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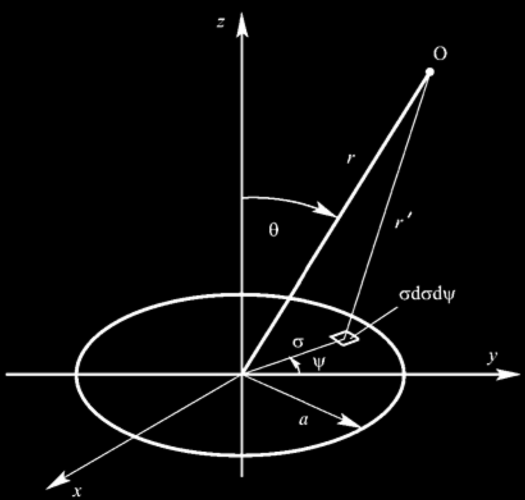
ENGINEERING NOISE CONTROL



THEORY AND PRACTICE

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Spon Press

CHAPTER ONE

Fundamentals and Basic Terminology

LEARNING OBJECTIVES

In this chapter the reader is introduced to:

- fundamentals and basic terminology of noise control;
- noise-control strategies for new and existing facilities;
- the most effective noise-control solutions;
- the wave equation;
- plane and spherical waves;
- sound intensity;
- units of measurement;
- the concept of sound level;
- frequency analysis and sound spectra;
- adding and subtracting sound levels;
- three kinds of impedance; and
- flow resistance.

1.1 INTRODUCTION

The recognition of noise as a source of annoyance began in antiquity, but the relationship, sometimes subtle, that may exist between noise and money seems to be a development of more recent times. For example, the manager of a large wind tunnel once told one of the authors that in the evening he liked to hear, from the back porch of his home, the steady hum of his machine 2 km away, for to him the hum meant money. However, to his neighbours it meant only annoyance and he eventually had to do without his evening pleasure.

The conflicts of interest associated with noise that arise from the staging of rock concerts and motor races, or from the operation of airports, are well known. In such cases the relationship between noise and money is not at all subtle. Clearly, as noise may be the desired end or an inconsequential by-product of the desired end for one group, and the bane of another, a need for its control exists. Each group can have what it wants only to the extent that control is possible.

The recognition of noise as a serious health hazard is a development of modern times. With modern industry has come noise-induced deafness; amplified music also takes its toll. While amplified music may give pleasure to many, the excessive noise of much modern industry probably gives pleasure to very few, or none at all. However, the relationship between noise and money still exists and cannot be ignored. If paying people, through compensation payments, to go deaf is little more expensive than implementing industrial noise control, then the incentive definitely exists to do nothing, and hope that decision is not questioned.

A common method of noise control is a barrier or enclosure and in some cases this may be the only practical solution. However, experience has shown that noise control at the design stage is generally accomplished at about one-tenth of the cost of adding a barrier or an enclosure to an existing installation. At the design stage the noise producing mechanism may be selected for least noise and again experience suggests that the quieter process often results in a better machine overall. These unexpected advantages then provide the economic incentive for implementation, and noise control becomes an incidental benefit. Unfortunately, in most industries engineers are seldom in the position of being able to make fundamental design changes to noisy equipment. They must often make do with what they are supplied, and learn to apply effective “add-on” noise-control technology. Such “add-on” measures often prove cumbersome in use and experience has shown that quite often “add-on” controls are quietly sabotaged by employees who experience little benefit and find them an impediment to their work.

In the following text, the chapters have been arranged to follow a natural progression, leading the reader from the basic fundamentals of acoustics through to advanced methods of noise control. However, each chapter has been written to stand alone, so that those with some training in noise control or acoustics can use the text as a ready reference. The emphasis is upon sufficient precision of noise-control design to provide effectiveness at minimum cost, and means of anticipating and avoiding possible noise problems in new facilities.

Simplification has been avoided so as not to obscure the basic physics of a problem and possibly mislead the reader. Where simplifications are necessary, their consequences are brought to the reader’s attention. Discussion of complex problems has also not been avoided for the sake of simplicity of presentation. Where the discussion is complex, as with diffraction around buildings or with ground-plane reflection, results of calculations, which are sufficient for engineering estimates, are provided. In many cases, procedures also are provided to enable serious readers to carry out the calculations for themselves. For those who wish to avoid tedious calculations, there is a software package, ENC, available that follows this text very closely. See www.causalsystems.com.

In writing the equations that appear throughout the text, a consistent set of symbols is used: these symbols are defined following their use in each chapter. Where convenient, the equations are expressed in dimensionless form; otherwise SI units are implied unless explicitly stated otherwise.

To apply noise-control technology successfully, it is necessary to have a basic understanding of the physical principles of acoustics and how these may be applied to the reduction of excessive noise. Chapter 1 has been written with the aim of providing the basic principles of acoustics in sufficient detail to enable the reader to understand the applications in the rest of the book.

Chapter 2 is concerned with the ear, as it is the ear and the way that it responds to sound, which generally determines the need for noise control and criteria for acceptable minimum levels. The aim of Chapter 2 is to aid in understanding criteria for acceptability, which are the subject of Chapter 4. Chapter 3 is devoted to instrumentation, data collection and data reduction. In summary, Chapters 1 to 4 have been written with the aim of providing the reader with the means to quantify a noise problem.

Chapter 5 has been written with the aim of providing the reader with the basis for identifying noise sources and estimating noise levels in the surrounding environment, while Chapter 6 provides the means for rank ordering sources in terms of emitted sound power. It is to be noted that the content of Chapters 5 and 6 may be used in either a predictive mode for new proposed facilities or products or in an analytical mode for analysis of existing facilities or products to identify and rank order noise sources.

Chapter 7 concerns sound in enclosed spaces and provides means for designing acoustic treatments and for determining their effectiveness. Chapter 8 includes methods for calculating the sound transmission loss of partitions and the design of enclosures, while Chapter 9 is concerned with the design of dissipative and reactive mufflers.

Chapter 10 is about vibration isolation and control, and also gives attention to the problem of determining when vibration damping will be effective in the control of emitted noise and when it will be ineffective. The reader's attention is drawn to the fact that less vibration does not necessarily mean less noise, especially since vibration damping is generally expensive.

Chapter 11 provides means for the prediction of noise radiated by many common noise sources and is largely empirical, but is generally guided by considerations such as those of Chapter 5.

Chapter 12 is concerned with numerical acoustics and its application to the solution of complex sound radiation problems and interior noise problems.

1.2 NOISE CONTROL STRATEGIES

Possible strategies for noise control are always more numerous for new facilities and products than for existing facilities and products. Consequently, it is always more cost effective to implement noise control at the design stage than to wait for complaints about a finished facility or product.

In existing facilities, controls may be required in response to specific complaints from within the work place or from the surrounding community, and excessive noise levels may be quantified by suitable measurements. In proposed new facilities, possible complaints must be anticipated, and expected excessive noise levels must be estimated by some procedure. Often it is not possible to eliminate unwanted noise entirely and more often to do so is very expensive; thus minimum acceptable levels of noise must be formulated, and these levels constitute the criteria for acceptability.

Criteria for acceptability are generally established with reference to appropriate regulations for the work place and community. In addition, for community noise it is advisable that at worst, any facility should not increase background (or ambient) noise levels in a community by more than 5 dB(A) over existing levels without the facility, irrespective of what local regulations may allow. Note that this 5 dB(A) increase applies to broadband noise and that clearly distinguishable tones (single frequencies) are less acceptable.

When dealing with community complaints (predicted or observed) it is wise to be conservative; that is, to aim for adequate control for the worst case, noting that community noise levels may vary greatly (±10 dB) about the mean as a result of atmospheric conditions (wind and temperature gradients and turbulence). It is worth careful note that complainants tend to be more conscious of a noise after making a complaint and thus subconsciously tend to listen for it. Thus, even after considerable noise reduction may have been achieved and regulations satisfied, complaints may continue. Clearly, it is better to avoid complaints in the first place and thus yet another argument supporting the assertion of cost effectiveness in the design stage is provided.

In both existing and proposed new facilities and products an important part of the process is to identify noise sources and to rank order them in terms of contributions to excessive noise. When the requirements for noise control have been quantified, and sources identified and ranked, it is possible to consider various options for control and finally to determine the cost effectiveness of the various options. As was mentioned earlier, the cost of enclosing a noise source is generally much greater than modifying the source or process producing the noise. Thus an argument, based upon cost effectiveness, is provided for extending the process of source identification to specific sources on a particular item of equipment and rank ordering these contributions to the limits of practicality.

Community noise level predictions and calculations of the effects of noise control are generally carried out in octave frequency bands. Current models for prediction are not sufficiently accurate to allow finer frequency resolution and less fine frequency resolution does not allow proper account of frequency-dependent effects. Generally, octave band analysis provides a satisfactory compromise between too much and too little detail. Where greater spectrum detail is required, one-third octave band analysis is often sufficient.

If complaints arise from the work place, then regulations should be satisfied, but to minimise hearing damage compensation claims, the goal of any noise-control program should be to reach a level of no more than 80 dB(A). Criteria for other situations in the work place are discussed in Chapter 4. Measurements and calculations are generally carried out in standardised octave or one-third octave bands, but particular care must be given to the identification of any tones that may be present, as these must be treated separately.

More details on noise control measures can be found in the remainder of this text and also in ISO 11690/2 (1996).

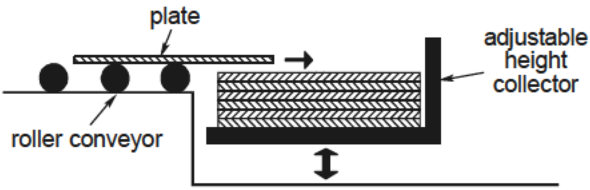
Any noise problem may be described in terms of a sound source, a transmission path and a receiver, and noise control may take the form of altering any one or all of these elements. When considered in terms of cost effectiveness and acceptability, experience puts modification of the source well ahead of either modification of the transmission path or the receiver. On the other hand, in existing facilities the last two may be the only feasible options.

1.2.1 Sound Source Modification

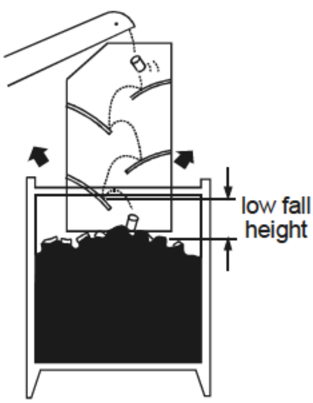
Modification of the energy source to reduce the noise generated often provides the best means of noise control. For example, where impacts are involved, as in punch presses, any reduction of the peak impact force (even at the expense of the force acting over a longer time period) will dramatically reduce the noise generated. Generally, when a choice between various mechanical processes is possible to accomplish a given task, the best choice, from the point of view of minimum noise, will be the process that minimises the time rate of change of force or jerk (time rate of change of acceleration). Alternatively, when the process is aerodynamic a similar principle applies; that is, the process that minimises pressure gradients will produce minimum noise. In general, whether a process is mechanical or fluid mechanical, minimum rate of change of force is associated with minimum noise.

Mechanical shock between solids should be minimised; for example, impact noise may be generated by parts falling into metal bins and the height that the parts fall could be reduced by using an adjustable height collector (see Figure 1.1a) or the collector could be lined with conveyor belt material. Alternatively the collector could have rubber flaps installed to break the fall of the parts (see Figure 1.1b).

The control of noise at its source may require maintenance, substitution of materials, substitution of equipment or parts of equipment, specification of quiet equipment, substitution of processes, substitution of mechanical power generation and



(a)



(b)

Figure 1.1 Impact noise reduction: (a) variable height collector; (b) interrupted fall.

transmission equipment, change of work methods, reduction of vibration of large structures such as plates, beams, etc. or reduction of noise resulting from fluid flow.

Maintenance includes balancing moving parts, replacement or adjustment of worn or loose parts, modifying parts to prevent rattles and ringing, lubrication of moving parts and use of properly shaped and sharpened cutting tools.

Substitution of materials includes replacing metal with plastic, a good example being the replacement of steel sprockets in chain drives with sprockets made from flexible polyamide plastics.

Substitution of equipment includes use of electric tools rather than pneumatic tools (e.g. hand tools), use of stepped dies rather than single-operation dies, use of rotating shears rather than square shears, use of hydraulic rather than mechanical presses, use of presses rather than hammers and use of belt conveyors rather than roller conveyors.

Substitution of parts of equipment includes modification of gear teeth, by replacing spur gears with helical gears – generally resulting in 10 dB of noise reduction, replacement of straight edged cutters with spiral cutters (for example, in wood working machines a 10 dB(A) reduction may be achieved), replacement of gear drives with belt drives, replacement of metal gears with plastic gears (beware of additional maintenance problems) and replacement of steel or solid wheels with pneumatic tyres.

Substitution of processes includes using mechanical ejectors rather than pneumatic ejectors, hot rather than cold working, pressing rather than rolling or forging, welding or squeeze rivetting rather than impact rivetting, use of cutting fluid in machining processes, changing from impact action (e.g. hammering a metal bar) to progressive pressure action (e.g. bending a metal bar with pliers), replacement of circular saw blades with damped blades and replacement of mechanical limit stops with micro-switches.

Substitution of mechanical power generation and transmission equipment includes use of electric motors rather than internal combustion engines or gas turbines, or the use of belts or hydraulic power transmissions rather than gear boxes.

Change of work methods includes replacing ball machines with selective demolition in building demolition, replacing pneumatic tools by changing manufacturing methods, such as moulding holes in concrete rather than cutting after production of the concrete component, use of remote control of noisy equipment such as pneumatic tools, separating noisy workers in time, but keeping noisy operations in the same area, separating noisy operations from non-noisy processes. Changing work methods may also involve selecting the slowest machine speed appropriate for a job (selecting large, slow machines rather than smaller, faster ones), minimising the width of tools in contact with the workpiece (2 dB(A) reduction for each halving of tool width) and minimising protruding parts of cutting tools.

Reductions of noise resulting from the resonant vibration of structures (plates, beams, etc.) may be achieved by ensuring that machine rotational speeds do not coincide with resonance frequencies of the supporting structure, and if they do, in some cases it is possible to change the stiffness or mass of the supporting structure to change its resonance frequencies (increasing stiffness increases resonance frequencies and increasing the mass reduces resonance frequencies). In large structures, such as a roof or ceiling, attempts to change low order resonance frequencies by adding mass or stiffness may not be practical.

Another means for reducing sound radiation due to structural vibration involves reducing the acoustic radiation efficiency of the vibrating surface. Examples are the replacement of a solid panel or machine guard with a woven mesh or perforated panel or the use of narrower belt drives. Damping a panel can be effective (see Section 10.7) if it is excited mechanically, but note that if the panel is excited by an acoustic field, damping will have little or no effect upon its sound radiation. Blocking the transmission of vibration along a noise radiating structure by the placement of a heavy mass on the structure close to the original source of the noise can also be effective.

Reduction of noise resulting from fluid flow may involve providing machines with adequate cooling fins so that noisy fans are no longer needed, using centrifugal rather than propeller fans, locating fans in smooth, undisturbed air flow, using fan blades designed using computational fluid dynamics software to minimise turbulence, using large low speed fans rather than smaller faster ones, minimising the velocity of fluid flow and maximising the cross-section of fluid streams. Fluid flow noise reduction may also involve reducing the pressure drop across any one component in a fluid flow system, minimising fluid turbulence where possible (e.g. avoiding obstructions in the flow), choosing quiet pumps in hydraulic systems, choosing quiet nozzles for compressed air systems (see Figure 11.4), isolating pipes carrying the fluid from support structures, using flexible connectors in pipe systems to control energy travelling in the fluid as well as the pipe wall and using flexible fabric sections in low pressure air ducts (near the noise source such as a fan).

Another form of source control is to provide machines with adequate cooling fins so that noisy fans are no longer needed. In hydraulic systems the choice of pumps, and in compressed air systems the choice of nozzles, is important.

Other alternatives include minimising the number of noisy machines running at any one time, relocating noisy equipment to less sensitive areas or if community noise is a problem, avoiding running noisy machines at night.

1.2.2 Control of the Transmission Path

In considering control of the noise path from the source to the receiver some or all of the following treatments need to be considered: barriers (walls), partial enclosures or full equipment enclosures, local enclosures for noisy components on a machine, reactive or dissipative mufflers (the former for low frequency noise or small exhausts, the latter for high frequencies or large diameter exhaust outlets), lined ducts or lined plenum chambers for air-handling systems, vibration isolation of machines from noise-radiating structures, vibration absorbers and dampers, active noise control and the addition of sound-absorbing material to reverberant spaces to reduce reflected noise fields.

1.2.3 Modification of the Receiver

In some cases, when all else fails, it may be necessary to apply noise control to the receiver of the excessive noise. This type of control may involve use of ear-muffs, ear-plugs or other forms of hearing protection; the enclosure of personnel if this is practical; moving personnel further from the noise sources; rotating personnel to reduce noise exposure time; and education and emphasis on public relations for both in-plant and community noise problems.

Clearly, in the context of treatment of the noise receiver, the latter action is all that would be effective for a community noise problem, although sometimes it may be less expensive to purchase complainants' houses, even at prices well above market value.

1.2.4 Existing Facilities

In existing facilities or products, quantification of the noise problem requires identification of the noise source or sources, determination of the transmission paths from the sources to the receivers, rank ordering of the various contributors to the problem and finally, determination of acceptable solutions.

To begin, noise levels must be determined at potentially sensitive locations or at locations from which the complaints arise. For community noise, these measurements may not be straightforward, as such noise may be strongly affected by variable weather conditions and measurements over a representative time period may be required. This is usually done using remote data logging equipment in addition to periodic manual measurements.

The next step is to apply acceptable noise level criteria to each location and thus determine the required noise reductions, generally as a function of octave or one-third octave frequency bands (see Section 1.10). Noise level criteria are usually set by regulations and appropriate standards.

Next, the transmission paths by which the noise reaches the place of complaint are determined. For some cases this step is often obvious. However, cases may occasionally arise when this step may present some difficulty, but it may be very important in helping to identify the source of a complaint.

Having identified the possible transmission paths, the next step is to identify (understand) the noise generation mechanism or mechanisms, as noise control at the source always gives the best solution. Where the problem is one of occupational noise, this task is often straightforward. However, where the problem originates from complaints about a product or from the surrounding community, this task may prove difficult. Often noise sources are either vibrating surfaces or unsteady fluid flow (air, gas or steam). The latter aerodynamic sources are often associated with exhausts. In most cases, it is worthwhile determining the source of the energy that is causing the structure or the aerodynamic source to radiate sound, as control may best start there. For a product, considerable ingenuity may be required to determine the nature and solution to the problem. In existing facilities and products, altering the noise generating mechanism may range from too expensive to acceptable and should always be considered as a means for possible control.

For airborne noise sources, it is important to determine the sound power and directivity of each to determine their relative contributions to the noise problem. The radiated sound power and directivity of sources can be determined by reference to the equipment manufacturer's data, reference to Chapter 11, or by measurement, using methods outlined in Chapters 5 and 6. The sound power should be characterised in octave or one-third octave frequency bands (see Section 1.10) and dominant single frequencies should be identified. Any background noise contaminating the sound power measurements must be taken into account (see Section 1.11.5).

Having identified the noise sources and determined their radiated sound power levels, the next task is to determine the relative contribution of each noise source to the noise level at each location where the measured noise levels are considered to be excessive. For a facility involving just a few noise sources this is a relatively straightforward task. However, for a facility involving tens or hundreds of noise sources, the task of rank ordering can be intimidating, especially when the locations of complaint are in the surrounding community. In the latter case, the effect of the ground terrain and surface, air absorption and the influence of atmospheric conditions must also be taken into account, as well as the decrease in sound level with distance due to the "spreading out" of the sound waves.

Commercial computer software is available to assist with the calculation of the contributions of noise sources to sound levels at sensitive locations in the community or in the work place. Alternatively, one may write one's own software (see Chapter 5). In either case, for an existing facility, measured noise levels can be compared with predicted levels to validate the calculations. Once the computer model is validated, it is then a simple matter to investigate various options for control and their cost effectiveness.

In summary, a noise control program for an existing facility includes:

- undertaking an assessment of the current environment where there appears to be a problem, including the preparation of noise level contours where required;
- establishment of the noise control objectives or criteria to be met;
- identification of noise transmission paths and generation mechanisms;
- rank ordering noise sources contributing to any excessive levels;
- formulating a noise control program and implementation schedule;
- carrying out the program; and
- verifying the achievement of the objectives of the program.

More detail on noise control strategies for existing facilities can be found in ISO 11690/1 (1996).

1.2.5 Facilities in the Design Stage

In new facilities and products, quantification of the noise problem at the design stage may range from simple to difficult. At the design stage the problems are the same as for existing facilities and products; they are identification of the source or sources, determination of the transmission paths of the noise from the sources to the receivers, rank ordering of the various contributors to the problem and finally determination of acceptable solutions. Most importantly, at the design stage the options for noise control are generally many and may include rejection of the proposed design. Consideration of the possible need for noise control in the design stage has the very great advantage that an opportunity is provided to choose a process or processes that may avoid or greatly reduce the need for noise control. Experience suggests that processes chosen because they make less noise, often have the additional advantage of being more efficient.

The first step for new facilities is to determine the noise criteria (see Chapter 4) for sensitive locations, which may typically include areas of the surrounding residential community that will be closest to the planned facility, locations along the boundary of the land owned by the industrial company responsible for the new facility, and within the facility at locations of operators of noisy machinery. Again, care must be taken to be conservative where surrounding communities are concerned so that initial complaints are avoided.

In consideration of possible community noise problems following establishment of acceptable noise criteria at sensitive locations, the next step may be to develop a computer model or to use an existing commercial software package to estimate expected noise levels (in octave frequency bands) at the sensitive locations, based on machinery sound power level and directivity information (the latter may not always be available), and outdoor sound propagation prediction procedures. Previous experience or the local weather bureau can provide expected ranges in atmospheric weather conditions (wind and temperature gradients and turbulence levels) so that a likely range and worst case sound levels can be predicted for each community location. When directivity information is not available, it is generally assumed that the source radiates uniformly in all directions.

If the estimated noise levels at any sensitive location exceed the established criteria, then the equipment contributing most to the excess levels should be targeted for noise control, which could take the form of:

- specifying lower equipment sound power levels, or sound pressure levels at the operator position, to the equipment manufacturer;
- including noise-control fixtures (mufflers, barriers, enclosures, or factory walls with a higher sound transmission loss) in the factory design; or
- rearrangement and careful planning of buildings and equipment within them.

Sufficient noise control should be specified to leave no doubt that the noise criteria will be met at *every* sensitive location. Saving money at this stage is not cost effective. If predicting equipment sound power levels with sufficient accuracy proves difficult, it may be helpful to make measurements on a similar existing facility or product.

More detail on noise control strategies and noise prediction for facilities at the design stage can be found in ISO 11690/3 (1997).

1.2.6 Airborne versus Structureborne Nois e

Very often in existing facilities it is relatively straightforward to track down the original source(s) of the noise, but it can sometimes be difficult to determine how the noise propagates from its source to a receiver. A classic example of this type of problem is associated with noise on board ships. When excessive noise (usually associated with the ship's engines) is experienced in a cabin close to the engine room (or in some cases far from the engine room), or on the deck above the engine room, it is necessary to determine how the noise propagates from the engine. If the problem arises from airborne noise passing through the deck or bulkheads, then a solution may include one or more of the following: enclosing the engine, adding sound-absorbing material to the engine room, increasing the sound transmission loss of the deck or bulkhead by using double wall constructions or replacing the engine exhaust muffler.

On the other hand, if the noise problem is caused by the engine exciting the hull into vibration through its mounts or through other rigid connections between the engine and the hull (for example, bolting the muffler to the engine and hull), then an entirely different approach would be required. In this latter case it would be the mechanically excited deck, hull and bulkhead vibrations which would be responsible for the unwanted noise. The solution would be to vibration isolate the engine (perhaps through a well-constructed floating platform) or any items such as mufflers from the surrounding structure. In some cases, standard engine vibration isolation mounts designed especially for a marine environment can be used.

As both types of control are expensive, it is important to determine conclusively and in advance the sound transmission path. The simplest way to do this is to measure the noise levels in octave frequency bands at a number of locations in the engine room with the engine running, and also at locations in the ship where the noise is excessive. Then the engine should be shut down and a loudspeaker placed in the engine room and driven so that it produces noise levels in the engine room sufficiently high for them to be readily detected at the locations where noise reduction is required.

Usually an octave band filter is used with the speaker so that only noise in the octave band of interest at any one time is generated. This aids both in generating sufficient level and in detection. The noise level data measured throughout the ship with just the loudspeaker operating should be increased by the difference between the engine room levels with the speaker as source and with the engine as source, to give corrected levels for comparison with levels measured with the engine running. In many cases, it will be necessary for the loudspeaker system to produce noise of similar level to that produced by the engine to ensure that measurements made elsewhere on the ship are above the background noise. In some cases, this may be difficult to achieve in practice with loudspeakers. The most suitable noise input to the speaker is a recording of the engine noise, but in some cases a white noise generator may be acceptable. If the corrected noise levels in the spaces of concern with the speaker excited are substantially less than those with the engine running, then it is clear that engine isolation is the first noise control that should be implemented. In this case, the best control that could be expected from engine isolation would be the difference in corrected noise level with the speaker excited and noise level with the engine running.

If the corrected noise levels in the spaces of concern with the speaker excited are similar to those measured with the engine running, then acoustic noise transmission is the likely path, although structure-borne noise may also be important, but at a slightly lower level. In this case, treatment to minimise airborne noise should be undertaken and after treatment, the speaker test should be repeated to determine if the treatment has been effective and to determine if structure-borne noise has subsequently become the problem.

Another example of the importance of determining the noise transmission path is demonstrated in the solution to an intense tonal noise problem in the cockpit of a fighter aircraft, which was thought to be due to a pump, as the frequency of the tone corresponded to a multiple of the pump rotational speed. Much fruitless effort was expended to determine the sound transmission path until it was shown that the source was the broadband aerodynamic noise at the air-conditioning outlet into the cockpit and the reason for the tonal quality was because the cockpit responded modally (see Chapter 7). The frequency of strong cockpit resonance coincided with a multiple of the rotational speed of the pump but was unrelated. In this case the obvious lack of any reasonable transmission path led to an alternative hypothesis and a solution.

1.3 ACOUSTIC FIELD VARIABLES

1.3.1 Variables

Sound is the sensation produced at the ear by very small pressure fluctuations in the air. The fluctuations in the surrounding air constitute a sound field. These pressure fluctuations are usually caused by a solid vibrating surface, but may be generated in other ways; for example, by the turbulent mixing of air masses in a jet exhaust. Saw teeth in high-speed motion (60 ms^{-1}) produce a very loud broadband noise of aerodynamic origin, which has nothing to do with vibration of the blade. As the disturbance that produces the sensation of sound may propagate from the source to the ear through any elastic medium, the concept of a sound field will be extended to include structure-borne as well as airborne vibrations. A sound field is described as a perturbation of steady-state variables, which describe a medium through which sound is transmitted.

For a fluid, expressions for the pressure, particle velocity, temperature and density may be written in terms of the steady-state (mean values) and the variable (perturbation) values as follows, where the variables printed in bold type are vector quantities:

Pressure:	$P_{tot} = P + p(\boldsymbol{r}, t)$ (Pa)
Velocity:	$\boldsymbol{U}_{tot} = \boldsymbol{U} + \boldsymbol{u}(\boldsymbol{r}, t)$ (m/s)
Temperature:	$T_{tot} = T + r(\boldsymbol{r}, t)$ (°C)
Density:	$\rho_{tot} = \rho + \sigma(\boldsymbol{r}, t)$ (kg/m^3)

Pressure, temperature and density are familiar scalar quantities that do not require discussion. However, explanation is required for the particle velocity $\boldsymbol{u}(\boldsymbol{r}, t)$ and the vector equation involving it, identified above by the word "velocity". The notion of particle velocity is based upon the assumption of a continuous rather than a molecular medium. "Particle" refers to a small bit of the assumed continuous medium and not to the molecules of the medium. Thus, even though the actual motion associated with the passage of an acoustic disturbance through the conducting medium, such as air at high frequencies, may be of the order of the molecular motion, the particle velocity describes a macroscopic average motion superimposed upon the inherent Brownian motion of the medium. In the case of a convected medium moving with a mean velocity $\boldsymbol{U}$, which itself may be a function of the position vector $\boldsymbol{r}$ and time t, the particle velocity $\boldsymbol{u}(\boldsymbol{r}, \underline{\underline{t}})$ associated with the passage of an acoustic disturbance may be thought of as adding to the mean velocity to give the total velocity. Combustion instability provides a notorious example.

Any variable could be chosen for the description of a sound field, but it is easiest to measure pressure in a fluid and strain, or more generally acceleration, in a solid. Consequently, these are the variables usually considered. These choices have the additional advantage of providing a scalar description of the sound field from which all other variables may be derived. For example, the particle velocity is important for the determination of sound intensity, but it is a vector quantity and requires three measurements as opposed to one for pressure. owever, instrumentation (Microflown) is available that allows the instantaneous measurement of particle velocity along all three cartesian coordinate axes at the same time. In solids, it is generally easiest to measure acceleration, especially in thin panels, although strain might be preferred as the measured variable in some special cases. If non-contact measurement is necessary, then instrumentation known as laser vibrometers are available that can measure vibration velocity along all three cartesian coordinate axes at the same time and also allow scanning of the surface being measured so a complete picture of the surface vibration response can be obtained for any frequency of interest. However, these instruments are quite expensive.

1.3.2 The Acoustic Field

In the previous section, the concept of sound field was introduced and extended to include structure-borne as well as airborne disturbances, with the implicit assumption that a disturbance initiated at a source will propagate with finite speed to a receiver. It is of interest to consider the nature of the disturbance and the speed with which it propagates. To begin, it should be understood that the small perturbations of the acoustic field may always be described as the sum of cyclic disturbances of appropriate frequencies, amplitudes and relative phases. In a fluid, a sound field will be manifested by variations in local pressure of generally very small amplitude with associated variations in density, displacement, particle velocity and temperature. Thus in a fluid, a small compression, perhaps followed by a compensating rarefaction, may propagate away from a source as a sound wave. The associated particle velocity lies parallel to the direction of propagation of the disturbance, the local particle displacement being first in the direction of propagation, then reversing to return the particle to its initial position after passage of the disturbance. A compressional or longitudinal wave has been described.

The viscosity of the fluids of interest in this text is sufficiently small for shear forces to play a very small part in the propagation of acoustic disturbances. A solid surface, vibrating in its plane without any normal component of motion, will produce shear waves in which the local particle displacement is parallel to the exciting surface, but normal to the direction of propagation of the disturbance. However, such motion is always confined to a very narrow region near to the vibrating surface and does not result in energy transport away from the near field region. Alternatively, a compressional wave propagating parallel to a solid surface will give rise to a similar type of disturbance at the fixed boundary, but again the shear wave will be confined to a very thin viscous boundary layer. Temperature variations associated with the passage of an acoustic disturbance through a gas next to a solid boundary, which is characterised by a very much greater thermal capacity, will likewise give rise to a thermal wave propagating into the boundary; but again, as with the shear wave, the thermal wave will be confined to a very thin thermal boundary layer of the same order of size as the viscous boundary layer. Such viscous and thermal effects, generally referred to as the acoustic boundary layer, are usually negligible for energy transport, and are generally neglected, except in the analysis of sound propagation in tubes and porous media, where they provide the energy dissipation mechanisms.

It has been shown that sound propagates in liquids and gases predominantly as longitudinal compressional waves; shear and thermal waves play no significant part. In solids, however, the situation is much more complicated, as shear stresses are readily supported. Not only are longitudinal waves possible, but so are transverse shear and torsional waves. In addition, the types of waves that propagate in solids strongly depend upon boundary conditions. In thin plates for example, bending waves, which are really a mixture of longitudinal and shear waves, predominate, with important consequences for acoustics and noise control. Bending waves are of importance in the consideration of sound radiation from extended surfaces, and the transmission of sound from one space to another through an intervening partition.

1.3.3 Magnitudes

The minimum acoustic pressure audible to the young human ear judged to be in good health, and unsullied by too much exposure to excessively loud music, is approximately 20×10^{-6} Pa, or 2×10^{-10} atmospheres (since one atmosphere equals 101.3×10^{3} Pa). The minimum audible level occurs between 3000 and 4000 Hz and is a physical limit; lower sound pressure levels would be swamped by thermal noise due to molecular motion in air.

For the normal human ear, pain is experienced at sound pressures of the order of 60 Pa or 6×10^{-4} atmospheres. Evidently, acoustic pressures ordinarily are quite small fluctuations about the mean.

1.3.4 The Speed of Sound

Sound is conducted to the ear through the surrounding medium, which in general will be air and sometimes water but sound may be conducted by any fluid or solid. In fluids, which readily support compression, sound is transmitted as longitudinal waves and the associated particle motion in the transmitting medium is parallel to the direction of wave propagation. However, as fluids support shear very weakly, waves dependent upon shear are weakly transmitted and often may be neglected. Consequently, longitudinal waves are often called sound waves. For example, the speed of sound waves travelling in plasma has provided information about the interior of the sun. In solids, which can support both compression and shear, energy may be transmitted by all types of waves, but only longitudinal wave propagation is referred to as "sound".

The concept of an "unbounded medium" will be introduced as a convenient and often used idealisation. In practice, the term, unbounded medium, has the meaning that longitudinal wave propagation may be considered sufficiently remote from the influence of any boundaries that such influence may be neglected. The concept of unbounded medium generally is referred as "free field" and this alternative expression will also be used where appropriate in this text.

The propagation speed of sound waves, called the phase speed in any conducting medium (solid or fluid), is dependent upon the stiffness, D, and the density, ρ, of the medium. The stiffness, D, however may be complicated by the boundary conditions of the medium and in some cases it may also be frequency dependent. These matters will be discussed in the following text. In this format the phase speed, c, takes the following simple form:

$$c = \sqrt{D/\rho} \qquad (\mathrm{m\,s^{-1}}) \tag{1.1}$$

The effect of boundaries on the longitudinal wave speed will now be considered but with an important omission for the purpose of simplification. The discussion will not include boundaries between solids, which generally is a seismic wave propagation problem not ordinarily encountered in noise control problems. Only propagation at boundaries between solids and fluids and between fluids will be considered, as they affect longitudinal wave propagation. At boundaries between solids and gases the characteristic impedance mismatch (see Section 1.12) is generally so great that the effect of the gas on wave propagation in the solid may be neglected; in such cases the gas may be considered to be a simple vacuum in terms of its effect on wave propagation in the solid.

In solids, the effect of boundaries is to relieve stresses in the medium at the unsupported boundary faces as a result of expansion at the boundaries normal to the direction of compression. Well removed from boundaries, such expansion is not possible and in a solid medium, the free field is very stiff. On the other hand, for the case of boundaries being very close together, wave propagation may not take place at all and in this case the field within such space, known as evanescent, commonly is assumed to be uniform. It may be noted that the latter conclusion follows from an argument generally applied to an acoustic field in a fluid within rigid walls. Here the latter argument has been applied to an acoustic field within a rigid medium with unconstrained walls.

For longitudinal wave propagation in solids, the stiffness, D, depends upon the ratio of the dimensions of the solid to the wavelength of a propagating longitudinal wave. Let the solid be characterised by three orthogonal dimensions h_i, i=1,2,3, which determine its overall size. Let h be the greatest of the three dimensions of the solid, where E is Young's modulus and f is the frequency of a longitudinal wave propagating in the solid. Then the criterion proposed for determining D is that the ratio of the dimension, h, to the half wavelength of the propagating longitudinal wave in the solid is greater than or equal to one. For example, wave propagation may take place along dimension h when the half wavelength of the propagating wave is less than or just equal to the dimension, h. This observation suggests that the following inequality must be satisfied for wave propagation to take place.

$$2hf \geq \sqrt{D/\rho} \tag{1.2}$$

For the case that only one dimension, h, satisfies the inequality and two dimensions do not then the solid must be treated as a wire or thin rod along dimension, h, on which waves may travel. In this case the stiffness constant, D, is that of a rod, D_r , and takes the following form:

$$D_r = E \tag{1.3}$$

The latter result constitutes the definition of Young's modulus of elasticity, E.

In the case that two dimensions satisfy the inequality and one dimension does not the solid must be treated as a plate over which waves may travel. In this case, where ν is Poisson's ratio (ν is approximately 0.3 for steel), the stiffness, $D = D_p$, takes the following form:

$$D_p = E/(1 - \nu^2) \tag{1.4}$$

For a material for which Poisson's ratio is equal to 0.3, $D = 1.099E$.

If all three dimensions, h_i, satisfy the criterion then wave travel may take place in all directions in the solid. In this case, the stiffness constant, $D = D_s$, takes the following form:

$$D_s = \frac{E(1 - \nu)}{(1 + \nu)(1 - 2\nu)} \tag{1.5}$$

For fluids, the stiffness, D_F, is the bulk modulus or the reciprocal of the more familiar compressibility. That is:

$$D_F = -V(\partial V/\partial P)^{-1} = \rho(\partial P/\partial \rho) \qquad \text{(Pa)} \tag{1.6a,b}$$

In Equation (1.6), V is a unit volume and $\partial V/\partial P$ is the incremental change in volume associated with an incremental change in static pressure P.

The effect of boundaries on the longitudinal wave speed in fluids will now be considered. For fluids (gases and liquids) in pipes at frequencies below the first higher order mode cut-on frequency (see Section 9.8.3.2), where only plane waves propagate, the close proximity of the wall of the pipe to the fluid within may have a very strong effect in decreasing the medium stiffness. The stiffness of a fluid in a pipe, tube or more generally, a conduit, will be written as D_C. The difference between D_F and D_C represents the effect of the pipe wall on the stiffness of the contained fluid. This effect will depend upon the ratio of the mean pipe radius, R, to wall thickness, t, the ratio of the density, ρ_w of the pipe wall to the density ρ of the fluid within it, Poisson's ratio, ν, for the pipe wall material, as well as the ratio of the fluid stiffness, D_F, to the Young's modulus, E, of the pipe wall. The expression for the stiffness, D_C, of a fluid in a conduit follows (Pavic, 2006):

$$D_C = \frac{D_F}{1 + \dfrac{D_F}{E}\left(\dfrac{2R}{t} + \dfrac{\rho_w}{\rho}\nu^2\right)} \tag{1.7}$$

The compliance of a pipe wall will tend to increase the effective compressibility of the fluid and thus decrease the speed of longitudinal wave propagation in pipes. Generally, the effect will be small for gases, but for water in plastic pipes, the effect may be large. In liquids, the effect may range from negligible in heavy-walled, small-diameter pipes to large in large-diameter conduits.

For fluids (gases and liquids), thermal conductivity and viscosity are two other mechanisms, besides chemical processes, by which fluids may interact with boundaries.

Generally, thermal conductivity and viscosity in fluids are very small, and such acoustical effects as may arise from them are only of importance very close to boundaries and in consideration of damping mechanisms. Where a boundary is at the interface between fluids or between fluids and a solid, the effects may be large, but as such effects are often confined to a very thin layer at the boundary, they are commonly neglected.

Variations in pressure are associated with variations in temperature as well as density; thus, heat conduction during the passage of an acoustic wave is important. In gases, for acoustic waves at frequencies well into the ultrasonic frequency range, the associated gradients are so small that pressure fluctuations may be considered to be essentially adiabatic; that is, no sensible heat transfer takes place between adjacent gas particles and, to a very good approximation, the process is reversible. However, at very high frequencies, and in porous media at low frequencies, the compression process tends to be isothermal. In the latter cases heat transfer tends to be complete and in phase with the compression.

For gases, use of Equation (1.1), the equation for adiabatic compression (which gives $D = \gamma P$) and the equation of state for gases, gives the following for the speed of sound:

$$c = \sqrt{\gamma P/\rho} = \sqrt{\gamma RT/M} \qquad \text{(m/s)} \tag{1.8a,b}$$