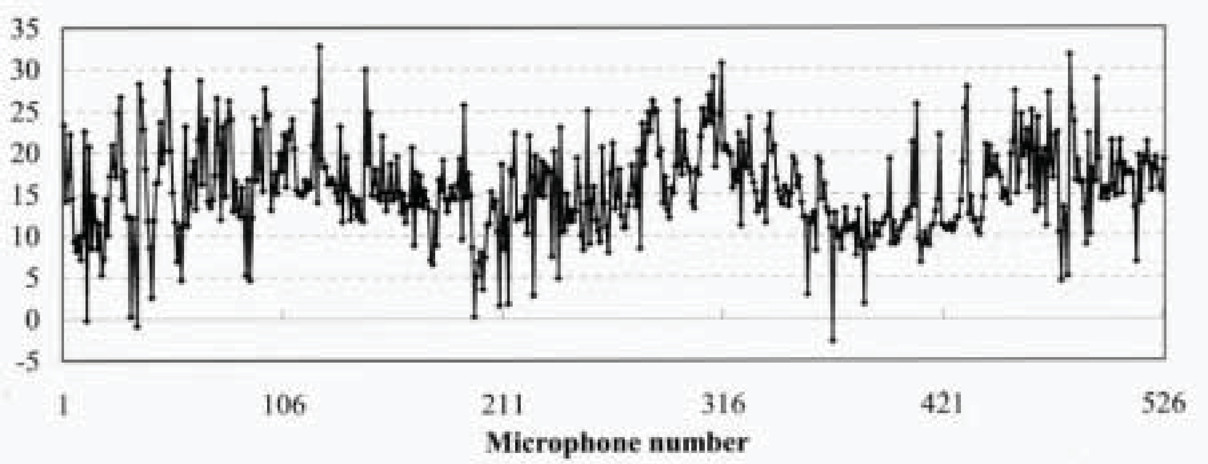


Fig. 5.11: Sound pressure level reduction at the monitor microphones for 100 Hz in the near field

5.2.3.2 Effect of the number of control sources on the control results

The effect of the number of control sources on the control performance at the error sensors is evaluated using measured data in this section. Due to the limitation of the number of controller channels, only 8 error sensors were used, so that the number of control sources were from one to eight, otherwise, an infinite number of 'optimum' control source strength vectors will be produced. For the case of 8 control sources, the sound pressure reduction at the eight error sensors tends theoretically to an infinite value. In other words, the residual (controlled) sound field at the error sensors is zero. To account for the influence of noise on the simulation results, a 1% error in the control forces was added. This results in the noise reduction predicted using 8 control sources and 8 error sensors dropping from an infinite value to 40 dB. Table 5.1 shows predicted results and measured results.

Table 5.1: Average sound pressure level reduction at 8 error sensors using different numbers of control sources [dB]

Number of controls		1	2	3	4	5	6	7	8
100 Hz	predicted	4,0	6,0	21,2	23,7	27,7	29,7	31,6	40,0
	measured	3,4	4,8	20,2	20,4	20,4	22,8	23,2	25,6
200 Hz	predicted	0,7	1,0	3,7	6,7	8,4	10,7	32,3	40,0
	measured	0,6	0,6	2,1	5,5	5,9	7,6	8,8	9,2

As expected, the average sound pressure level reduction at the error sensors increased as the number of control sources increased for both predicted and measured results. Table 5.1 shows very good agreement between the predicted and measured results for a small number of control sources. The difference between predicted and measured values increases as the number of control sources increases, and it is worse at the lower frequencies. This may be due to some background noise influencing the transfer function measurements.

5.2.4 Control mechanism

For active structural acoustic control systems, there are generally two control mechanisms responsible for attenuating the sound field when vibration control sources are used. The first is a reduction in the velocity levels of the principal offending modes. By this mechanism, termed modal control, the control effort is directed towards reducing the Vibration levels of the structure, especially the modal vibration levels of the dominant modes, so that the total sound power radiated by the vibrating structure is attenuated, provided that only the dominant vibration mode contributes significantly to the radiated sound power. The other way of attenuating the radiated sound field is to reduce the radiation efficiency of the structure by changing the structural velocity distribution without necessarily reducing the structural velocity levels. Attenuating the sound field by altering the amplitude and phase relationship between modes to reduce the sound radiation efficiency has been named modal rearrangement.

To analyze the control mechanisms in the described study, individual structural responses to each of the shakers (control and primary) were measured at 224 points at 100 mm intervals on the transformer tank. Then the vibration levels of the transformer tank after control were determined using resulting optimum force values calculated using a procedure for the minimization of the sum of the squared sound pressures at the error microphones.

The results demonstrate that the average squared structural velocity reduction level at the sensing points is 4.3 dB for 100 Hz excitation. By comparing this value with the predicted sound pressure reduction of 18.7 dB, it can be concluded that the sound field must be minimized by modal rearrangement of the transformer tank vibration. Figure 5.12 shows the distribution of vibration level on the transformer tank before and after control.

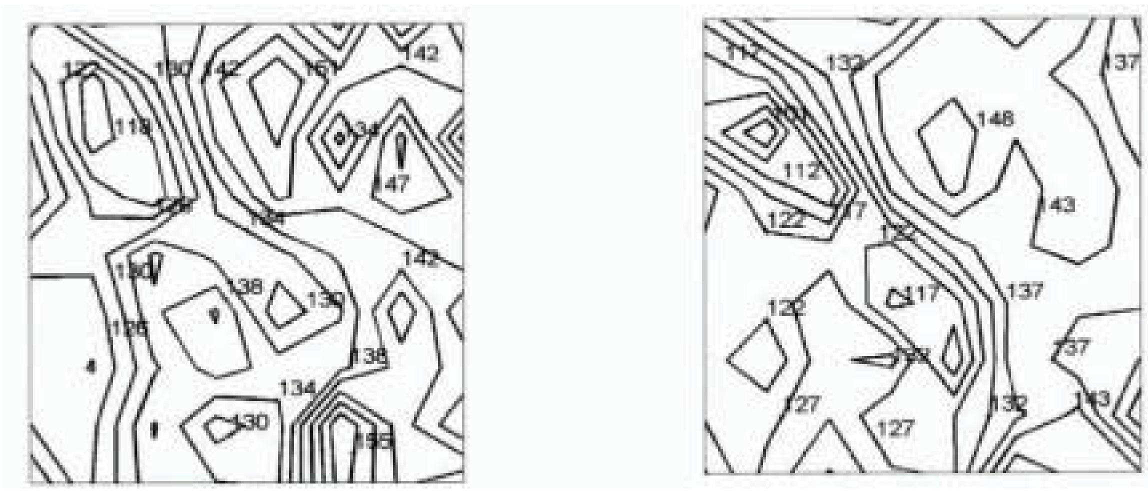


Fig. 5.12: Distribution of velocity level on the transformer tank corresponding to the minimization of the sum of the squared sound pressure using 8 control sources and 8 error sensors, at 100 Hz

5.2.5 Conclusions

The evaluation of the effect of error sensing strategies on the performance of an active system to control tonal noise radiation from a small transformer (0,35mx0,82mx0,82m) was described in this chapter.

The minimization of the sum of the sound intensities and the minimization of the sum of the squared sound pressures (two near field strategies), were numerically evaluated using transfer function data measured on a small transformer. The results represent the best that could be expected using an ideal feedforward active control system. The use of intensity minimization in the near field shows better results, provided that there are a large number of error sensors, otherwise, better results are achieved by using sqared pressure minimization.

For sound intensity minimization, good results can be achieved using a small number of optimally located error sensors. For example, when using 25 optimally located error sensors, approximatelx 37 dB of reduction was achieved, even though the result was around 15 db less than that achieved using squared pressure minimization. Further on, the results demonstrate that it was very difficult to minmize the sound field at 200 Hz, mainly because the modal rearrangement control mechanism was not very effective at this higher frequency that was characterized by shorter wavelength.

In an anechoic room the sound field radiated from a small transformer tank was also minimzed experimentally with a real time control system. To generate the primary sound field, the transformer tank was excited by an inertial shaker at 100 Hz and 200 Hz. Two inertial shakers were mounted to each side of the transformer. Eight error sensors were located on a frame 0,8m from the transformer. The sound pressure reduction achieved using 8 control actuators and 8 error sensors was evaluated at 526 monitor sensing points over the frame surface.

By analyzing the vibration pattern with and without control, it was shown that the sound reduction was mainly achieved by the modal rearrangement control mechanism.

5.3 Applications for cars

5.3.1 Introduction

The reduction of external noise from cars and trucks is nowadays an important issue. The legislators are continuously lowering the noise emission standards in the different countries. Also, the combustion engine developers increases the engine efficiency by lowering inlet and outlet valve resistance. As consequence, exhaust system manufacturers have to develop exhaust systems which combines a higher noise attenuation level with a lower flow resistance.

What sounds put the motorist to sleep? What sequences of tones increase a person's feeling of well-being? These are typical questions for psychoacoustics, a relatively new scientific discipline whose findings are increasingly falling on open ears in automobile development. Fundamental studies have shown, for example, that a low but constant sound level is by no means always regarded positively. In fact motorists make similarly high demands on the sound of a good engine as they do when listening to a symphony: Harmonious sequences of tones are particularly popular.

Previous researchers have demonstated the potential of active noise control in automobiles. Most of these works tended to deal with automobiles with low acoustic damping where modal seperation was high. There remains much work to be done to investigate the use of active noise control in cars with typical damping where there is increased modal overlap and reduced acoustic resonances. Other issues that need to be investigated are an improvement in the frequency bandwidth and global nature of the control. Because of the low frequency characteristics of typical disturbances (road noise), large, and often massive, conventional electromagnetic speakers are required to obtain good active control of the sound field in the car interior. This situation creates a demand in the automobile industry for new compact and lightweight acoustic sources.

5.3.2 How far is research?

5.3.2.1 Electronic Controlled Active Silencer Exhaust System for Car Engines

The engine noise is caused by pressure pulses released by the exhaust. When the expansion stroke of the engine comes to his end, the outlet valve opens and the remaining pressure in the cylinder discharges as a pulse into the exhaust system. These pulses are between 0.1 and 0.4 atmosphere in amplitude, with a pulse duration between 2 and 5 milliseconds. The frequency spectrum is directly correlated with the pulse duration. The cut-off frequency lies between 200 and 500 Hz.

A passive exhaust system attenuates the noise by absorption and reflection [11]. The acoustic waves are reflected back and forth into the exhaust system. At each reflection, the sound waves lose energy. As result, only a fraction of the noise leaves the exhaust outlet.

The proposed active exhaust system attenuates the noise in a very different way. The active exhaust system is mor or less a cube mounted on the engine block. It consists of a buffer volume which is connected to the exhaust outlet of the engine and on which an electrically controllable valve is mounted. The engine acts as a volume velocity source and pushes the flow into the buffer volume. The valve is controlled such that only the mean flow passes the valve. The flow fluctuations are stored temporally in the buffer volume. As result, the flow out of the exhaust outlet is free of fluctuations, and consequently free of noise.

Experimentally, a machine generates realistic engine exhaust noise with the gas flow using compressed air. It permits to study acoustical and flow-dynamic phenomena in exhaust systems and to experiment with new concepts of exhaust systems without taking precautions against the hot corrosive gases of a real engine.

In a combustion engine, the pressure enclosed in the cylinder discharges in the exhaust as soon the outlet valve opens. The piston is at his lower dead point and the cylinder volume changes only 10 a 15% during the escape time of the pulse. This means that the exhaust pulse can be approximated by the discharge of a constant volume. This assumption forms the basis of the cold engine simulator. A scheme of it is presented in Figure 5.13 below.

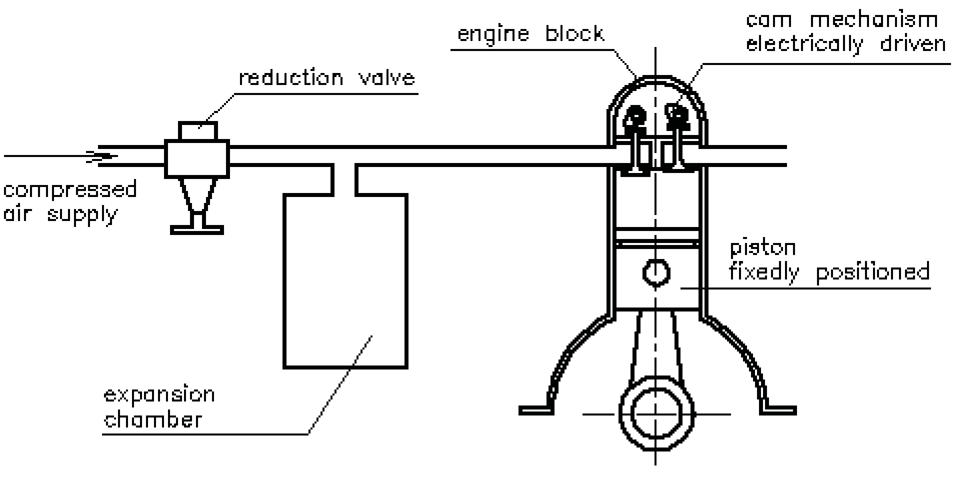


Fig. 5.13: Scheme of the cold engine simulator

It consists of a regular engine block whose pistons are fixed at their lower dead points. The inlet collector is connected via an expansion vessel and a pressure reduction valve to a normal pressurized air supply network. The cam mechanism of the engine block is driven by an electric motor. The supplied pressure at the inlet collector is equal to the pressure in the cylinder of an operational combustion engine at the end of the expansion stage. During the inlet stage, the cylinder charges at the same pressure level as applied at the inlet. When the outlet opens, the cylinder discharges and the pressure pulse enters the exhaust. The discharge takes a few milliseconds. These pressure pulses are compared with the pulses from a real combustion engine. The shape and frequency spectrum of the pulses are very similar.

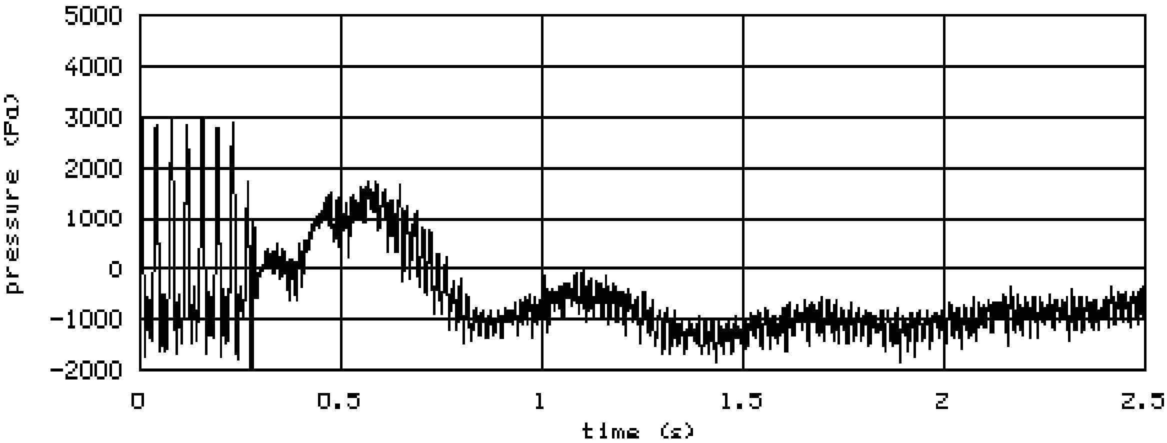


Fig. 5.14: Measurement of the sound pulses out of the exhaust outlet

The active silencer is tested on this setup using two types of electronic controllers. The first one tested is a feedforward non-adaptive controller resulting in a sound attenuation of 13 dB. The second one is a feedback controller, resulting in 16 dBA attenuation. In Figure 5.14 above a measurement of the sound pulses out of the exhaust outlet are presented. The controller is started after 0.35 seconds. The backpressure to the engine is between 0.03 and 0.1 atmosphere, while a passive system with the same attenuation needs 0.15 to 0.3 atmosphere.

The feedforward controller uses two sensors, one for pressure in the buffer volume and one for the flow through the control valve. The feedback controller uses only one sensor for the pressure after the control valve. Despite the better performance and the simpler instrumentation, the feedback controller is too sensitive for the dynamics of the engine and the exhaust system. At this instant, feedforward controllers are in investigation, in order to eliminate the flow sensor.

5.3.2.2 Active noise control inside a car with advanced speakers [16]

5.3.2.2.1 Introduction

This chapter demonstrates the applicability of active control of cabin noise in a sports utility vehicle, in which passive noise and vibration treatments are present. When controlling interior noise caused by an external disturbance, finding a proper location for the reference sensors is a key factor. The location of the error sensors, control actuators and reference sensors is of paramount importance in the design of a feedforward controller. Global control of the pressure field of highly damped cavity with multiple modes, such as an automobile cabin, is nearly impossible to achieve with a limited number of control sources (less than four), even at low frequencies (40 Hz to 500 Hz).

A more suitable goal is to create a zone of quiet around the heads of the driver or passenger. To reach this goal, the control sources can be located in the doors and the error sensors close to the position of the driver's and passenger's head. Nevertheless, the placement of the reference sensors requires more attention. When the disturbance source inside the cabin of an automobile is caused by either the power train or the interaction of the tires with the road surface, many other sources that radiate noise into the cabin need to be considered. It is difficult to analyse these sources because the acoustic and structural paths are complex. As a result the determination of the number and location of appropriate reference signals is not straightforward. A principal component analysis was performed to determine the number of independent noise sources present when the car was driven on a coarse road. Although this analysis does not provide any information regarding the location of the sensors, it provides information regarding the minimum number of reference sensors to be used. If the maximum achievable noise reduction is computed for various sets of reference sensors, their location can be optimized experimentally.

5.3.2.2.2 Active control of power train noise

The reference signals can be classified into two different categories:

- Those directly connected to the engine: accelerometers glued to the engine mount, to the oil fill system, as well as the signal from the tachometer and a microphone.
- Those attached to the body of the car: the vibration of the engine and power train induces the vibration of other component of the car. Each component connection induces non-linearity. As a result, the vibration of the firewll and body of the car, which radiate sound into the cabin, are not nessacerily linearly reladted to the vibration of the engine. Therefore, the reference sensors are often not sufficient to characterize the sound pressure measured by a microphone located inside the automobile cabin.

The pressure measured at the error sensors is multi-harmonic. The engine rotation speed, and particularly the firing frequency of the cylinders, determine the frequency of the signal. When the engine is running at 3300 rpm the firing frequency is 22,5 Hz (one half of the engine speed).

$$\frac{3300 \text{ rpm}}{60 \text{ s/min}} = 55 \text{ rps} \quad 55 \text{ Hz}$$

At the predominant frequency (110 Hz), the amplitude of the response at the harmonic varies from 30 dB above the background noise of the engine down to 5 dB at frequencies (275 Hz). The background noise of the engine is defined as the noise off peak (25 dB at a f = 250 Hz).

When six reference signals are used, good coherence (above 0,99) was obtained at the harmonics of the firing frequency of the cylinder. At other frequencies, where the sound pressure level is low before control, the coherence drops and very little control was achieved. For instance, the coherence was close to 0,3 at 250 Hz, which corresponds to weak attenuation. Nevertheless, some control was achieved offpeak, which decreased the background noise of the engine a couple of decibel on a wide frequency band. In fact, the total reduction at the error sensors was 12 dB in the 40-500 Hz band, while the maximum peak reduction at 110 Hz was 31 dB. Noise reductions obtained with various reference signal configurations have to be computed. As the number of reference signals decreases, the global attenuation decreases also. Reductions of 12, 9.7, 8.7 down to 4.5 dB were obtained respectively with six, four, three and one reference signal. The same trend was also seen for the peak reduction. As illustrated in Figure 5.15, using three reference sensors correctly positioned, good coherence (above 0.9) was obtained for all the harmonics of the firing frequency, and a maximum peak reduction of 25 dB was achieved. All the peaks were cancelled when this configuration of reference signals was used (less than one decibel above the background noise of the engine) as shown in Figure 5.15. Nevertheless, very little off-peak reduction was obtained. In the case where only one reference sensor was used, a 17 dB peak attenuation was achieved at one harmonic and 13 dB at the others.

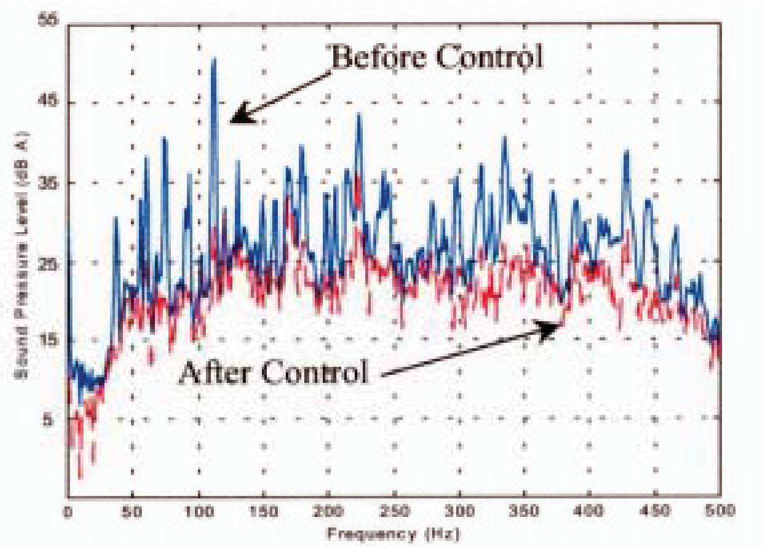
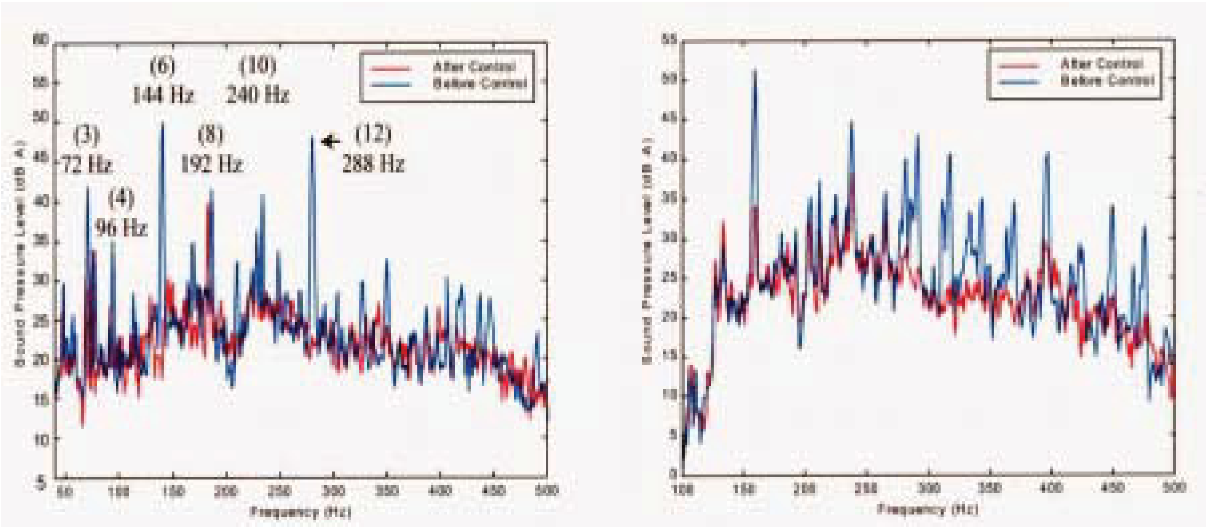


Fig. 5.15: Simulated sound pressure levels results with three reference signals

In conclusion, simulations showed that achievable reduction was maximized when six reference signals were used (31 dB peak and 12 dB global), but using three well positioned reference signals provided adequate results, since all the harmonics were cancelled resulting in a flat frequency response at the error sensors. In the experiments presented in the next section, three reference signals were used. One accelerometer was positioned on the oil fill stem and two on the firewall (one on the side of the driver and one on the side of the passenger).

Experimental active noise control results. The frequency response function measured at the error sensor located at the head of the passenger before and after control is shown in Figure 5.16 (a), in the case where two actuators and two error sensors were used. The error sensors were positioned symmetrically at the heads of the driver and passenger (Configuration number 2 in Table 5.2). One actuator was located in each front door at the position of the stereo system speakers. The response is shown A-weighted between 40 and 500 Hz. This response is typical of what was measured at the error sensor. The signal before control is shown in a solid line. Note from Figure 5.16 (a) that the response is highly dominated by the even harmonics of the firing frequency of the cylinders of the engine (the harmonics of the firing frequency that have the highest amplitude are shown with their frequency and number in parenthesis). For the case presented in Figure 5.16 (a) the engine was tuming at 2900 rpm, which corresponds to a fundamental frequency of 24Hz. The signal at the harmonics is as much as 30 dB (144 Hz) higher than the background noise (20 dB A).

The effect of the control is to cancel most of the peaks that contribute most significantly to the response. The maximum peak reduction is 30 dB at 144 Hz. In fact, the harmonics are reduced to a level close to the background noise of the engine. The global reduction in the frequency band of interest (40 to 500 Hz) is 7 dB.



a) Results obtained with conventional speakers **b) Results obtained with advanced speakers**

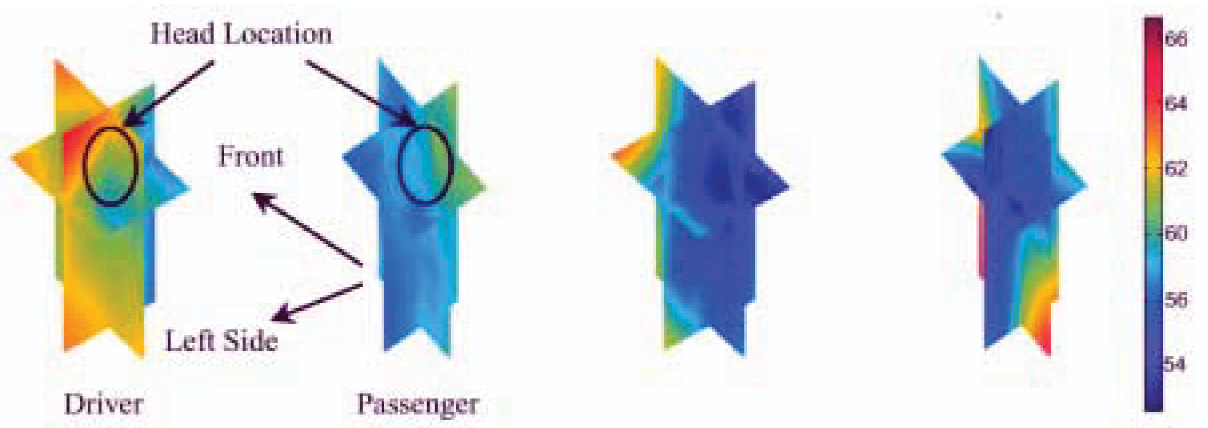
Fig. 5.16: Sound pressure level at error sensor 2

The noise reduction at the error sensors noted above is of great importance. In order to determine the effect that spillover may have on the results, the interior volume of the cabin was scanned to monitor the spatial distribution of the pressure before and after control. The total volume scanned is a cube of 80 by 80 by 70 cm (H by W by L), which represents the volume occupied by the head and torso of a person normally seated in the automobile.

Four configurations of actuators and error sensors were investigated for control in the front of the cabin. The first three configurations involved two control sources located in the front doors as described above. In configuration four, two extra moveable speakers were added on the dashboard. In the first configuration, two error sensors were located at head level on the side of the driver. In configuration two, one error sensor was located at the head position on the side ofthe driver and one on the side of the passenger. In configuration three and four, two extra error sensors were added at the head level towards the center of the cabin. After control, for all configurations, a zone of quiet was created around the error sensors. The dimensions of this zone are different for all the configurations. Because of high damping associated with the acoustic modes of the cabin global control of the pressure would require more actuators than the ones actually used; thus global control of the cabin was never achieved.

In the last three configurations the error sensors were located at both the driver and passenger positions. These locations were chosen in order to avoid the spillover occurring at the position of the passenger when no sensor is located on the passenger side. In the three cases reduction was obtained at both positions and spatial spillover occurred only in the corners of the scanned volume, where the head of the driver or passenger would not normally be while the car is in motion. The size of the zone of quiet created around the error microphones varied with the dimensions of the system. Although little difference in the pressure profile was obtained between configuration 2 (2 error sensors and 2 control sources) and configuration 4 (4 error sensors and 4 control sources), the configuration involving four speakers and four error sensors shows the best results.

As the number of speakers is increased, the number of degrees of freedom being controlled is higher and therefore the reduction is higher. Nevertheless, in the latter case, an increase of 6 dB occurred in one corner of the scanned zone, as shown in Figures 5.17 (a) and (b). In configuration 3 (4 error sensors and 2 control sources), very little spillover occurred. Only a 1 dB increase in the pressure was recorded in one corner of the volume. The reduction obtained at the error sensors is not as high as in configuration 4 (maximum 4 dB at the error sensors in configuration 3 and maximum 7 dB at the error sensors in configuration 4), but the reduction in the SPL is spatially more global.



a) Pressure before control [dB] **b) Pressure after control**

Fig. 5.17: Spatial distribution of the pressure (configuration 4)

These preliminary tests showed that it is possible to achieve power train noise control at the error sensors of up to seven decibel. In terms of spatial distribution of the pressure field, although true global control of the cabin noise would require more sources, it was shown that a 90-cm diameter zone of quiet could be created around the head of the driver and passenger over the 40 to 500 Hz bandwidth.

Control using advanced piezoelectric speakers. In the previous section, it was shown that active control of power train noise was achievable using premium stereo speakers. Results obtained using new compact lightweight sources are presented in this section. This new source is based on the use of double amplifier piezoelectric drivers that induce displacement of a membrane, which in tums radiates sound. The same configurations used during the experiments with the premium stereo speakers were no longer suitable because only two advanced sources could be built. For comparison purposes, configuration two and three presented in the previous section were repeated with the new source.

Figure 5.16 (b) shows the response at microphone one for the case where the sound field is controlled using two error sensors and two control sources. As it has been pointed out before, the sound pressure level before control, is highiy dominated by the response at the harmonics of the firing frequency of the cylinders.

As discussed by Couche and Füller [18], the output of the advanced source is low at frequencies below 150 Hz. Therefore, the control was performed in a reduced frequency band (150 to 500 Hz). In this particular case, the engine was tuming at 3200 rpm. The dashed line shows the response after control. Similarly to what was described above, the response after control at the harmonics is reduced to a level dose to the background noise of the engine. Reduction of 18 dB was obtained at 159 Hz, so that the response after control at this frequency is 6 dB above the background noise. Even though the attenuation is lower at other frequencies, for instance 14 dB at 291 Hz, the response after control is completely reduced into the background noise. Similar results are obtained at the highest harmonics.

In Table 5.2, global reductions measured at the error sensors and obtained with advanced sources are compared to reductions obtained with conventional speakers. The general trends of the results are very similar. When two error sensors were used, noise reduction close to seven decibel was obtained at error sensor one. In the case of the advanced source, lower attenuation was recorded at error sensor two. When four error sensors were used, maximum noise reduction close to five decibel was obtained at error sensor two. In the latter configuration results obtained with the conventional and advanced source were slightly different. Two main reasons were identified.

Table 5.2: Attenuation at error sensors (dB_A), advanced / conventional speakers

#	Configuration	Error 1	Error 2	Error 3	Error 4
1	2 by 2	- / 3,0	- / 3,3		
2	2 by 2	6,9 / 6,4	4,3 / 6,4		
3	2 by 4	2,4 / 2,6	4,8 / 4,0	1,7 / 2,7	3,8 / 4,0
4	4 by 4	- / 4,6	- / 6,7	- / 5,1	- / 7,0

First, when the control was performed with the piezoelectric sources, the engine speed was higher (3200 rpm instead of 2900 rpm). Thus, the amplitudes of the harmonics are larger and the frequencies at which they occur are higher. Second, when the control was performed with the advanced speakers, the first harmonic was filtered out because of the low output of these sources at low frequency (below 150 Hz).

Discussion. It has been shown that application of active control of power train noise in the interior cabin of a sport utility vehicle (Ford Explorer) yields good results. First, tests demonstrated that attenuation of up to seven decibel over a large frequency band (40 to 500 Hz) was achievable at the error sensors. The best results were obtained with a system involving three reference accelerometers, four conventional speakers used as control sources, and four error sensors located at head level inside the cabin of the vehicle. The response at the error sensors before control was strongly dominated by the response at the harmonics of the firing frequency of the cylinders. During the tests, the fundamental frequency was 24 Hz when the engine was tuming at 2900 rpm. The effect of the control was to reduce the amplitude of the peaks (maximum of 30 dB attenuation) at these frequencies. The response after control was flatter, and it was possible to lower the response at the harmonics close to the background noise level of the engine. Due to the very high damping inside the cabin of the automobile, many modes contribute to the response. Thus, in order to obtain global control of the pressure field, more control speakers would be required. In terms of spatial effect, spillover appeared only in the corners of the volume scanned (head and torso of passenger and driver), at positions where the head would not be positioned during normal use of the vehicle. In fact, a large zone of quiet was created around the error sensors (i.e. around the head of the driver and passenger). The extent of the zone of quiet was increased as the number of error sensors and actuators was increased. The largest zone of quiet had the shape of a sphere centered at the error sensors (i.e., head of the driver and passenger). The diameter of the sphere was approximately 90 cm when four error sensors and four control sources were used.

In the last section, prototype compact lightweight piezoelectric speakers were implemented as control sources. The results obtained with these speakers were comparable to those obtained with high end commercially available speakers. Nevertheless, the pressure response of these sources at very low frequencies (below 150 Hz) was too low to cancel the first harmonics of the signal. If the first harmonics were not filtered, the overall noise reduction would be decreased. In the case of configuration 2, the reduction at error sensor 2 was estimated to be 4.6 dB, instead of 6.9 dB if the first harmonic was not filtered.

5.3.2.2.3 Active control of simulated road noise

In most of the previous work conducted on simulated road noise [19], a smooth drum is used to drive one or both of the front wheels of the vehicle at a constant speed. As a result, the frequency response measured at a microphone located in the cabin is multi harmonic (due to the periodicity of the rotating drum). The fundamental of the signal occurs at the frequency of rotation of the drum. In this experiment, performed at Goodyear in Akron, Ohio, the drum, which is driving only the front left wheel, is covered with a rough aggregate such that the sound pressure measured in the cabin is not multi harmonic. Most of the signal energy is concentrated from 100 to 150 Hz and from 200 to 350 Hz. Such a response is typical of the response measured in the cabin while the automobile is being driven on a rough road. The vehicle used to perform the experiments presented in this section is a similar Ford Explorer residing at Goodyear. This is significant since it shows that the system can be readily applied to another vehide of same type.

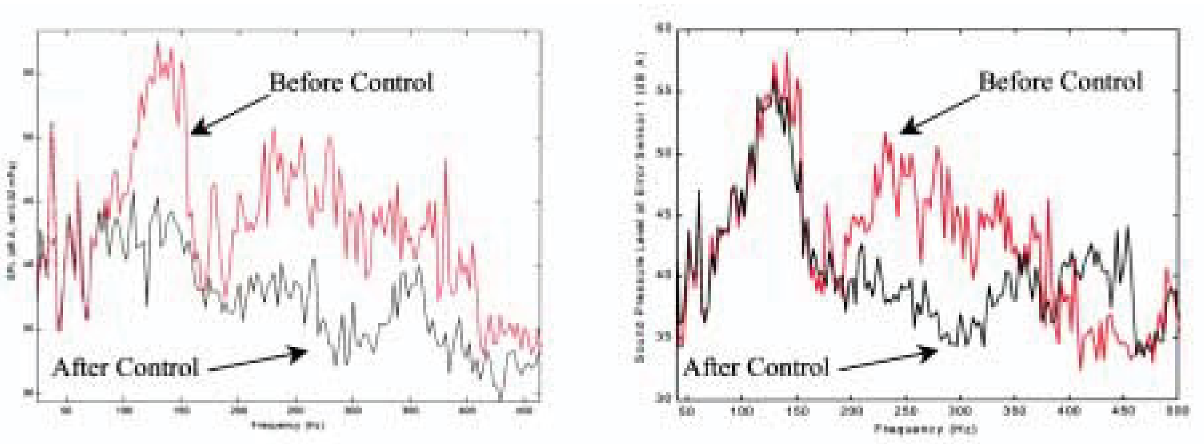
Principal component analysis. During this test, six accelerometers were fixed to the body of the car. They were equally spaced and located close to the wheel. In the entire frequency band, the amplitude of one of the singular values is much larger than the others, implying that only one independent source of vibration dominates. This is because only one wheel is being excited. Therefore, good control of the sound pressure inside the cabin is achievable with a low number of reference sensors. The main reason for the difference between the two tests is the fact that in the former the cabin is being excited by only one wheel. Principal component analysis as well as simulations based on the multiple coherence between candidate reference sensors and a microphone located in the cabin showed that a total of eight reference sensors would be necessary to obtain significant noise control during the rough road test.

Five candidate reference signals were investigated using the multiple coherence based Simulation. Three accelerometers were positioned in three perpendicular directions on the control arm as close as possible to the driven wheel. One accelerometer was located on the firewall, on the same side. Because of non linearities that may occur at each connector linking the wheel to the body of the car, the vibration of the wall may not be correlated with the vibrations measured close to the wheel. One microphone was also located just behind the tuming wheel to measure the noise due to the interaction of the road with the tire. Because the principal component analysis showed that only two reference signals were necessary to perform active noise control when only one wheel was driven, the best combination of two of these reference sensors was investigated. The best result was obtained when the reference accelerometers were located on the suspension arm in the z-direction (vertical motion of the wheel) and in the x-direction (transversal motion). A noise reduction close to 15 dB over the bandwidth 100 to 500 Hz was predicted at the error sensors. Other configurations involving the accelerometer located on the firewall and the signal from the control arm in the z-direction led to similar results. When the microphone was used with any another reference signal, poor attenuation (under 8 dB) was predicted.

Experimental active noise control results. In order to implement active control of road noise, two acoustic sources and two error sensors were used in a position similar to configuration two, used in the configuration for the control of power train noise. The reference sensors were located on the suspension arm as close as possible to the wheel: one measuring the acceleration in the z-direction (vertical motion ofthe wheel) and one in the x-direction (transversal motion of the wheel).

In Figure 5.18 (a), experimental results at one error sensor are shown using the conventional speakers as control sources. At error sensor 2, the reduction is over 11 dB in the frequency band of the control (100 - 500Hz). If the attenuation is computed on a wider frequency band (20 - 1000 Hz), the reduction is 9 dB. The reduction obtained at error sensor 1 is lower, 9 dB and 6 dB, when computed respectively on the frequency band of control and on the wider band. Note that for frequencies above 1000 Hz the response is much lower and therefore does not contribute much to the global response. These results are somewhat lower, than those predicted by the simulation (15 dB over the 100 - 500 Hz frequency band). The general trends in Figure 5.18 (a) are very obvious. Most of the control is obtained in the 100 - 150 Hz and 200 - 350 Hz frequency bands.

Results from the case where the control is performed with piezoelectric sources is presented in Figure 5.18 (b). The test set up remained the same, and only the sources were changed. The reduction obtained in the 100 - 500 Hz frequency band is low compared to the reduction achieved with conventional speakers. The sound pressure level was decreased by only 2.5 dB in this frequency band. As illustrated in Figure 5.18 (b), no control was achieved below 200 Hz. Poor results at low frequencies are caused by the dynamics of the source. As discussed before, the output of the source is low at frequencies below 200 Hz. Control achieved above 200 Hz is very similar to that achieved with conventional speakers. Reduction of 7.2 dB was obtained at error sensor 1 between 200 and 400 Hz.



a) **Results obtained with conventional speakers**

b) **Results obtained with advanced speakers**

Fig. 5.18: Response at error sensors

Microphones were positioned throughout the cabin of the automobile in order to monitor the effect of the control away from the error sensors. Three microphones were located in the back of the car, each of them at the position of the head of a passenger. Noise reduction of about 2 dB was measured in the 100 to 500 Hz frequency band. More microphones were positioned in the front, closer to the error sensors. While noise reduction of 6 dB was measured at ear level close to the front door, increases in sound level were measured at ear height in the center of the automobile and close to the windshield. Due to the positioning of the monitoring microphones it is not possible to estimate the extent of the zone of quiet around the error sensors. A more dense microphone array would have been required. Nevertheless, it can be concluded that the radius of the zone of quiet is less than thirty centimeters because no noise reduction over 2 dB was measured thirty centimeters away from the error sensors. Although the control obtained is very localized, no significant spillover was measured away ofthe sensors.

The experiment presented in this section showed that active control of simulated road noise is possible using two carefully positioned reference sensors. These results agree with those of the principal component analysis, where it was stated that good control was achievable with a number of reference sensors as low as two.

Best results were obtained with conventional speakers. The piezoelectric sources performed well at frequencies above 200 Hz. The reduction obtained with the latter sources was low compared to the reduction obtained with the conventional speakers, because most of the energy of the signal from the error sensors was between 100 and 150 Hz. When the car is driven on a road, assuming that the disturbances caused by each wheel are not correlated, the controller would require eight reference sensors. However, if it is assumed that the noise measured in front of the car is mainly due to the front wheels, then noise control can be achieved at the position of the driver and front passenger with only four reference sensors.

5.3.2.2.4 Conclusions

In chapter 5.3.2.2 the active control of both power train and simulated road noise inside a sport utility vehicle with standard passive treatment is described. Experimental results obtained with premium conventional speakers have been compared to results obtained with new compact lightweight piezoelectric based acoustic sources. First, experimental active control of power train and road noise was implemented in the cabin under various test conditions. It was shown that active control of power train noise was feasible with two conventional Speakers, two error sensors and three reference sensors from 40 to 500 Hz. Attenuation of 6.5 dB was obtained at the error sensors and a zone of quiet was created around the heads of the driver and front passenger. The zone of quiet was approximately a sphere of 45-cm radius (in which the attenuation is greater than 3 dB). It was also shown that active control of simulated road noise (one wheel excited by a roller covered with rough aggregate) was feasible between 100 and 500 Hz with two reference sensors, two error sensors and two conventional Speakers as control sources. Attenuation of up to 12 dB was recorded at one of the error sensors for this test case. Second, it was shown that the control of power train noise was feasible with the piezoelectric based speakers above 150 Hz. Due to the low output of these sources at frequencies below 200 Hz, it was not possible to control the engine firing frequency harmonics below 150 Hz. These sources were also used for the control of simulated road noise. No control of the sound pressure was obtained at the error sensors below 200 Hz. Attenuation of 7 dB was measured at the error sensors between 200 and 400 Hz. It is predicted that good control of real road noise would require 8 reference sensors.

5.3.3 Some commercial approaches of the car industry

5.3.3.1 System by Siemens VDO in the air intake

Siemens VDO Automotive AG (The companies Siemens Automotive and Mannesmann VDO have come together to form a partnership.) has developed a technology that allows the engine sound and ist source to be actively influenced. With an ANC installed in the air intake system.

The field tests conducted by Siemens VDO in the USA have shown that absolute silence is not always what is wanted. The motorist wishes for quiet only at night or when driving in the rain, in other words in driving situations where particular concentration is required. When driving cross-country during the day, harmonious sequences of tones are particularly preferred. In town an engine sound is considered to be exceptionally good when certain harmonics are significantly emphasized during acceleration phases.

Siemens VDO's patented concept for the technical realization of Active Noise Control impresses with its simple construction. The sound level is continuously monitored by a microphone on the air intake side of the engine. On the basis of this measurement and a driving situation-related setpoint, the ANC controller calculates a corresponding tone that is generated by a loudspeaker in the intake system. Thanks to the proximity of the loudspeaker to the source of the sound, the power consumption of the system can be limited to just a few watts.

Due to the fact that individual frequencies can be suppressed and others emphasized, ANC enables the automobile manufacturer to design the engine sound more freely. Selective sound design can be used to match the sound behavior of an engine - within the limits of the law - to the character of a car model.

Today passive filters are employed to reduce the noise in the air intake system whose air resistance hinders the natural air intake of the engine. Because the performance of an engine is directly dependent on the intaken air volume, ANC serves not only to enhance comfort but also to increase the rated output of the engine.

Siemens VDO, supplier of complete intake modules integrating intake manifold, fuel supply system and engine controller, is continuing to develop the system in Windsor, Ontario (Canada). Siemens VDO aims to be ready for the start of series production by the year 2005.

5.3.3.2 ANC as standard equipment in Honda cars

Honda Motor has equipped its "Accord Wagon" with an active noise control system that reduces the noise of driving inside the car for better audibility of music. This system, developed jointly with Matsushita Electric Industrial, comes as a standard equipment of the model Accord Wagon released on June 2000.

This system is capable of reducing low frequency noise of around 40kHz produced inside the car when driving on rough roads. There is a microphone under the driver's seat to detect noise from the road surface and, by radiating a sound of the opposite phase from the audio speaker, it becomes possible to cut the noise level. In 1991, Nissan Motor installed on its "Bluebird" a system using similar technologies to reduce the engine noise. Honda's system is the first to reduce noise from the road surface. In addition, the noise control system was an option on Bluebird, whereas it comes pre-installed on Accord Wagon. Use of analog filters instead of digital filters has cut costs for standard installation. The company plans to install its noise control system on other cars as well in the future.

5.4 Industrial applications

5.4.1 Active exhaust silencer [10]

Active noise control has proven effective in the reduction of low frequency noise in ducts, e.g. in air conditioning systems. In many other applications the sound reducing capability of active silencers is limited by the harsh environment. In fact, higher temperatures, static and dynamic pressure loads, extreme sound power levels and abrasive flow often prohibit the use of active components at all. To turn the active technology in these cases to practical use it is necessary either to use resistant but costly components or to protect the sensitive components of the active system from rough influences. One possible way of protection is to remove the active parts physically from the duct thereby avoiding its direct contact to the gas flow. Hereby, the acoustic connection is maintained by a pipe which builds up a quarter-wave-resonator with its well-known acoustic characteristic. The insertion loss (IL) of such a rigidly terminated side branch resonator can be altered and improved significantly if the end of the resonator is terminated by an active silencer cassette (ASC).

The active exhaust silencer, shown in Figure 5.19, forms itself around the circular main duct and consists of two parts. At the bottom a porous absorber layer surrounds the main duct half. At the top side of the main duct there is a tube attached via a small opening. To protect the side branch mechanically and to avoid an exchange of gas from the main duct with that of the side branch the opening is covered by perforated sheet metal (4) and fibre fabric (3). Additionally, the opening is sealed hermetically by heat-resistant foil (2).

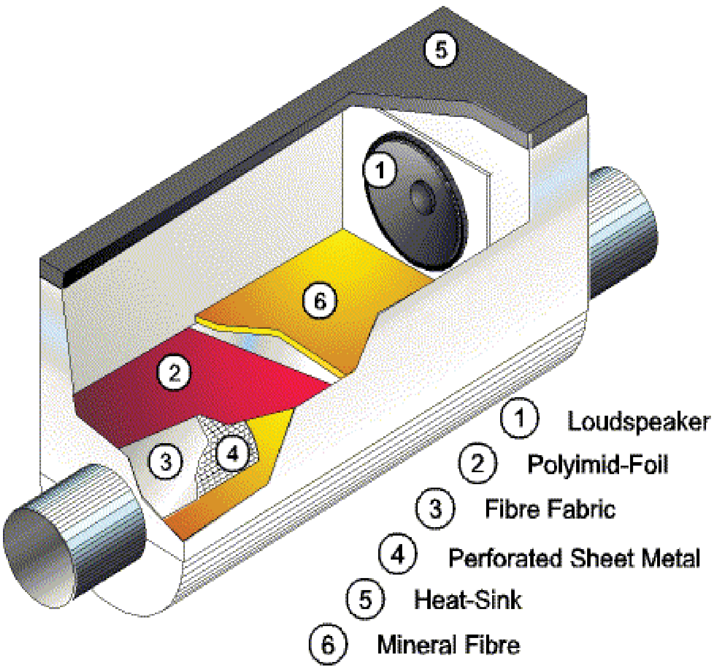


Fig. 5.19: Active exhaust silencer

In the side a porous absorber layer (5) is attached over the whole length which serves at one hand as heat insulation and at the other hand as an acoustic absorber for the interior of the resonator. The end of the side branch is terminated by a small box containing a standard loudspeaker (1), an electret microphone and an analogue controller which forms an active silencer cassett. With these active components an electro-acoustic feedback loop is established. The feedback gain can be manually or automatically adjusted at the controller and determines within the stability limits the acoustic wall impedance of the active silencer cassett.

Figure 5.20 shows the results of the insertion loss measurements of the silencer in Figure 5.19. The black curve describes the case where a plate of sheet metal rigidly terminates the end of the silencer branch as in conventional passive side resonators. This configuration forms a combination of Helmholtz- and Quarter-Wavelength-Resonator with a first insertion loss maximum of 17 dB at about 160 Hz. Additionally, the insertion loss increases with rising frequency due to the porous layer at the bottom half-pipe.

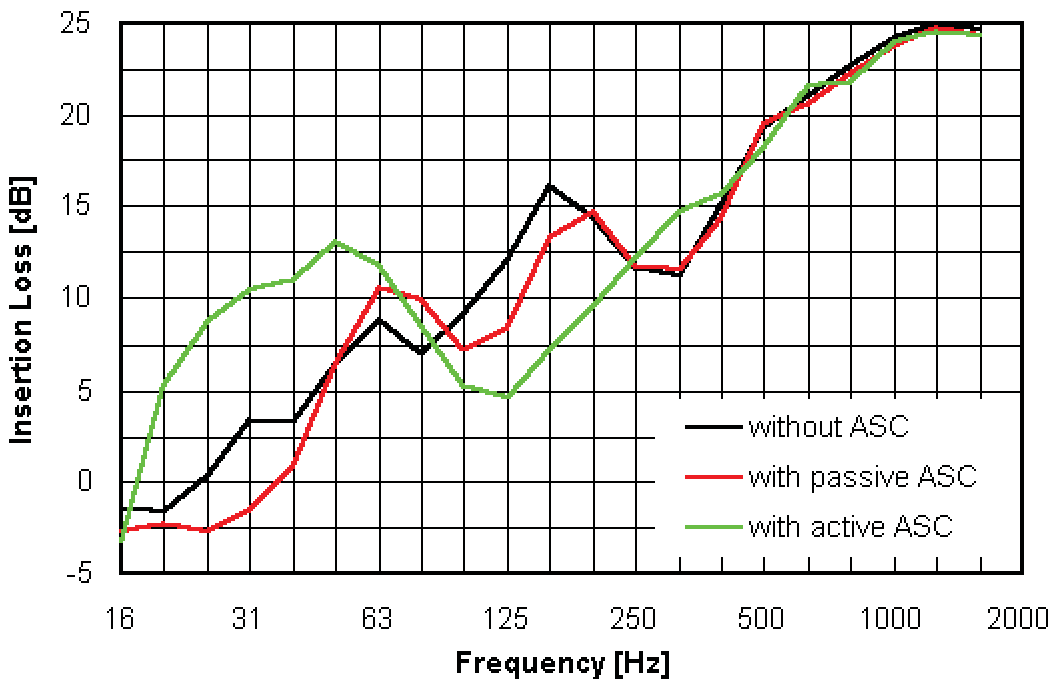


Fig. 5.20 : Measured insertion loss of the active exhaust silencer

The measured insertion loss changes little if an active cassette without feedback is assembled at the end of the side branch instead of the rigid termination (red curve). If the feedback loop is closed with the highest possible feedback gain the insertion loss maximum becomes broader and is shifted to about 50 Hz (green curve). With a feedback gain between the maximum and zero the low-frequency-maximum in the insertion loss will be located somewhere between 160 Hz and 50 Hz. It is possible to tune the insertion loss maximum from about 160 Hz to about 50 Hz simply by adjusting the feedback gain. This opens up the opportunity for a manual or an automatic control of the insertion loss depending on operational states of a noise emitting device as e.g. the rotational speed of an internal combustion engine or the temperature of a burners exhaust gas.

5.4.2 ANC in a large exhaust stack [17]

5.4.2.1 Introduction

Practical issues associated with the installation of an active noise control (ANC) system in an 80m high exhaust stack, containing a hot, wet and dirty air flow, are discussed in the following chapter. The noise problem to be controlled was a 165Hz tone generated by the fan at the bottom of the stack and radiated by the stack into the surrounding community from which complaints were received on a regular basis, sometimes from residents living more than 1km away.

The stack was split into three parallel axial sections in the vicinity of the ANC system to ensure that only plane waves were present (Figure 5.14). The use of the ANC system resulted in reductions of the tonal noise of 10dB inside one section of the duct and 20dB inside the other two sections. It is expected that similar reductions in the tonal noise would be measured in the community. The result of the installation of the axial splitters which divided the duct into 3 sections such that the acoustic path length difference was 1/3 of a wavelength between adjacent sections.

The project described here was undertaken to reduce the 165Hz tonal noise radiated by an 80m high exhaust stack to the surrounding community. A large, 4m diameter centrifugal fan at the base of the exhaust stack forces air into the stack and generates tonal noise at the blade pass frequency (BPF). The fan had 10 blades and rotated with a speed of about 993 rpm under normal plant operating conditions.

The stack is characterised by a normal operating temperature of 100°C which rises to 180°C at times. The exhaust from the fan consists of very moist and abrasive clay dust which sticks to non-vertical surfaces, forming a thick sludge. Previous attempts at the installation of a passive muffler had not been successful on a long term basis, partly because of the abrasive and sticky nature of the exhaust flow. With community complaints increasing, those responsible for the stack were left with the option of installing a passive muffler (of uncertain life) at a cost of over 200,000€ plus considerable plant downtime and inconvenience or giving active noise control a try. The system design and performance corresponding to the latter option is described here. The design specification was a noise reduction of 10 dB to 15 dB in the community at the frequency of the blade pass tone which varied from 152 Hz to 166 Hz, depending on plant operation conditions.

The project began by installing two axial splitters in a section of duct about 2 metres upstream from the fan so that the duct was divided into 3 parallel axial sections (Figure 5.21). This required waiting until a scheduled plant shut down, then removing the duct insulation.

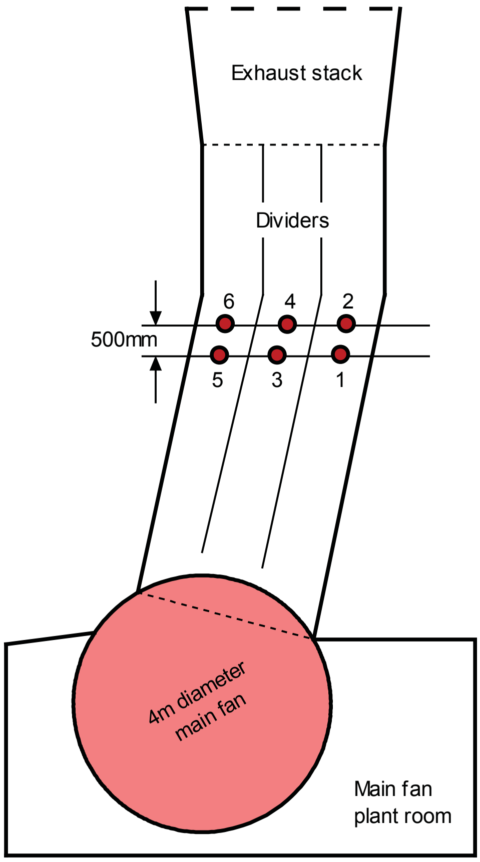


Fig. 5.21: Main fan, microphone and loudspeaker locations, numbers represent loudspeaker locations

The reason for splitting the duct was to ensure that only plane waves were propagating in the vicinity of the ANC system. A system without splitters would have resulted in the propagation of higher order acoustic modes in the duct and would have required considerably more effort to develop a suitable ANC system. Thus the splitters were installed to ensure that no duct section dimension exceeded 0.5 wavelengths (about 1.23m). An added benefit of the splitters was the possibility of additional noise reduction due to passive cancellation as a result of the sound path differences between adjacent sections of the duct being fortuitously close to one third of a wavelength (at the blade pass frequency). At 100°C, the speed of sound is 388m/s. Assuming an air flow speed in the duct of 20m/s, the effective speed of sound is 408m/s and the wavelength at 165Hz is 2.47m. One third of this is 0.82m which is the path length difference between adjacent duct sections. The passive cancellation effect was not expected to replace the ANC system because the passive system would not be able to track temperature, fan speed and flow speed variations, all of which affect the effective wavelength of sound at the blade passage frequency.

The design of feedforward active noise control systems may be divided into two separate tasks; the design of the physical system and the design of the electronic control system as shown in Figure 5.22.

The physical system consists of:

- the **fan** which is responsible for the unwanted tonal the noise,
- the **exhaust stack** along which the noise propagates,
- a **tachometer** for measuring the fan rotational speed,
- **microphones** which measure the noise inside the exhaust stack,
- **loudspeakers** which provide additional sound to "cancel" the BPF.

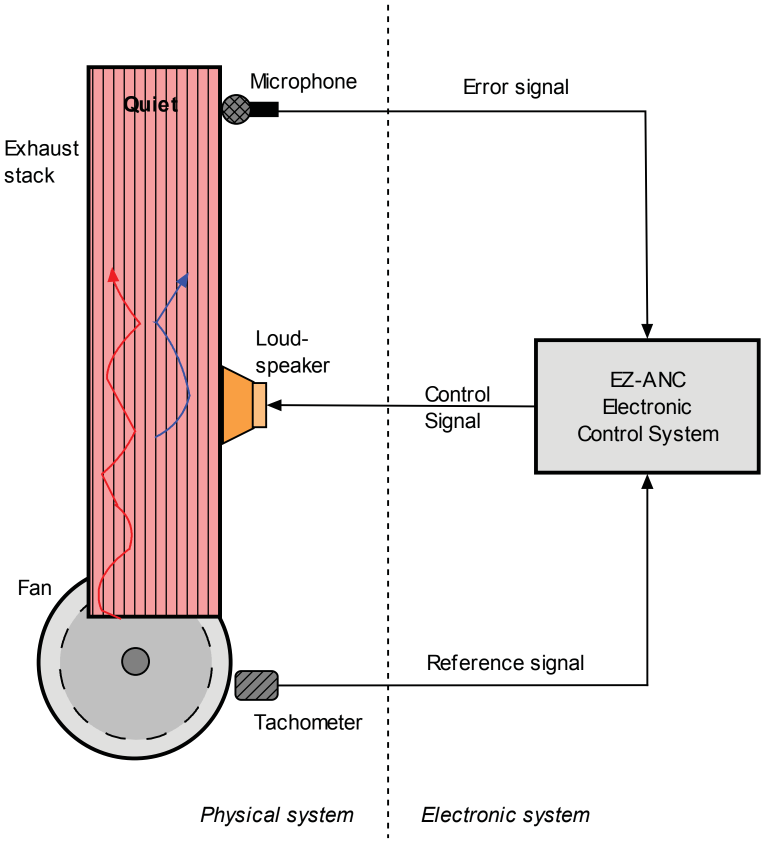


Fig. 5.22: Basic components of the active noise control system.

The physical system includes transducers (loudspeakers and microphones) which convert physical properties into electrical signals. These electrical signals are used by the electronic system which comprises:

- a **digital controller** which generates an appropriate control signal based on the microphone and tachometer signals,
- **power amplifiers** (not shown) which amplify the control signal for the loudspeakers,
- **instrumentation filters and amplifiers** (not shown) which are used to condition the electrical signal so they are suitable for the digital controller.

5.4.2.2 Physical system design

Before discussing the design of the components of the physical system, it is worth mentioning the factors that limit the performance of the overall system and the importance of each factor. The first factor which limits control system performance is the location of the control sources. Once these locations have been optimised, the error sensor locations will determine the maximum achievable noise reduction. The next factor is the quality of the reference signal. If this is contaminated with frequency components which need not be controlled, then the achievable control of the components which do require control will be reduced. In fact, in an ideal situation, the relative strength of each frequency component in the reference signal should reflect the relative desired amount of reduction of each of the components in the error signal. Perhaps the hierarchical nature of active control can best be understood with Figure 5.23.

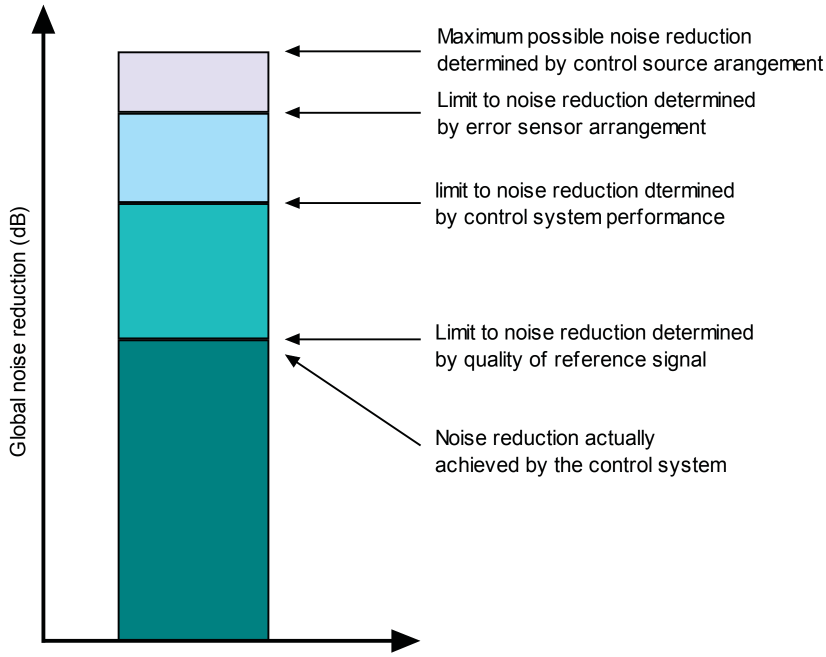


Fig. 5.23: Performance hierarchy for active noise control

5.4.2.2.1 Control source location

The location of the control source in terms of wavelengths of separation between the control source and effective location of the offending noise source is crucial to the success of the installation, in terms of how hard the loudspeaker will have to be driven and the maximum amount of control achievable. The optimum locations can be calculated exactly for a constant pressure source or a constant volume velocity source.

Unfortunately, the worst locations for a constant volume velocity source are the best locations for a constant pressure source and most industrial noise sources are somewhere between these two ideal cases The ideal location requires some trial and error testing to asure a high degree of certainty. As a centrifugal fan is close in nature to a constant pressure source, this idealisation will be used here for illustrative purposes. Another complicating factor is the effective impedance of the primary source in terms of the phase and amplitude of the reflection of upstream propagating acoustic waves. This also affects the optimum control source location.

For the case under consideration, the duct temperature varies from 100°C to 180°C and for a flow speed of 20m/s, this corresponds to a wavelength variation from 2.47m to 2.72m. To cover all possibilities it is clear that three speakers in each duct section separated axially by about 0.45m will be needed. For the trial installation, two speakers on opposite sides of each duct section and separated axially by 0.55m were used.

5.4.2.2.2 Control source equipment

Power amplifiers supplied the control signals from the electronic controller to the control loudspeakers to generate the cancelling acoustic signals in the three sections of the exhaust stack. The loudspeaker cones were sprayed with lacquer to prevent deterioration in the moist environment. The loudspeakers were rated at 250W and were mid-range speakers (1m).

The enclosures housing the loudspeakers have provision for cooling air flow through the backing cavity and purging air flow in the front of the loudspeaker cone and then into the duct to keep the speaker as clean as possible. The experimantal enclosure design is illustrated in Figure 5.24.

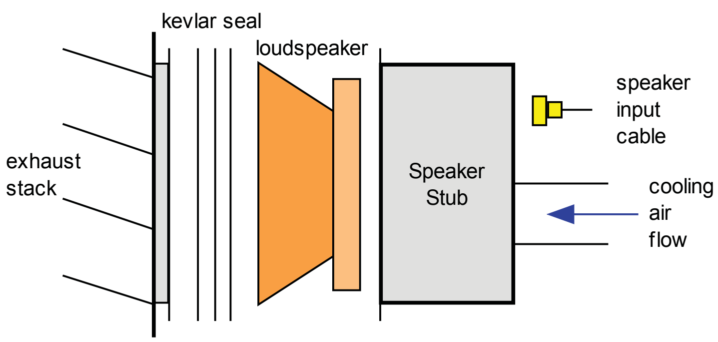


Fig. 5.24: Loudspeaker enclosure configuration.

Very early in the trials, loudspeakers were failing on a regular basis, even though they were not being driven at more than half their maximum rating. The loudspeaker enclosures are heated by conduction and radiation from the duct walls and by convection from the hot air in the duct. When the plant operating condition is such that the duct temperature rises to 180°C, the diaphragm which supports the loudspeaker cone becomes soft and easily distorts. This has two effects: Firstly, when the cone distorts, the voice coil attached to the cone rubs on the loudspeaker magnet. The rubbing removes the insulation on the coil and then an electrical "short circuit" occurs which destroys the voice coil. The rubbing also occurs when no control signal is applied to the loudspeaker because the high noise levels in the duct result in a significant movement of the loudspeaker cone. This voice coil failure occurred with several loudspeakers and caused significant delays in the progress of the project. Secondly, when the diaphragm is soft, the loudspeaker cone is pulled into the exhaust stack by the suction effect generated by the large air flow up the stack. When the air temperature returns to normal operating temperature, the diaphragm is permanently displaced towards the end of the cone's traverse. This damage reduces the efficiency of the loudspeaker in converting electrical power into sound power. Thus the original design of the loudspeaker enclosure shown in Figure 5.24 was modified to include cooling air flow through the backing cavity. It is anticipated that a chilled water jacket will be needed around the loudspeaker enclosures and loudspeaker driving magnets if the loudspeakers are to have a reasonable life.

5.4.2.2.3 Error sensor location

The error sensor locations were placed by the following constraints:

- At least one was necessary for each duct section.
- They had to be as far away from the control sources as possible to minimise the effect of the near field on the overall control system performance.
- They had to be well below the top of the splitters (500mm) to minimise contamination from sound propagating in adjacent duct sections.
- They should not be near a node in the standing wave in the duct section caused by reflection from the end of the splitter. The 500mm criterion also satisfied this criterion.

5.4.2.2.4 Error signal equipment

Error microphones were mounted in sets of three in a microphone stub as shown in Figure 5.25. The stub includes a filter cover to protect the microphone from airborne contaminants, a microphone holder which holds three microphones and an air-line coupling which forces cooling air over the microphones.

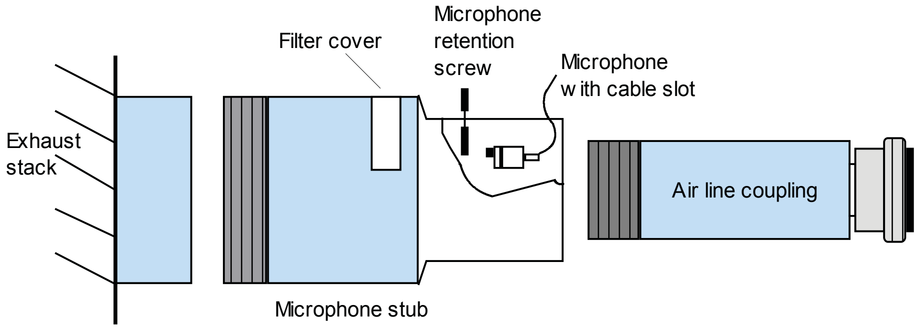


Fig. 5.25: Microphone stub assembly

The three microphones in each stub are connected to a microphone preamplifier and a summation circuit mounted in the ANC System Control Box, where the signals are combined into a single 'error signal' for each microphone stub. In this way, the error sensing system is triple-redundant. The ANC system will still function upon failure of up to two of the microphones in each stub.

A typical frequency spectrum of the noise level measured by an error microphone in the duct section closest to the control box is shown in Figure 5.26. It can be seen that the noise level at the BPF (165 Hz) is approximately 120 dB (linear). This signal varies from 116 dB up to 130 dB, depending on the location of the microphone along the duct axis.

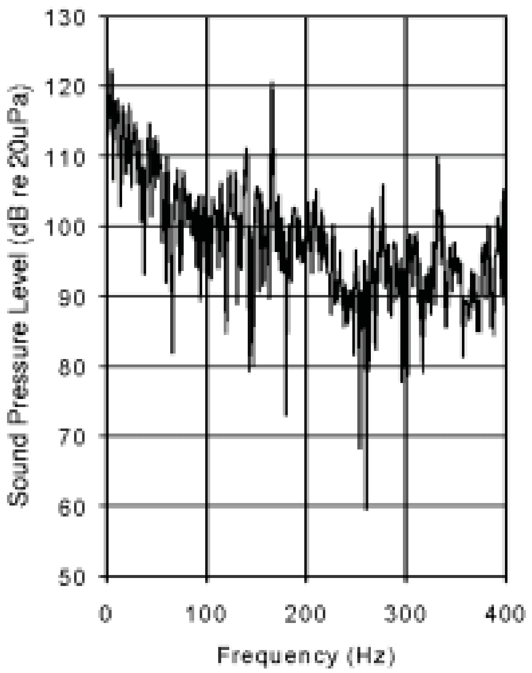


Fig. 5.26: Sound pressure level in the exhaust stack, measured in the duct section closest to the control box

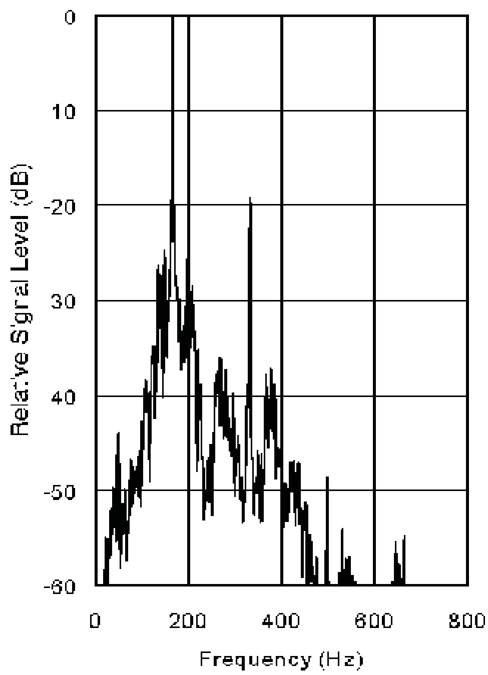


Fig. 5.27: Typical filtered error signal frequency spectrum

The electrical signal from the microphone passes through an analog band pass filter to reduce the signal levels at frequencies other than at the BPF. The reason for filtering the error microphone signal is to provide the digital controller with a measure of the noise at the BPF and to ignore noise at other frequencies. A typical frequency spectrum of the filtered error signal is shown in Figure 5.27. The peak in the spectrum at the blade pass frequency (165Hz) is clearly evident and the second harmonic (330 Hz) is nearly 20 dB less. The amount by which the tonal peak exceeds the background noise varies by about ±5 dB and the background noise, including that due to the air cooling, is always at least 25 dB below the peak. The delay through the error sensor filter has no measurable effect on the performance of the control system.

The cooling air which is used to reduce the temperature of the microphones has been obtained from the plant compressed air line. Oil in the compressed air line has entirely covered all the microphones and they are all functioning after several months of use.

5.4.2.2.5 Reference signal considerations

In the described study were two choices for the reference signal: a tachometer on the fan shaft or reference microphone upstream of the control sources. The advantage of the tachometer is that it is relatively straightforward to implement and is much more reliable. The disadvantage is that only frequencies corresponding to the fan blade pass frequency and its harmonics will be derived for control by the electronic controller. Although the noise causing the problem did appear to be tonal in nature, it did not appear as a sharp spectral peak, probably as a result of slight speed variations of the fan. If the lack of sharpness of the peak were a result of noise being generated by an instability phenomenon, then the signal from the tachometer would be an unsuitable reference signal.

One advantage of using a microphone reference signal is that the noise generating mechanism is unimportant and the dominant part of the spectrum will be controlled regardless. The disadvantage is that the microphone signal will be contaminated with fluid pressure fluctuations which propagate at the speed of flow and not the speed of sound. Also any filtering of the reference signal to remove the unwanted signals is likely to result in unacceptable delays through the filter with the result that the controller is unlikely to receive the reference signal in time to generate the required control signal.

5.4.2.2.6 Reference signal equipment

The tachometer system is used to provide a reference signal to the digital controller. It consists of a digital inductive pickup mounted close to the notched shaft encoder disk. The notched encoder disc has the same number of evenly spaced notches as there are blades on the fan.

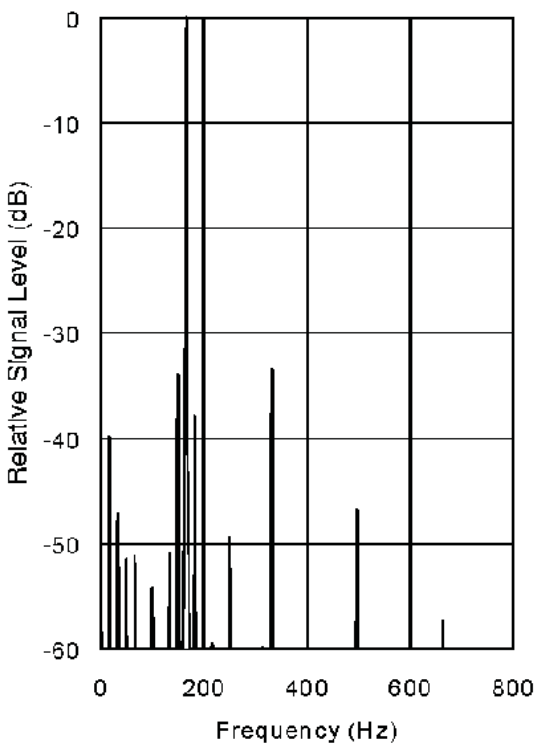


Fig. 5.28: Reference signal frequency spectrum

The pickup head supplies a square wave signal of frequency equal to the blade pass frequency to the tachometer amplifier in the ANC System Control Box which incorporates a power supply and signal conditioner. The digital signal from the pickup is converted into a sinusoidal reference signal at the same frequency as the noise produced by the fan, by filtering out all multiples of the BPF with a low pass filter.

A typical frequency spectrum of the filtered tachometer signal is shown in Figure 5.28. The BPF can be seen clearly at 165Hz. The first harmonic and side bands are approximately 35dB lower; thus resulting in a high quality reference signal being provided to the digital controller.

5.4.2.2.7 Electronic controller

The electronic control system used to process the incoming tachometer and microphone signals was the Causal System's EZ-ANC.

It was necessary to measure the transfer function between the output to the loudspeakers and the input from the error microphones on-line on a continuous basis to maintain algorithm stability. The algorithm was used to adapt the weights of an FIR filter which simulated this transfer function and which was easily incorporated in the control algorithm. It was found that best results were obtained when the controller was configured as three 2-channel systems, as shown in Figure 5.29. Each system had one error signal input and two control outputs and operated on one of the three duct sections. A significant amount of leakage was used in the control algorithm to even out the driving signals to the two loudspeakers in each duct section. This was necessary to prevent the loudspeaker in the poorer location in the duct (from the control viewpoint) from being over driven.

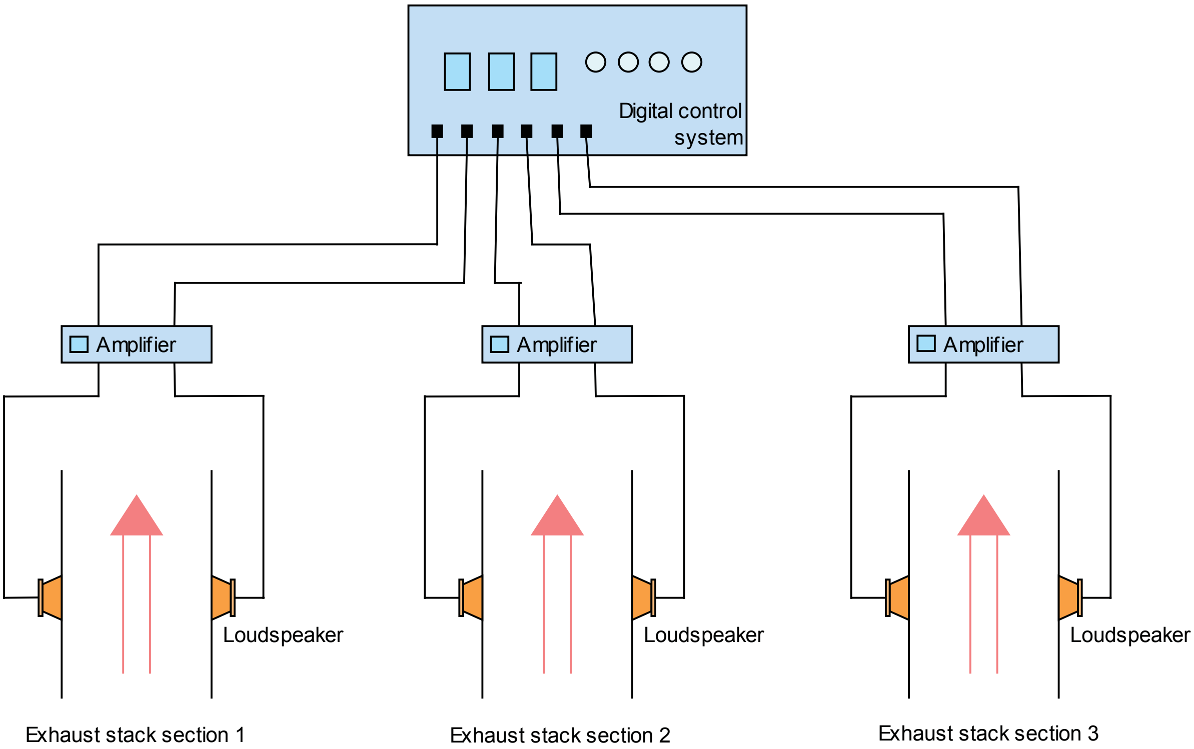


Fig. 5.29: Control system configuration

To prevent the digitised error signal voltage becoming too large, which results in the digital controller becoming unstable, the value of the input gain for the error signal is adjusted so that the error signal is at approximately half of the maximum allowed value to allow for possible fluctuations.

5.4.2.3 Active noise control trials

The ANC system described in the preceding sections has been used to reduce the noise levels in the exhaust stack. Trials have demonstrated the effectiveness of an ANC solution to the noise problem. The active noise control trials involved using the system described in the preceding sections, to simultaneously reduce the noise levels in each of the three sections of the exhaust stack. Figures 5.30 to 5.32 show the spectrum of the filtered error microphone voltage in each of the three sections in the exhaust stack, with no active control and when active control is used. The duct section numbers are identified in Figure 5.21, where duct section 1 corresponds to loudspeakers 1 and 2, duct section 2 corresponds to loudspeakers 3 and 4 and duct section 3 corresponds to loudspeakers 5 and 6.

The differences between the two spectra shown in each figure are directly comparable to the expected reductions in the in-duct sound pressure levels at the BPF. Figure 5.30 shows the poorest result of all three sections. Only 10dB reduction was possible at the error microphone. Figure 5.30 and 5.31 show noise level reductions of about 20 dB. The reason for the poor result in duct section 1 is due to the non-optimum placement of the loudspeaker enclosures in the axial direction along the duct.

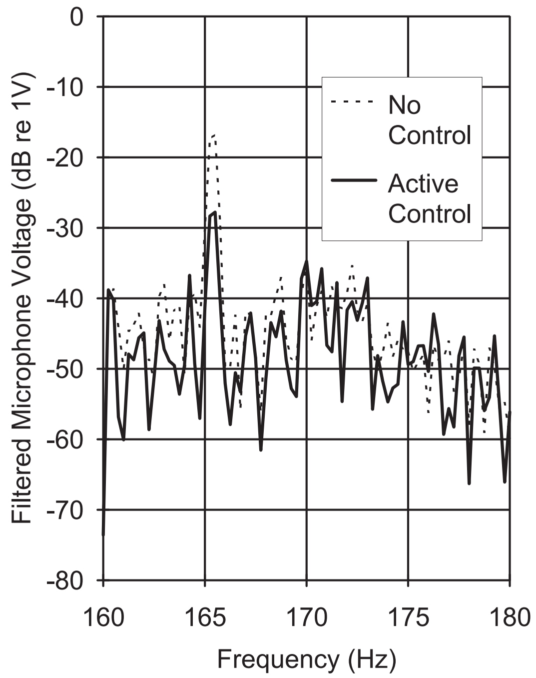


Fig. 5.30: Filtered error microphone voltage in duct closest to control box (duct section 1).

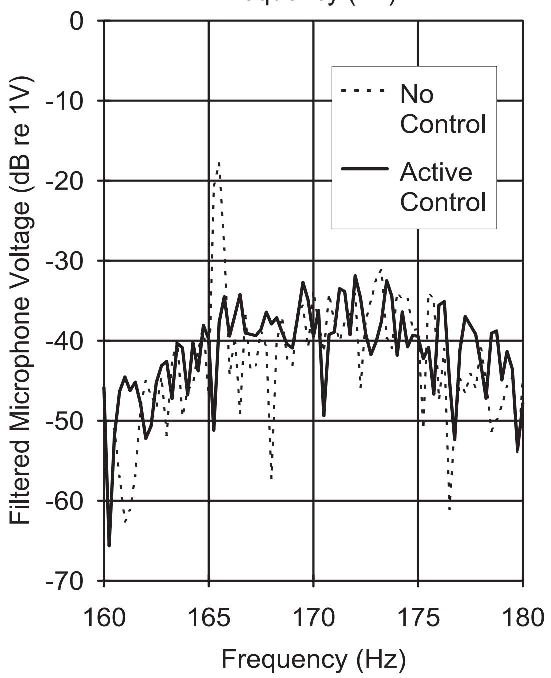


Fig. 5.31: Filtered error microphone voltage in the middle section (duct section 2).

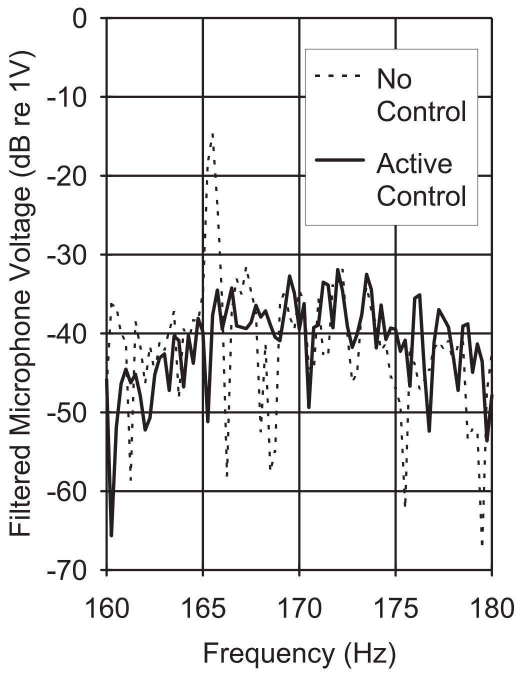


Fig. 5.32: Filtered error microphone voltage in duct (duct section 3).

The required loudspeaker outputs to achieve the control shown in the figures are listed in Table 5.3.

Table 5.3 Loudspeaker voltage across terminals during active control trials.

Speaker number	Duct section number	Voltage (Vrms)	Power (Watts)
1	1	33	140
2	1	26	85
3	2	29	105
4	2	23	65
5	3	33	140
6	3	21	55

The loudspeakers are numbered as shown in Figure 5.21 with the numbered loudspeakers located inside the liquid starter room (one in each of the three sections) and with the even numbered speakers outside on the opposite side of the duct (with number 2 opposite number 1, etc).

System additions. The following two extensions to the current system are desirable to provide remote control and monitoring of the ANC system performance.

Time averaged Sound Pressure Levels in the stack. Filtered outputs from the microphones can be rectified then low pass filtered to provide calibrated DC levels (4-20mA) which can be integrated with the current plant control system. This will enable the continual observation and tracking of the sound levels in the stack. Such information could be correlated with other process information to identify potential process conditions where noise emitted by the stack is at a minimum. In addition, in the unlikely event that the system becomes unstable, the operator could be alerted.

A remote reset for the EZ-ANC control system. This would allow remote reset of the control system from the process control room should the controller become unstable.

5.4.2.4 Conclusions

In spite of the harsh environmental conditions, an active noise control system has been shown to be effective in attenuating the noise generated by a large fan at the blade pass frequency and radiated by an 80m high exhaust stack. In-duct reductions of up to 20dB were obtained after special treatment of the loudspeakers and microphones. The installation of axial splitters in the duct to cut out cross modes for the active noise control system had the added benefit of providing some passive cancellation due to the propagation path in each of the three sections being one third of a wavelength different to that in other sections.

6 Summary

Noise is usually still reduced by passive means like damping plates and sound absorbing materials. All these passive means show only good attenuation results for high frequencies and have several disadvantages. In the last decades, the active control of sound and vibration has emerged as a viable technology to fill the low-frequency technology gap. „Active“ in this case means the cancelling of unwanted noise by producing the same noise with an 180 degree phase-shift. This phenomenon is called destructive interference (chapter 4.3.2) with the result of quiet.

The first patent on an ANC (Active Noise Control) system got the German Lueg in 1936. He was the first scientist who experimented with acoustic waves and constructive or destructive interference, which cause intensification and weakening of the sound field. The accelerating interest in ANC began in the late 1960s with several scientific publications. A general trend since the late 80s is now seen towards the application of digital signal processing techniques to the active control of noise. The current advance in digital technology makes the implementation of digital controlers easier in use and cheaper.

Today the most important applications of active noise attenuation are exhaust mufflers for internal combustion engines, headphones for aircraft use, aircraft cabin noise reduction, ambulances, dishwashers, vibration isolators for engines, vibration isolation to protect electronics from vibration and shock, furnaces and boiler. From the environmental point of view the actively quietning systems in cars, transformers and exhaust stacks are described in detail (chapter 5).

The evaluation of the effect of error sensing strategies on the performance of an active system to control tonal noise radiation from a small transformer is described in chapter 5.2. The results represent the best that could be expected using an ideal feedforward (chapter 4.3.3) active control system. It is shown that the sound reduction was mainly achieved by the modal rearrangement control mechanism.

As far as road noise in cars is concerned (chapter 5.3) there remains much work to be done to investigate the use of ANC in automobiles with typical damping where there is increased modal overlap and reduced acoustic resonances. Nevertheless there are some good examples of state of the art technology like the Siemens VDO system (5.3.3.1) and ANC as standard equipment in Honda cars (chapter 5.3.3.2).

In chapter 5.4 it is shown that the active attenuation of noise generated by an exhaust pipe as well as a large industrial fan in a 80m high exhaust stack is possible by active means.

For future work researchers should develop better integrated active control solutions for complex noise problems (industrial noise) and more accurate active automobile audio systems.

7 Glossary

7.1 Dictionary

Intensity of Three-Dimensional Waves

A two-dimensional sound wave looks like a series of concentric circles that get bigger as they move further away from their origin. These circles are called wavefronts. In real life, sound waves grow in three dimensions. Three-dimensional waves move out in all directions away from their origin in wavefronts that are concentric spherical surfaces. The space in between wavefronts is the wavelength. Rays indicate the motion of a set of wavefronts. Rays are lines perpendicular to the wavefronts that originate at the source of the sound and follow the wavefronts outward. If the sound is emitted evenly in all directions, the energy at a distance *r* from the source will be uniform on the spherical shell. If we let *P* equal the original power the sound has when emitted from the source, the intensity per unit area (the surface area of a sphere is the denominator) at a distance *r* from the source will be: $I = P/4\pi r^2$

The intensity level of sound is measured in decibels (dB). Decibels are units of intensity that are based upon a logarithmic scale. This means that a sound with an intensity of 20 dB is ten times as loud as one with an intensity of 10 dB, 30 dB is ten times as intense as 20 dB, and so on. The reason for this logarithmic scale is that humans hear intensity on a similar logarithmic scale. So, while a 20 dB sound is ten times as intense as a 10 dB sound, we perceive it as only twice as loud. The hearing threshold (level at which humans begin to perceive sound) is 0 dB. When a sound reaches upwards of 120 dB, it is above the threshold of pain (point at which most people begin feeling pain). Everything in between can be heard by a human with normal hearing.

Source	**Decibels**	**Description**
	0	Hearing Threshold
Normal breathing	10	Barely Audible
Rustling Leaves	20	
Soft Whisper	30	Very Quiet
Library	40	
Quiet Office	50	Quiet
Conversation	60	
Busy Traffic	70	
Average Factory	80	
Niagara Falls	90	Constant Hearing
Train	100	Endangers Hearing
Construction Noise	110	
Rock Concert	120	Pain Threshold
Machine Gun	130	
Jet Takeoff	150	

But, these levels are not constants. What a human perceives as loud or soft depends on the frequency as well as the intensity of the sound. The graph below displays intensity levels compared with the frequencies for sounds of equal loudness for humans. The bottom line is the threshold of hearing. At a 1 kHz frequency, the hearing threshold is 0 dB, but at 60 Hz the decibel level is 50. Only one percent of all human beings can hear sounds this low, so, the lower line is mainly for those with very good hearing. The next line up is the hearing threshold for the majority of people. The top line is the pain threshold. Other than at one point, about 4 kHz, this line varies little. All of the other lines also dip down at 4 kHz. We can gather from this graph, then, that the human ear is most sensitive at about 4 kHz.

Speed of Sound

Sound travels at different speeds depending on what it is traveling through. Of the three mediums (gas, liquid, and solid) sound waves travel the slowest through gases, faster through liquids, and fastest through solids. Temperature also effects the speed of sound.

Gases: When we look at the properties of a gas, we see that only when molecules collide with each other can the condensations and rarefactions of a sound wave move about. So it makes sense that the speed of sound has the same order of magnitude as the average molecular speed between collisions. In a gas, it is particularly important to know the temperature. This is because at higher temperatures, molecules collide more often, giving the sound wave more chances to move around rapidly. At freezing (0º Celcius), sound travels through air at 331 meters per second. But, at 20ºC, room temperature, sound travels at 343 meters per second.

Liquids: Sound travels faster in liquids than in gases because molecules are more tightly packed. In fresh water, sound waves travel at 1482 meters per second. That's well over 4 times faster than in air. Several ocean-dwelling animals rely upon sound waves to communicate with other animals and to locate food and obstacles. The reason that they are able to effectively use this method of communication over long distances is that sound travels so much faster in water.

Solids: Sound travels fastest through solids. This is because molecules in a solid medium are much closer together than those in a liquid or gas, allowing sound waves to travel more quickly through it. In fact, sound waves travel over 17 times faster through steel than through air. But, this is only for the majority of solids. The speed of sound in all solids are not faster than in all liquids.

Substancs	Temperature (°C)	Speed (m/s)
Gases		
Carbon Dioxide	0	259
Oxygen	0	316
Air	0	331
Air	20	343
Helium	0	965
Liquids		
Chloroform	20	1004
Ethanol	20	1162
Mercury	20	1450
Water	20	1482
Solids		
Lead	-	1980
Copper	-	5010
Glass	-	5640
Steal	-	5960

Superposition

The first, and most important, difference between wave motion and the motion of objects is that waves do not display any repulsion of each other analogous to the normal forces between objects that come in contact. Two wavepatterns can therefore overlap in the region of space (imagine two waves crossing each other in a fluid medium). Where the two waves coincide, they add together. For instance, when two waves have a 3cm crest above the normal water level, their combination makes a 6cm crest. On the other hand, if two depressions cross, they make an extra deep 6cm trough. A 3cm crest and a 3cm trough result in a height of zero, i.e. the waves cancel each other out at that point. This additive rule is referred to as the 'principle of superposition'.

Superposition can occur not just with sinusoidal waves but with waves of any shape. Sound waves obey superposition. Sounds do not knock out sounds out of the way when they collide, and we can hear more than one sound at once if they both reach our ear simultaneously. Experiments to date have not shown any deviation from the principle of superposition in the case of light waves. For other types of waves, it is typically a very good approximation for low-energy waves.

Wavelength and Period

The wavelength is the horizontal distance between any two successive equivalent points on the wave. That means that the wavelength is the horizontal length of one cycle of the wave. The period of a wave is the time required for one complete cycle of the wave to pass by a point. So, the period is the amount of time it takes for a wave to travel a distance of one wavelength.

7.2 Internet links

7.2.1 Research

- Centre for Computer systems Architecture, Halmstad University, (Sweden)
- Centre for Acoustics and Vibration , Penn State University,(USA)
- Institute of Sound and Vibration Research, University of Southampton,(UK)
- Mechanical Engineering , Trinity College, Dublin, (EIRE)
- University of Utah, (USA)
- Signal Processing Division, University of Strathclyde, Glasgow, (Scottland)
- Department of Signal Processing, University of Karlskrona Ronneby, (Sweden)
- Acoustics and Signal Processing Institute for Micro structural Sciences National Research Council of Canada Bldg, Ottawa (Canada)
- Laboratoire de Mécanique et d'Acoustique, Marseille, (France)
- VIRGINIA TECH. Mechanical Engineering Department. (USA)
- Departement of Mechanical Engineering ,University of Adeleide, (Australia)
- Production Engineering, Machine Design and Automation, K.U.Leuven (Belgium)
- Fraunhofer institut Stuttgart (Germany)
- Delft University of Technology, Delft, Netherlands http://www.tudelft.nl/home.html
- Duke University, Durham, North Carolina, USA, Professor Robert Clark, Director, Center for Applied Control, 919-660-5435, rclark@egr.duke.edu, http://www.duke.edu/
- Georgia Institute of Technology, Atlanta, Georgia, USA http://www.gatech.edu/

- Norwegian University of Science and Technology, Trondheim, Norway http://www.ntnu.no/ Massachusettes Institute of Technology, Cambridge, Massachusetts, USA http://web.mit.edu/ Northern Illinois University, DeKalb, Illinois, USA http://www.niu.edu/
- Old Dominion University, Norfolk, Virginia, USA http://www.odu.edu/
- Pennsylvania State University: The Graduate Program in Acoustics, Penn State University, PO Box 30, State College, PA 16804, Phone (814) 865-6364, Fax (814) 865-3119 http://www.acs.psu.edu
- Purdue University, West Lafayette, Indiana, USA http://www.purdue.edu/
- RWTH Aachen, Germany, http://www.itm.rwth-aachen.de/
- Southampton University, Southampton, England http://www.soton.ac.uk/
- Technical University of Denmark, Denmark http://www.dtu.dk/dtu/dtu.html
- Technical University of Berlin, Germany http://www.tk.tu-berlin.de/
- Germany Technical University of Munich, Germany
- Germany University of Adelaide, Adelaide, South Australia, Australia, http://www.mecheng.adelaide.edu.au/anvc/
- University of Auckland, New Zealand, Prof Marshall, h.marshall@auckland.ac.nz, Dr. George Dodd g.dodd@auckland.ac.nz
- University of Goettingen
- Germany University of Hamburg
- Germany University of Karlskrona/Ronneby, Ronneby, Sweden http://www.hk-r.se/isb/isb_e.html
- University of London/Royal Holloway, Egham, Surrey, England http://www.ph.rhbnc.ac.uk/
- University of Salford, England
- Universite de Sherbrooke, Sherbrooke, Quebec, Canada http://www.usherb.ca/index.html
- Universite de Technologie de Compiegne, Compiegne, France http://www.univ-compiegne.fr/

- University of Utah, Salt Lake City, Utah, USA http://www.utah.edu/HTML_Docs/Campus_Info.html

- Villanova University, Philadelphia, Pennsylvania, USA http://www.vill.edu/

- Virginia Polytechnic Institute & State University, Blacksburg, Virginia, USA http://www.vt.edu/

- Blekinge Institute of Technology, http://www.bth.se

7.2.2 Companies

- ABS GmbH, Gesellschaft für Automatisierung, Bildverarbeitung und Software mbH, Wildenbruchstraße 15, 07745 Jena, Germany, phone +49 3641 615258, fax +49 3641 675410, email abs@gtc.ibu.de

- ANR Headsets, 2955 Fawn Drive, Burlington, KY 41005, mailto://mmorgan@anr-headsets.com, http://www.anr-headsets.com/

- Active Vibration Control Instrumentation, PCB Piezotronics, Inc., 3425 Walden Ave. Depew, NY 14043-2495, phone 716-684-0001

- BBN Physical Systems & Technologies, 70 Fawcett St, Cambrdige, MA 02138, phone 617-873-3533, fax 617-873-2918, email fberkman@bbn.com (Dr. E. Frank Berkman, Technical Director)

- Causal Systems Pty Ltd., P.O. Box 100, Rundle Mall, South Australia 5000, Australia, phone 61.8.303.5460, fax 61.8.303.4367, e-mail chansen@aelmg.adelaide.edu.au (Colin Hansen), http://www.causal.on.net/

- Digisonix, Inc., 8401 Murphy Drive, Middleton, WI 53562-2243 USA, phone 608.836.3999, fax 608.836.5583, email: information@digisonix.com, Web http://www.mailbag.com/users/dgsnx_mr/

- dSPACE Inc., 22260 Haggerty Road, Suite 120, Northville, MI 48167, phone 810.344.0096, fax 810.344.2060, email 75371.36@compuserve.com, Web http://www.ti.com (search for DSPACE once there)

- Headsets, Inc., 2330-B Lakeview, Amarillo, Texas 79109, USA, phone 806.358.6336, fax 806.358.6449, Paige Brittain, President

- Noise Cancellation Technologies, Inc., Headquarters: Stamford, Connecticut, 203.961.0500 (Joanna Lipper). Engineering facilities: Linthicum, Maryland, USA, 410.636.8700, http://www.nct-active.com/

- Peltor Inc., 41 Commercial Way, E. Providence, RI 02914, Phone: 401.438.4800, Fax: 401.434.1708

- Walker Electronic Silencing, Inc. (formerly Walker Noise Cancellation Technologies), http://www.WESilence.com .

- Sennheiser Electronic Corporation & Neumann USA, 6 Vista Drive, P.O. Box 987, Old Lyme, CT 06371, TEL 860.434.9190, FAX: 860.434.1759, info@sennheiserusa.com, http://www.sennheiserusa.com/

- Sennheiser electronic GmbH & Co. KG, D-30900 Wedemark, 106005.55@Compuserve.com, http://www.sennheiser.com/

- Sennheiser (Canada) Inc., 221 Labrosse Avenue, Pointe-Claire, Quebec, Canada, H9R 1A3, TEL: 514.426.3013/800.463.1006, FAX: .514.426.3953/800.463.3013, info@sennheiser.ca, http://www.sennheiser.ca/

- Technofirst, Parc Technologique et Industriel de Napollon, 399, Avenue des Templiers, 13676 Aubagne Cedex France, Tel(33) 4 42 03 46 60, Fax (33) 4 42 03 06 43, technof@technofirst.com, http://www.technofirst.com

7.2.3 More noisy links

- Österreichischer Arbeitsring für Lärmbekämpfung (ÖAL): www.oeal.at

- Homepage of the European Acoustics Association: www.eaa-fenestra.de/

- Zeitschrift für Lärmbekämpfung: www.technikwissen.de/laerm/aktuell/news.asp

- German Acoustical Society (Deutsche Gesellschaft für Akustik): www.dega.itap.de

- Firmenkatalog Schallschutz: www.fluglaerm.de/bvf/links6.htm

- Deutscher Arbeitsring für Lärmbekämpfung: http://www.dalaerm.de

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[5] *Understandintg Active Noise Cancellation*, Colin H. Hansen, E & FN Spon, 2001

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[15] *Active control of sound radiation from a small transformer using near field sensing*, X. Li – X. Qui – C. Hansen, Department of mechanical Engineering, Adelaide University, Australia

[16] *Active Control of Power Train and Road Noise in the Cabin of a Sport Utility Vehicle with Advanced Speakers*, J. Couche – C. Fuller, Dept. of Mechanical Engineering, Virginia Polytechnic Institute and State University, 1999

[17] *Practical Implementation of an Active Noise Control System in a hot exaust stack,* Colin H. Hansen - Carl Q. Howard - Kym A. Burgemeister - Ben S. Cazzolato, University of Adelaide, South Australia

[18] *Development of a Compact Lightweight Acoustic Source for the Control of Automobile Interior Noise,* J. Couche – C. Fuller – S. Booth, Proceedings of Internoise'99

[19] *Determination ofthe Number of Input Transducers required for Active Control ofRoad Noise Inside Automobiles*, C. Heatwole - X. Dian - R. Bernhard, Proceedings of Noise-Conference 93

[20] *Active Control of Noise radiated by Structures using near field sensing*, X. Li – X. Qui – C. Hansen, International Journal of Acoustics and Vibration, 2001

7.4 Figures

7.5 Tables