10

ENGINEERING NOISE CONTROL

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10.1. INTRODUCTION

As with any occupational hazard, control technology should aim at reducing noise to acceptable levels by action on the work environment. Such action involves the implementation of any measure that will reduce noise being generated, and/or will reduce the noise transmission through the air or through the structure of the workplace. Such measures include modifications of the machinery, the workplace operations, and the layout of the workroom. In fact, the best approach for noise hazard control in the work environment, is to eliminate or reduce the hazard at its source of generation, either by direct action on the source or by its confinement.

Practical considerations must not be overlooked; it is often unfeasible to implement a global control program all at once. The most urgent problems have to be solved first; priorities have to be set up. In certain cases, the solution may be found in a combination of measures which by themselves would not be enough; for example, to achieve part of the required reduction through environmental measures and to complement them with personal measures (e.g. wearing hearing protection for only 2-3 hours), bearing in mind that it is extremely difficult to make sure that hearing protection is properly fitted and properly worn.

This chapter presents the principles of engineering control of noise, specific control measures and some examples. Reading of chapter 1 is indispensable for the understanding of this chapter. Note that many of the specific noise control measures described are intended as a rough guide only. Further information on the subject can be found in ISO 11690 and in the specialised literature. Also suppliers of equipment and noise control hardware can often provide helpful noise control advice.

10.2. NOISE CONTROL STRATEGIES *(See ISO 11690)*

Prior to the selection and design of control measures, noise sources must be identified and the noise produced must be carefully evaluated. Procedures for taking noise measurements in the course of a noise survey are discussed in chapter 7.

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To adequately define the noise problem and set a good basis for the control strategy, the following factors should be considered:

- type of noise
- noise levels and temporal pattern
- frequency distribution
- noise sources (location, power, directivity)
- noise propagation pathways, through air or through structure
- room acoustics (reverberation).

In addition, other factors have to be considered; for example, number of exposed workers, type of work, etc. If one or two workers are exposed, expensive engineering measures may not be the most adequate solution and other control options should be considered; for example, a combination of personal protection and limitation of exposure.

The need for control or otherwise in a particular situation is determined by evaluating noise levels at noisy locations in a facility where personnel spend time. If the amount of time spent in noisy locations by individual workers is only a fraction of their working day, then local regulations may allow slightly higher noise levels to exist. Where possible, noise levels should be evaluated at locations occupied by workers' ears.

Normally the noise control program will be started using as a basis A-weighted immission or noise exposure levels for which the standard ISO 11690-1 recommends target values and the principles of noise control planning. A more precise way is to use immission and emission values in frequency bands as follows.

The desired (least annoying) octave band frequency spectrum for which to aim at the location of the exposed worker is shown in Figure 10.1 for an overall level of 90 dB(A). If the desired level after control is 85 dB(A), then the entire curve should be displaced downwards by 5 dB. The curve is used by determining the spectrum levels (see chapter 1) in octave bands and plotting the results on the graph to determine the required decibel reductions for each octave

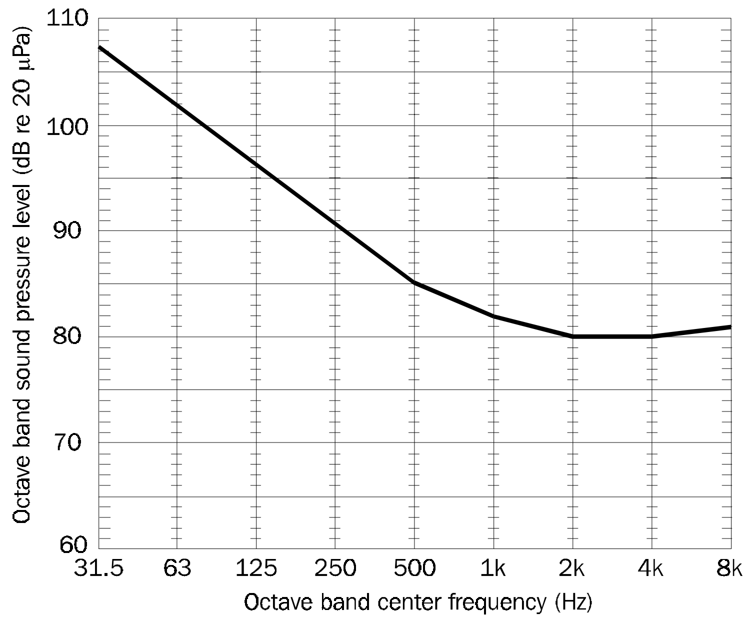

Figure 10.1. Desired noise spectrum for an overall level of 90 dB(A).

band. Clearly it will often be difficult to achieve the desired noise spectrum, but at least it provides a goal for which to aim.

It should be noted that because of the way individual octave band levels are added logarithmically, an excess level in one octave band will not be compensated by a similar decrease in another band. The overall A-weighted sound level due to the combined contributions in each octave band is obtained by using the decibel addition procedure described in chapter 1.

Any noise problem may be described in terms of a **source**, a **transmission path** and a **receiver** (in this context, a worker) and noise control may take the form of altering any one or all of these elements. The noise source is where the vibratory mechanical energy originates, as a result of a physical phenomenon, such as mechanical shock, impacts, friction or turbulent airflow. With regard to the noise produced by a particular machine or process, experience strongly suggests that when control takes the form of understanding the noise-producing mechanism and changing it to produce a quieter process, as opposed to the use of a barrier for control of the transmission path, the unit cost per decibel reduction is of the order of one tenth of the latter cost. Clearly, the best controls are those implemented in the original design. It has also been found that when noise control is considered in the initial design of a new machine, advantages manifest themselves resulting 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, occupational hygienists 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 use effective "add-on" noise control technology, which generally involves either modification of the transmission path or the receiver, and sometimes the source.

If noise cannot be controlled to an acceptable level at the source, attempts should then be made to control it at some point during its propagation path; that is, the path along which the sound energy from the source travels. In fact, there may be a multiplicity of paths, both in air and in solid structures. The total path, which contains all possible avenues along which noise may reach the ear, has to be considered.

As a last resort, or as a complement to the environmental measures, the noise control problem may be approached at the level of the receiver, in the context of this document, the exposed worker(s).

In existing facilities, controls may be required in response to specific complaints from within the workplace, and excessive noise levels may be quantified by suitable measurements as described previously. In proposed new installations, possible complaints must be anticipated, and expected excessive noise levels must be estimated by some procedure. As it is not possible to entirely eliminate unwanted noise, minimum acceptable levels of noise must be formulated and these levels constitute the criteria for acceptability (see chapter 4) which are generally established with reference to appropriate regulations in the workplace.

In both existing and proposed new installations 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 sourceidentification to specific sources on a particular item of equipment and rank ordering these contributions to the limits of practicality.

10.2.1. Existing installations and facilities *(See ISO 11690)*

In existing facilities, quantification of the noise problem involves identification of the 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 the locations from which the complaints arise. Once levels have been determined, the next step is to apply acceptable noise level criteria to each location and thus to determine the required noise reductions, generally as a function of octave or one-third octave frequency bands (see chapter 1).

Once the noise levels have been measured and the required reductions determined, the next step is to identify and rank order the noise sources responsible for the excessive noise. The sources may be subtle or alternatively many, in which case rank ordering may be as important as identification. Where many sources exist, rank ordering may pose a difficult problem.

When there are many 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 (ISO 4871) or by measurement, using methods outlined in chapter 1. The sound power should be characterised in octave or one third octave frequency bands (see chapter 1) and dominant single frequencies should be identified. Any background noise interfering with the sound power measurements must be taken into account and removed (see chapter 1).

NOTE: This is the ideal procedure. In reality, many people choose machinery or equipment using only noise emission values according to ISO 4871 and they make comparisons according to ISO 11689.

Often noise sources are either vibrating surfaces or unsteady fluid flow (air, gas or steam). The latter are referred to as aerodynamic sources and they are often associated with exhausts. In most cases, it is worthwhile to determine the source of the energy which is causing the structure or the aerodynamic source to radiate sound, as control may best start there.

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 excessive. For a facility involving just a few noise sources, as is the case for most occupational noise problems at a specific location, this is usually a relatively straightforward task.

Once the noise sources have been ranked in order of importance in terms of their contribution to the overall noise problem, it is often also useful to rank them in terms of which are easiest to do something about and which affect most people, and take this into account when deciding which sources to treat first of all. This is discussed in more detail in Chapter 7.

10.2.2. Installations and facilities in the design stage

In new installations, quantification of the noise problem at the design stage may range from simple to difficult but never impossible. At the design stage the problems are the same as for existing installations; 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. Mostimportantly,at the design stage the options for noise control are generally many and may include rejection of the proposed design.

The first step for new installations is to determine the noise criteria for sensitive locations which may typically include locations of operators of noisy machinery. 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 noise levels to the equipment manufacturer (care must be taken whenever importing equipment, particularly second hand which can be very noisy and hence no longer acceptable in the country of origin);
- including noise control fixtures (mufflers, barriers, vibration isolation systems, 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. In this context, note should be taken of the discussion on directivity in chapter 1. The essence of the discussion is that sources placed near hard reflective surfaces will result in higher sound levels at the approximate rate of 3 dB for each large surface, as illustrated in Figure 10.2. Note that the shape of the building space generally is not important, as a reverberant field can build-up in spaces of any shape. Care should be taken to organise production lines so that noisy equipment is separated from workers as much as possible.

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 in the long term.

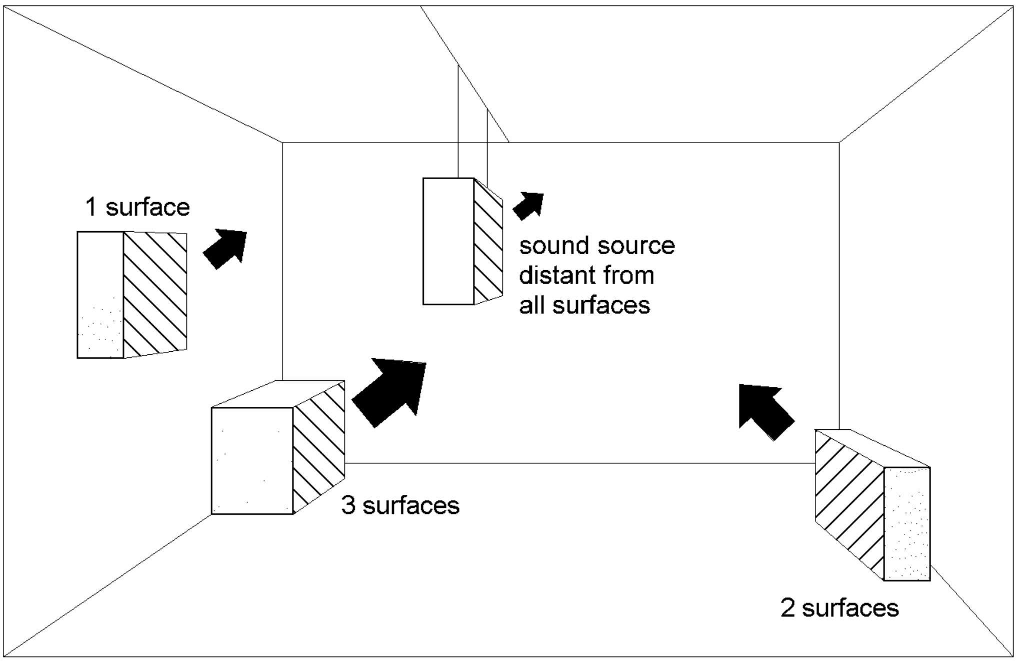

Figure 10.2. Sound sources should not be placed near corners (ASF, 1977)

10.3. CONTROL OF NOISE AT THE SOURCE *(See ISO TR 11688)*

To fully understand noise control, fundamental knowledge of acoustics is required. Although well covered in the specialised literature (OSHA, 1980; Beranek, 1988; Beranek and Ver, 1992;

Harris, 1991; Bies and Hansen, 1996), some fundamental concepts have been presented in chapter 1, and some additional concepts relevant to noise control are hereby reviewed.

To control noise at the source, it is first necessary to determine the cause of the noise and secondly to decide on what can be done to reduce it. 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 a longer time period over which the force acts) will dramatically reduce the noise generated.

Generally, when a choice of mechanical processes is possible to accomplish a given task, the best choice, from the point of view of minimum noise, will be the process which 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 which 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.

Among the physical phenomena which can give origin to noise, the following can be mentioned:(see also chapter 5)

- mechanical shock between solids,
- unbalanced rotating equipment
- friction between metal parts,
- vibration of large plates,
- irregular fluid flow, etc.

Control of noise at the source may be done either indirectly,i.e.generally, or directly,i.e. related to the design process addressing one of the causes cited above. The latter is the aim of ISO TR 11688.

NOTE: In noise control by design the terms direct and indirect sometimes are used for the path of sound from the generation to propagation in the air. So airborne sound in a fan is radiated directly but solidborne sound in a gear is transmitted to the wall of the casing and radiated as airborne sound indirectly.

GENERAL SOURCE NOISE CONTROL CAN INVOLVE:

- **Maintenance:**
  - replacement or adjustment of worn or loose parts;
  - balancing of unbalanced equipment;
  - lubrication of moving parts;
  - use of properly shaped and sharpened cutting tools.

- **Substitution of materials** (e.g., plastic for metal), a good example being the replacement of steel sprockets in chain drives with sprockets made from flexible polyamide plastics.

- **Substitution of equipment:**
  - electric for pneumatic (e.g. hand tools);
  - stepped dies rather than single-operation dies;
  - rotating shears rather than square shears;
  - hydraulic rather than mechanical presses;
  - presses rather than hammers;
  - belt conveyors rather than roller conveyors.

- **Specification of quiet equipment.**

- **Substitution of parts of equipment:**
  - modification of gear teeth, by replacing spur gears with helical gears - generally resulting in 10 dB of noise reduction);
  - replace straight edged cutters with spiral cutters (e.g. wood working machines - 10 dB(A) reduction);
  - replace gear drives with belt drives;
  - replace metal gears with plastic gears (beware of additional maintenance problems);
  - replace steel or solid wheels with pneumatic tyres.

- **Change of work methods**
  - in building demolition, replace use of ball machine with selective demolition;
  - replace pneumatic tools by changing manufacturing methods, such as moulding holes in concrete rather than cutting after production of concrete component;
  - use remote control of noisy equipment such as pneumatic tools;
  - separate noisy workers in time, but keep noisy operations in the same area, separated from non-noisy processes;
  - select slowest machine speed appropriate for a job - also select large, slow machines rather than smaller faster ones;
  - minimise width of tools in contact with workpiece (2 dB(A) reduction for each halving of tool width);
  - woodchip transport air for woodworking equipment should move in the same direction as the tool;
  - minimise protruding parts of cutting tools.

- **Substitution of processes.**
  - mechanical ejectors for pneumatic ejectors;
  - hot for cold working;
  - pressing for rolling or forging;
  - welding or squeeze rivetting for impact rivetting;
  - welding for rivetting;
  - use cutting fluid in machining processes;
  - change from impact action (e.g. hammering a metal bar) to progressive pressure action (e.g. bending metal bar with pliers as shown in Figure 10.3, or increase of time during which a force is applied, as shown in Figure 10.4);
  - replace circular saw blades with damped blades (see Figure 10.9);
  - replace mechanical limit stops with micro-switches.

- **substitution of mechanical power generation and transmission equipment**
  - electric motors for internal combustion engines or gas turbines;
  - belts or hydraulic power transmissions for gear boxes;

- **replacement of worn moving parts** (e.g., replace new rolling element bearings for worn ones);

- **minimising the number of noisy machines running at any one time**.

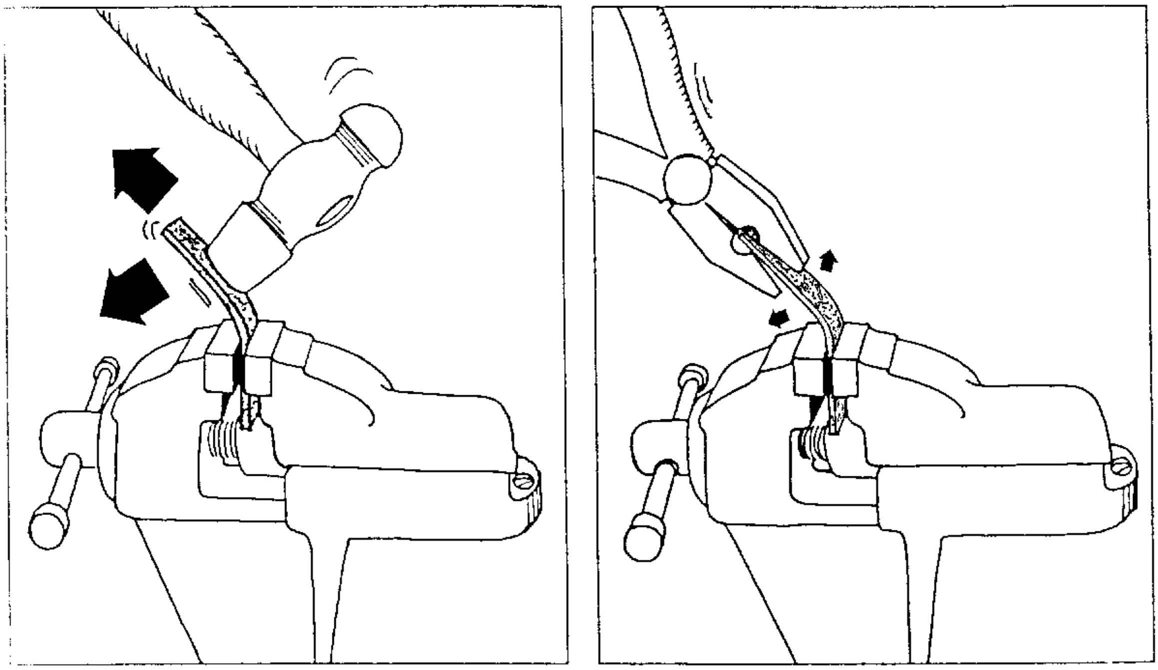

Figure 10.3. Example of bending instead of hammering (ASF, 1977)

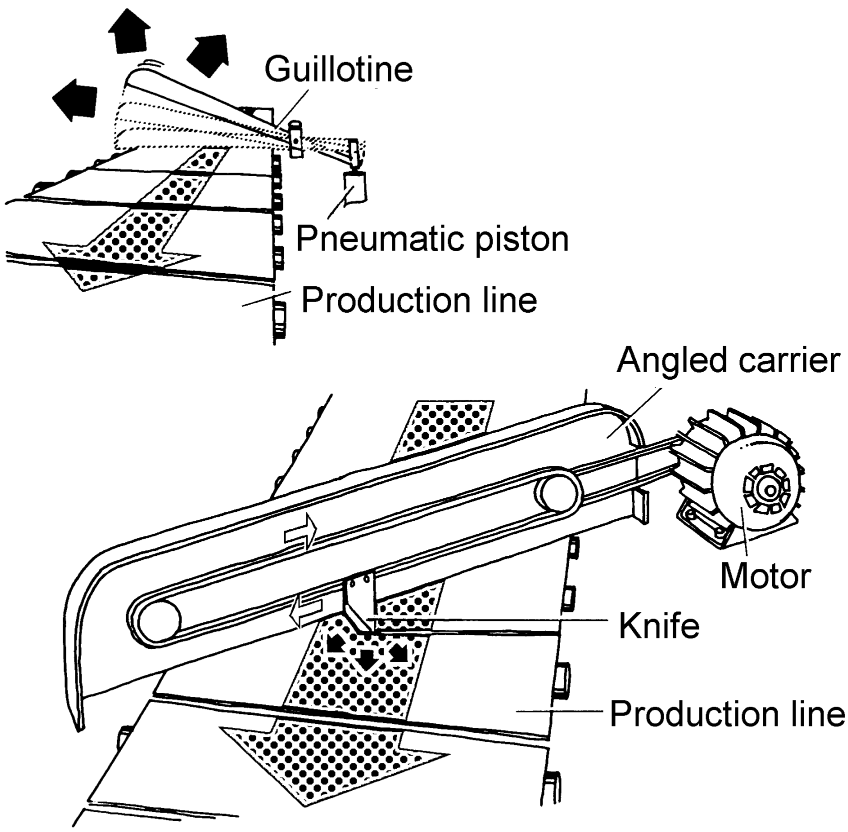

Figure 10.4. Example of increasing the time during which a force is applied (ASF, 1977).

SOURCE CONTROL BY DESIGN INVOLVES ***(See ISO/TR 11688)***

- **reduction of mechanical shock between parts** by:
  - **modifying parts** to prevent rattle and ringing;
  - using an **adjustable height collector** (see Figure 10.5a) for parts falling into a bin, so that impact speed and thus radiated noise is reduced;
  - using an **adjustable height conveyor and rubber flaps** to minimise the fall height of the parts (see Figure 10.5b);
  - **lining** of tumbling barrels, parts collecting bins, metal chutes, hoppers, etc. with elastic material, e.g. cork, hard rubber, plastic, conveyor belt material, with the choice of material being as soft as possible but sufficiently hard to withstand the particular operating environment without wearing out prematurely. In extreme cases, an effective alternative is to line the chute or bin with a thin layer of viscoelastic material such as silicone rubber or silastic, sandwiched between the bin and a second layer of steel or other abrasion resistant material, with the latter layer being of similar thickness to the wall of the bin or chute (see Figure 10.6);
  - **covering** metal tables, metal wheels, etc. with a **material**, such as rubber;
  - using conveyor belts instead of chutes to avoid noisy falls.

- **Reduction of noise resulting from out-of-balance** by:
  - **balancing** moving parts;
  - use of **vibration absorbers and dampers** tuned to equipment resonances (see Bies and Hansen, 1996, Ch. 10).

- **Reduction of noise resulting from friction between metal parts** by:
  - **lubrication or use of soft elastic interspacing** (the classical example of a noisy door to which oil is applied to the hinges demonstrates the efficiency of this measure).

- **Reduction of noise resulting from the vibration of large structures (plates, beams, etc.)** by:
  - **ensuring that machine rotational speeds do not coincide with resonance frequencies of the supporting structure**, and if they do, changing 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);
  - **reducing the acoustic radiation efficiency of the vibrating surface** by
    - replacement of a solid panel or machine guard with a woven mesh or perforated panel (see Figures 10.7a and b);
    - use of narrower belt drives, etc. (see Figure 10.8);
  - **damping a panel if it is excited mechanically** (see Figure 10.9), but note that if the panel is excited by an acoustic field, damping will have little or no effect upon its sound radiation;
    - the amount of damping already characterising a structure can be approximately determined by tapping it with a steel tool or rod. If the structure "rings" for a period after it is struck, then the damping is low. If only a dull thud is heard, then the damping is high. If the damping is low, then the surface may be treated either with a single layer damping treatment or a constrained layer treatment as described below. The noise reduction achieved is usually in the range 2 to 10 dB.

- **single layer damping treatments** are viscoelastic materials which include filled bitumens, silicone sealant and elastomeric polymers, all of which are available in the form of adhesive sheets or thick liquids for spraying or trowelling on to the surface to be treated. Care is necessary to ensure that the material selected can withstand the dirt, water or chemical environment to which it will be subjected. For effective results, the damping material thickness must be between one and three times the thickness of the surface being damped. Clearly this type of damping is most effective for thin structures.

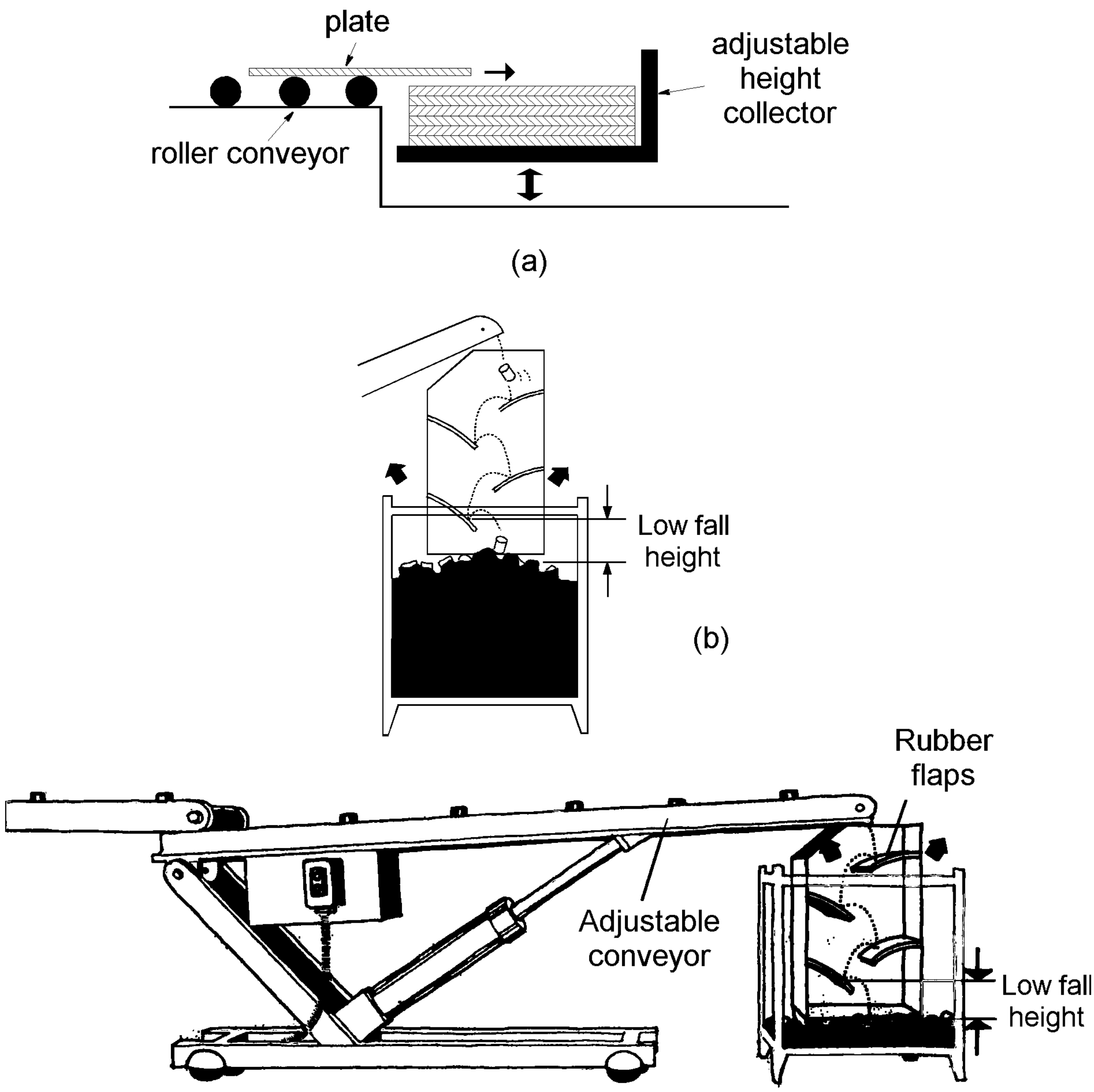

Figure 10.5. Examples of decrease of dropping height (ASF, 1977, with additions)
(a) Adjustable height collector.
(b) Adjustable height conveyor with rubber flaps.

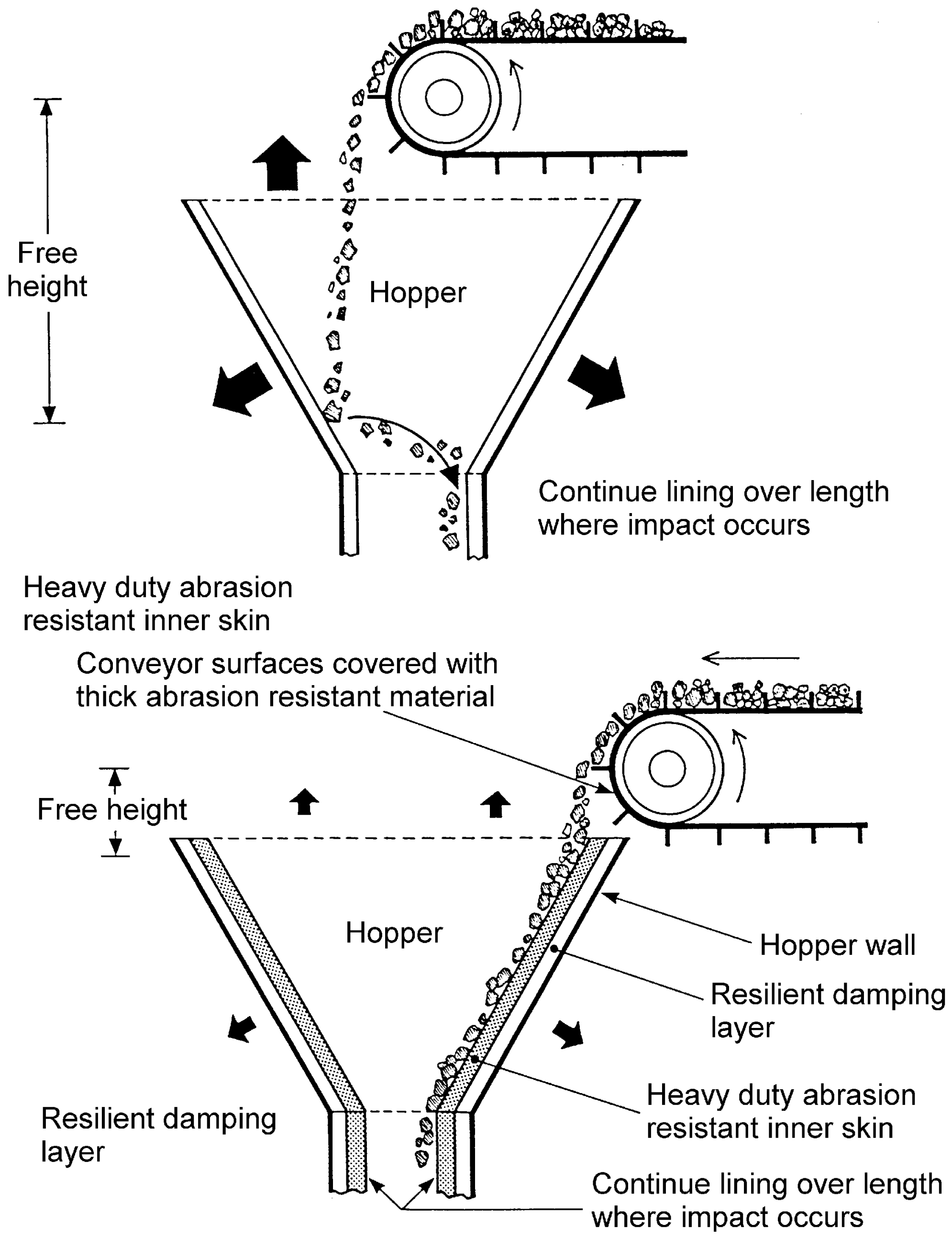

Figure 10.6. Lining a hopper with an impact absorbing and damping construction. Note that to achieve a constraint layer treatment, the "heavy duty abrasion resistant inner skin" in the lower figure could be replaced with a steel plate.

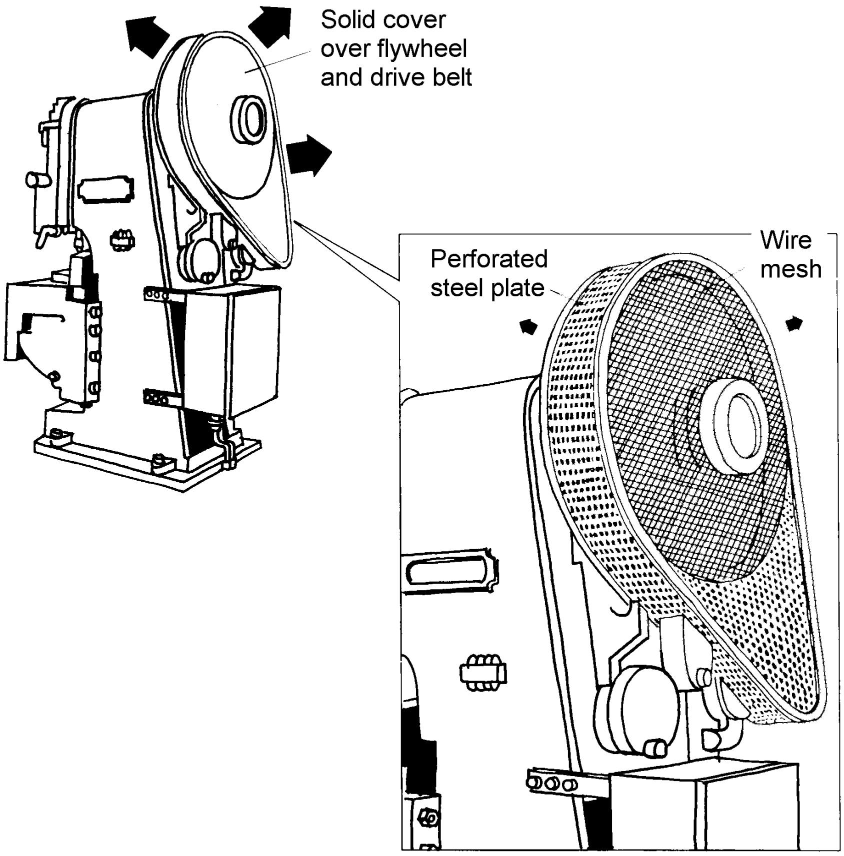

Figure 10.7(a). Use of a mesh protective cover for a flywheel, instead of a solid metal cover (ASF, 1977).

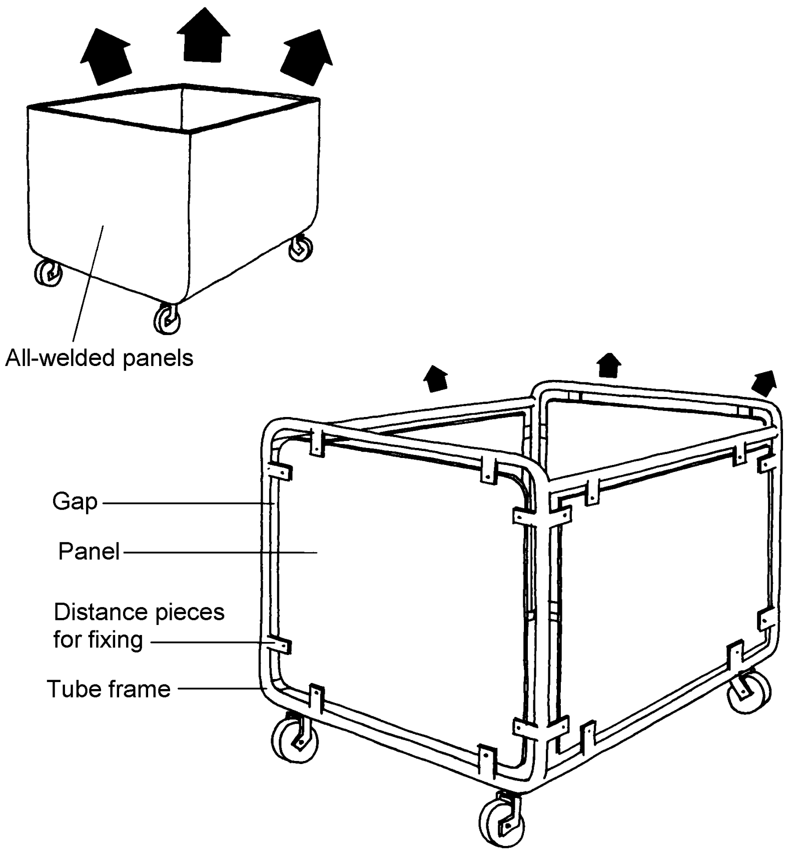

Figure 10.7(b). Use of open sided trolleys to transport material (ASF, 1977).

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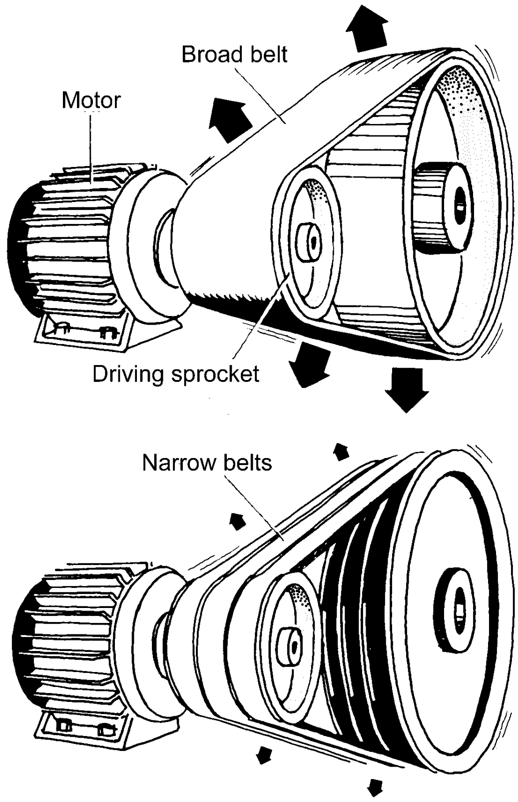

Figure 10.8. Use of narrower belts instead of a large belt drive (ASF, 1977).

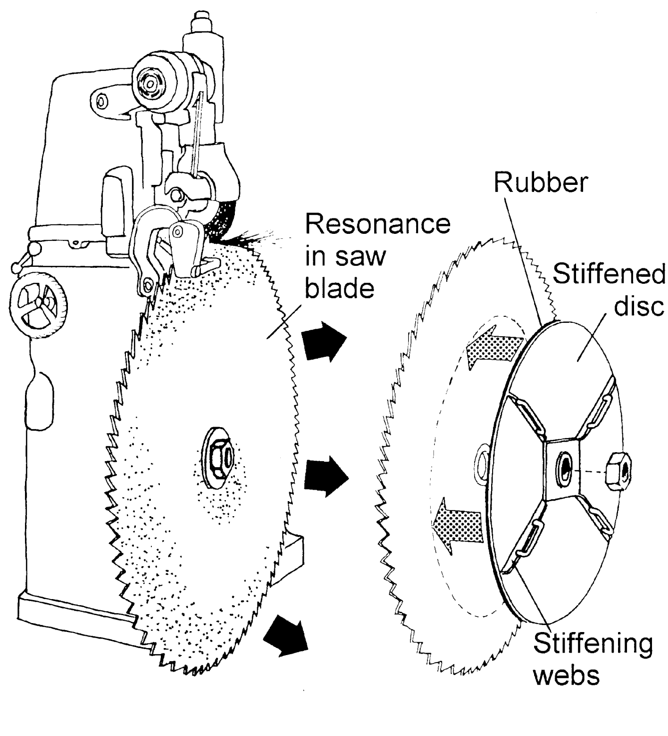

Figure 10.9. Damping of a circular saw (ASF, 1977).

  - **constrained layer damping** treatments, such as illustrated in figure 10.6, consist of a layer of viscoelastic material sandwiched between the structure and a layer of steel or aluminium. As a rule of thumb, the viscoelastic layer is about 1/3 as thick as the surface to be damped and for vibrating structures less than 1.5 mm thick the constraining outer layer should be the same thickness. For vibrating structures of thickness between 1.5 and 3mm, the constraining layer should be 1.5mm thick and for vibrating structures with a thickness of greater than 3mm, the constraining layer should be 3mm thick. Sometimes these rules of thumb produce a structure which is effectively damped in a particular frequency range which may not be the frequency range in which the noise radiation is a problem. In this case it may be necessary to fine tune the layer thicknesses, either experimentally or theoretically (see Cremer et. al., 1988).

- **reducing area of radiating surfaces**;
  - use of perforated sheet metal machine covers;
  - use a number of narrow drive belts rather than one wide one;
- **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;
- **isolating the vibration source from the noise radiating structure** by physically separating them (see Figure 10.10) or by using one or more of isolating elements (see Figure 10.11) - see for example Figures 10.12 and 10.13 - and taking into account the following factors (Bies and Hansen, 1996**)**
  - the **resonance frequency**, f_0 (Hz), associated with the stiffness of the isolating spring (k Newtons/metre) and the mass which it is supporting (m kg) and given by $f_0 = [1/(2\pi)]\sqrt{k/m}$ Hz, should be well below (less than half) the lowest frequency which is to be isolated (see Figure 10.14). The resonance frequency may also be calculated by knowing how much the isolating spring compresses (d cm) under the weight of the machine (static deflection); That is, $f_0 = 4.98/\sqrt{d}$ Hz.
  - the **excitation frequency**, f (Hz), for a rotating machine mounted on an isolator is generally equal to the rotational speed, expressed as revolutions per second.
  - the **transmissibility**, T, of an isolator is given by

$$T = \sqrt{\frac{1 + (2\zeta X)^2}{(1 - X^2)^2 + (2\zeta X)^2}} \qquad (1)$$

  - where $X=f/f_0$ and ζ is the critical damping ratio which is approximately 0.005 for steel springs, 0.05 for rubber mounts, 0.12 to 0.15 for silicone or low-T elastomers, 0.1 to 0.2 for glass fibre pads and 0.3 for a composite pad. Note that increasing damping reduces the vibration amplitude of the isolated system as the exciting frequency passes through resonance (on machine start-up, for example), but decreases the isolation achieved for excitation frequencies above this frequency;
  - the **isolation efficiency**, η, of the isolator is related to the transmissibility, T, by $\eta = (1 - T) \times 100\%$. Generally a value of η between about 85 and 95% is used.
  - for systems using **more than one isolator** (generally 4 are used to support the 4 corners of the base of the equipment being isolated), then resonance frequencies associated with twisting and rocking motions must also be calculated to ensure that they are well below the excitation frequency range (see Bies and Hansen, 1996, Ch. 10);

- **elastomeric materials** such as rubber are often preferred over steel springs because their greater damping reduces the large vibration amplitudes which occur when an excitation frequency coincides with the isolation system resonance (see Figure 10.14). Also, rubber materials prevent the transmission of vibration in the audio frequency range which is often transmitted along the coils of a steel spring. A disadvantage of rubber is its lack of tolerance for oily or sunny environments and this should be taken into account in a regular maintenance program;
- if **steel springs** are used, then rubber inserts should be placed between the spring and its attachment to the supporting structure to prevent the transmission of vibration in the mid-audio frequency range;
- **isolating materials** such as foam rubber, mineral wool and cork are often used for heavy equipment, but become ineffective in a relatively short time due to the elastic nature of the deflection of the material gradually changing to a permanent deflection;
- the dimensions of the **equipment support base** must be much larger than the height of the centre of gravity of the equipment, to minimise the risk of the equipment swaying unstably when the base is supported by flexible vibration isolators (see Figures 10.15);
- **isolator selection procedure:**
  - determine lowest continuous forcing frequency of machine to be isolated;
  - establish desired isolation efficiency and then calculate required transmissibility;
  - use the above transmissibility equation to determine X, which together with lowest continuous forcing frequency may be used to calculate the required resonance frequency for the isolator (as $X=f/f_0$);
  - use the static deflection equation with the required f_0 to calculate the required static deflection of the isolator;
  - knowing the weight supported by each isolator, refer to manufacturer's load deflection data to select the most suitable isolator;
- **lateral restraints (or snubbers)** are available to prevent too much sideways movement during machine start-up or during earthquakes;

- **reduction of noise resulting from fluid flow** by:
  - providing machines with **adequate cooling fins** so that noisy fans are no longer needed;
  - using **centrifugal rather than propeller fans** when fan use is unavoidable;
  - **locating fans in smooth, undisturbed air flow** (see Figure 10.16);
  - using **curved fan blades** designed to minimise turbulence (see Figures. 5.2 and 5.3 in chapter 5)or use **irregular spacing** in fans with straight blades as in traction motors (see Figure 10.17);
  - using of large, **low speed fans** rather than smaller, faster ones;
  - **minimising velocity** of fluid flow and increase cross-section of fluid streams;
  - **reducing the pressure drop** across any one component in a fluid flow system;
  - **minimising fluid turbulence** where possible (eg avoid obstructions in the flow);
  - choosing **quiet pumps** in hydraulic systems;
  - choosing **quiet nozzles** for compressed air systems (see Figures. 10.18(a) to (c));
  - **isolating pipes** carrying the fluid from support structures, as in Figure 10.11;
  - using **flexible connectors** in pipe systems to control energy travelling in the fluid as well as the pipe wall (see Figure 10.19);
  - using **flexible fabric sections** in low pressure air ducts (near the noise source such as a fan).

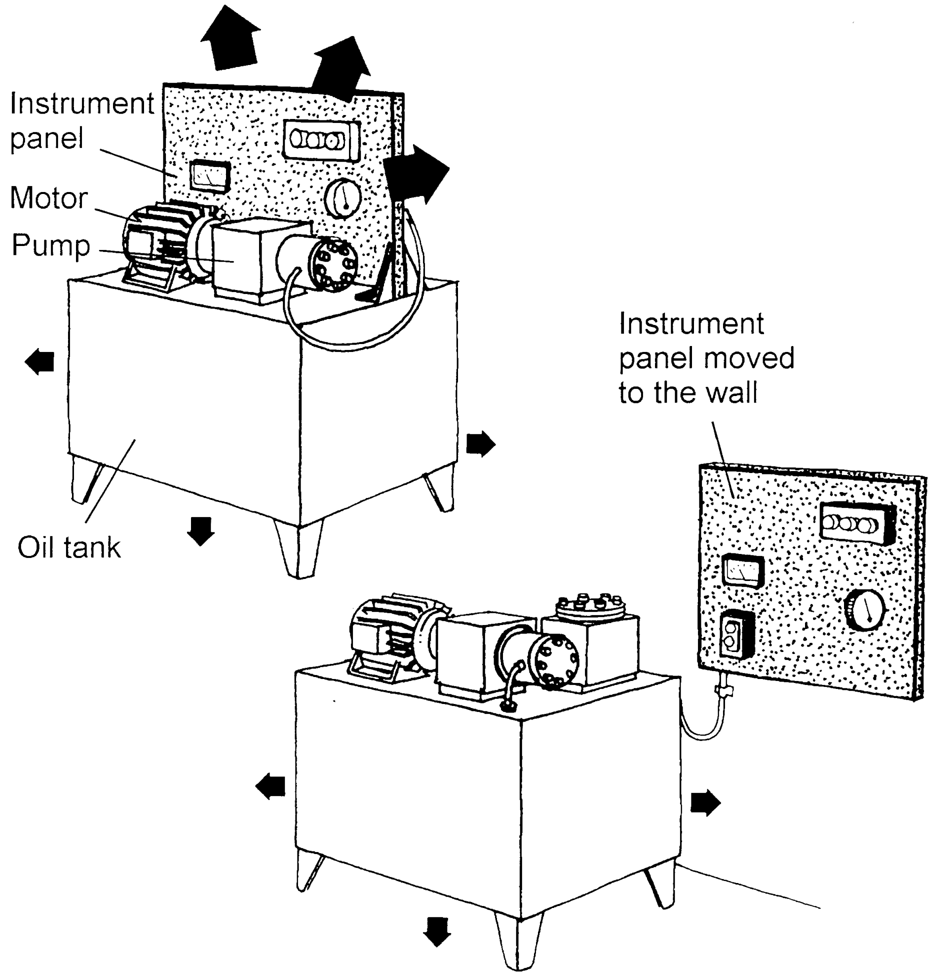

Figure 10.10. Vibration isolation by separation (ASF, 1977).

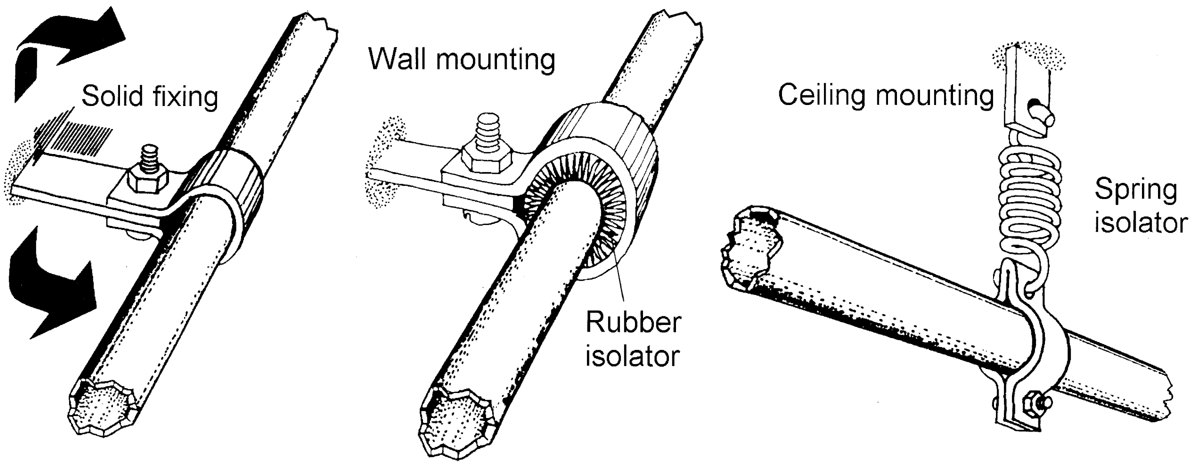

Figure 10.11. Reduction of vibration transmission from piping systems (ASF, 1977).

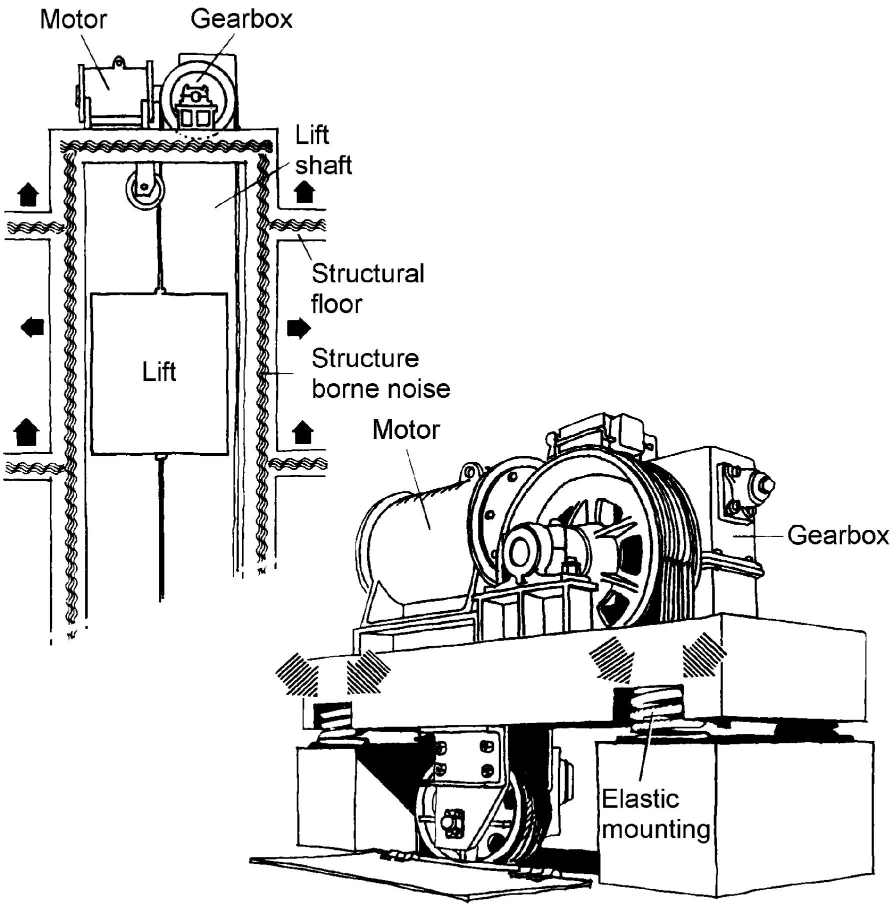

Figure 10.12. Vibration isolation of a lift to minimise lift noise transmitted throughout a building structure and then into occupied spaces (ASF, 1977).

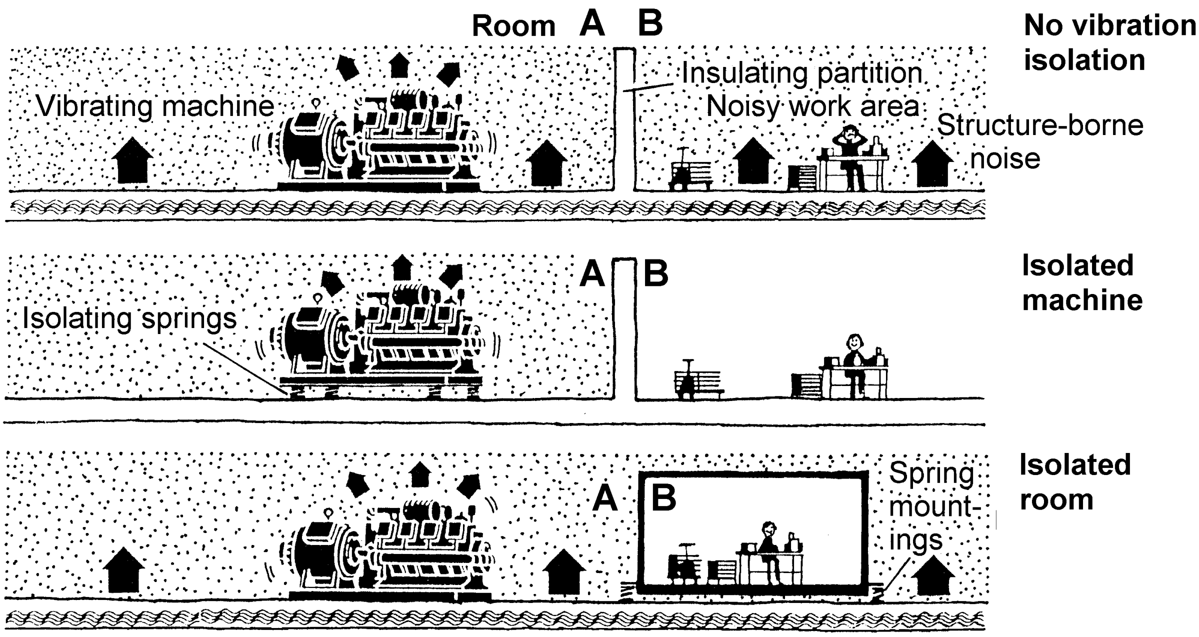

Figure 10.13. Vibration isolation to prevent noise transmitted through the machine supports to occupied spaces (ASF, 1977).

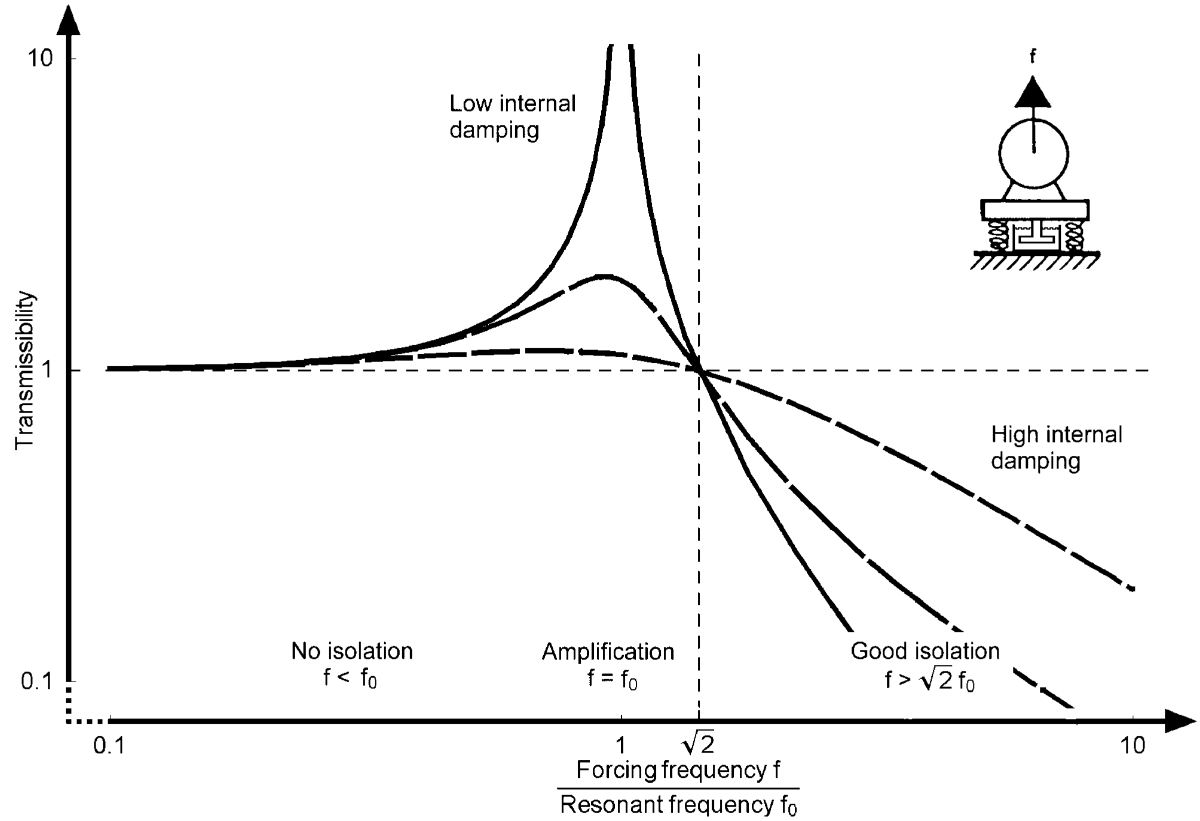

Figure 10.14. Illustration of vibration transmission through isolators as a function of frequency.

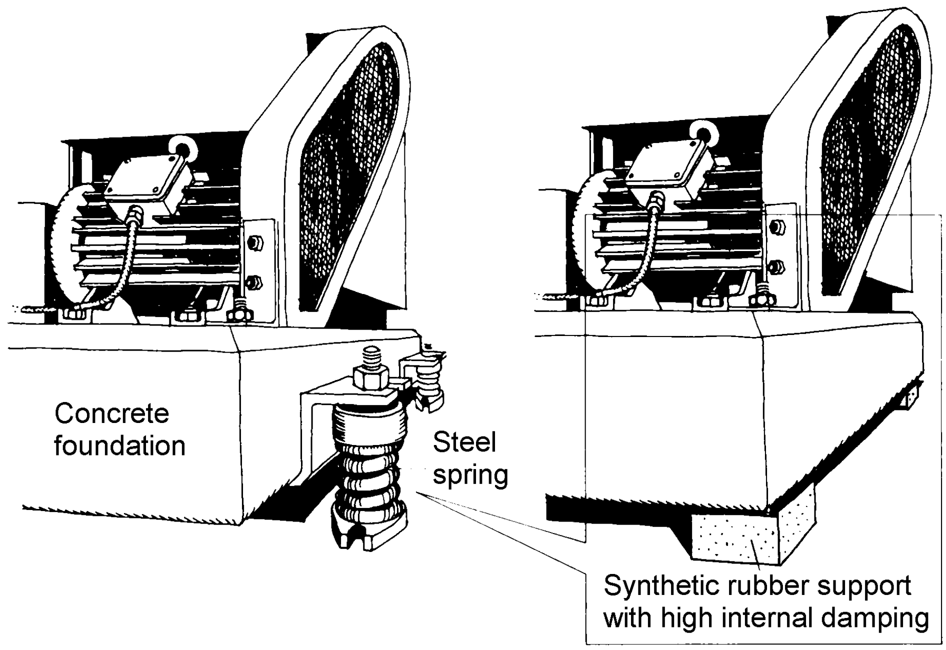

Figure 10.15. Steel vs rubber isolators (ASF, 1977).

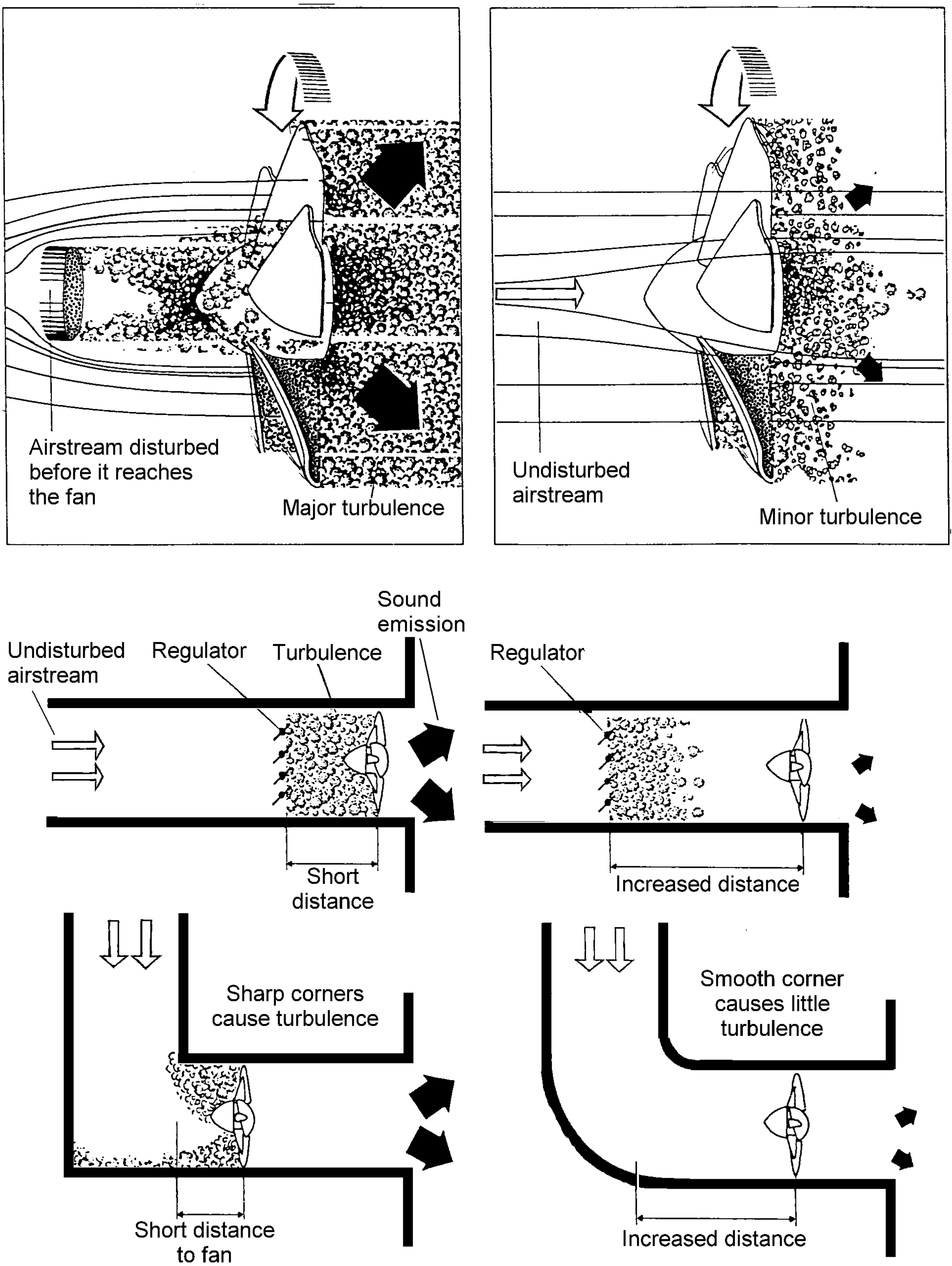

Figure 10.16. Location of fans in smooth air flows to reduce aerodynamic noise (ASF, 1977).

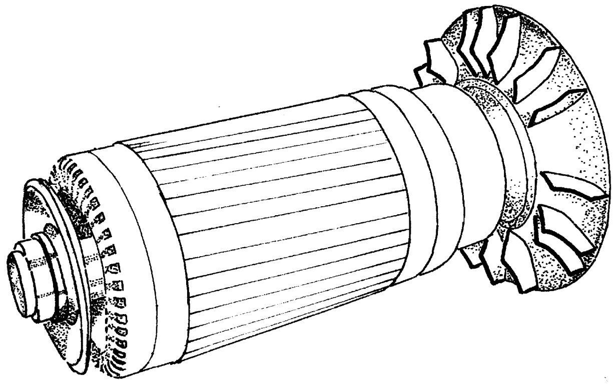

Figure 10.17. Centrifugal impeller with irregular spacing used in the self-ventilation of traction motors to reduce noise of straight blade fan (after Huebner, 1963).

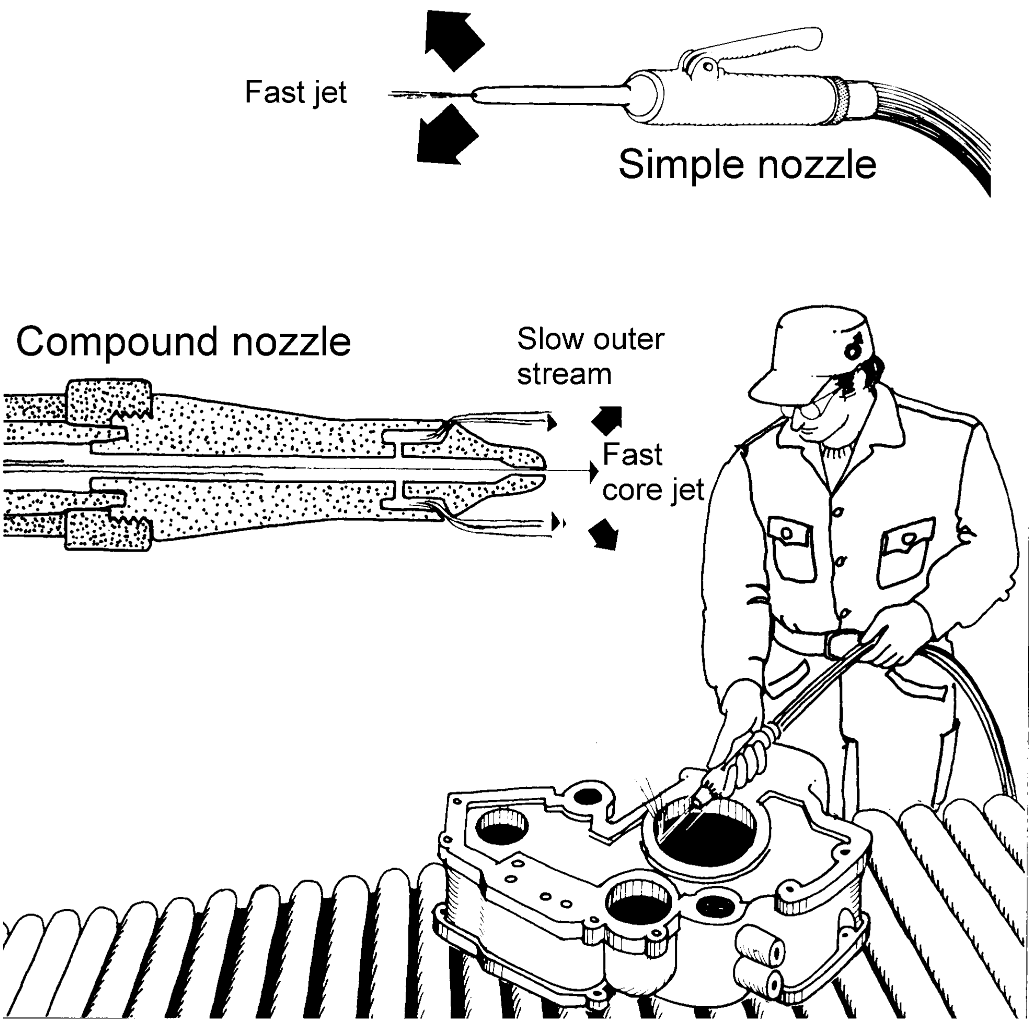

Figure 10.18(a). Use of a quiet air nozzle for air blasting of equipment (ASF, 1977).

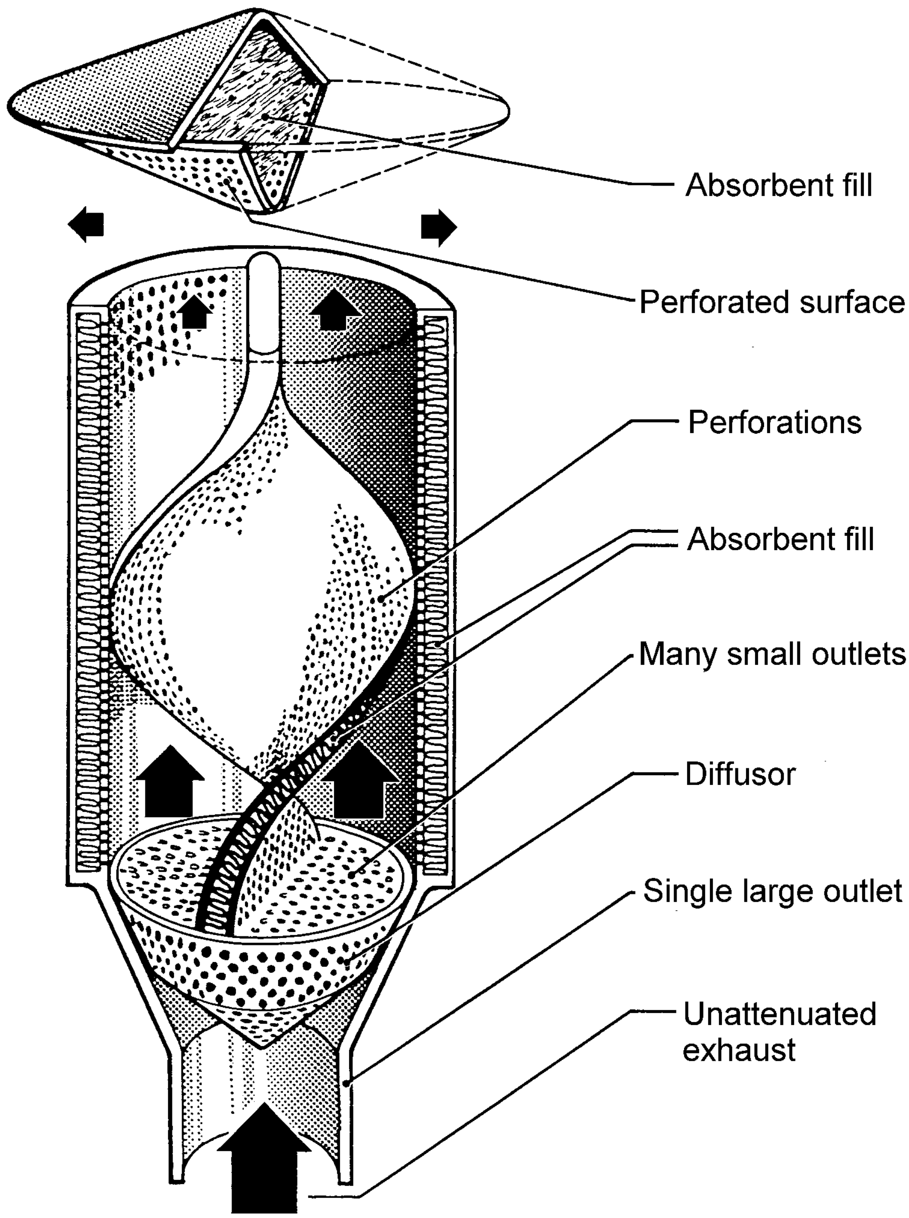

Figure 10.18(b). Use of a quiet nozzle for steam or air venting (ASF, 1977)

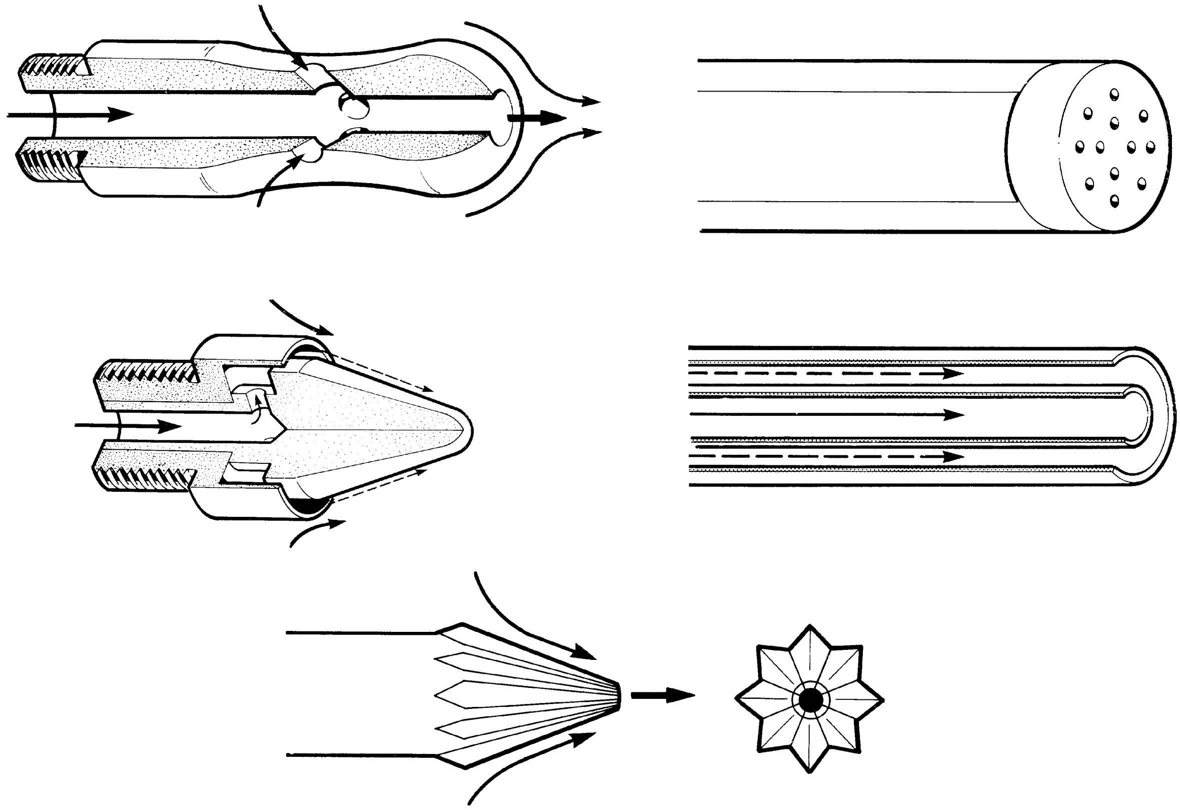

Figure 10.18(c). Commercially available low noise nozzles (after HSE 1985) for the same purpose as the nozzle of fig. 10.18(a), showing clockwise from left aspirated venturi nozzle, multi-jet nozzle, coannular jet nozzle, geometry effect nozzle and coanda effect nozzle; the most common and successful combination is the combined Coannual-Coanda effect nozzle, see also the compound nozzle in fig. 10.18(a).

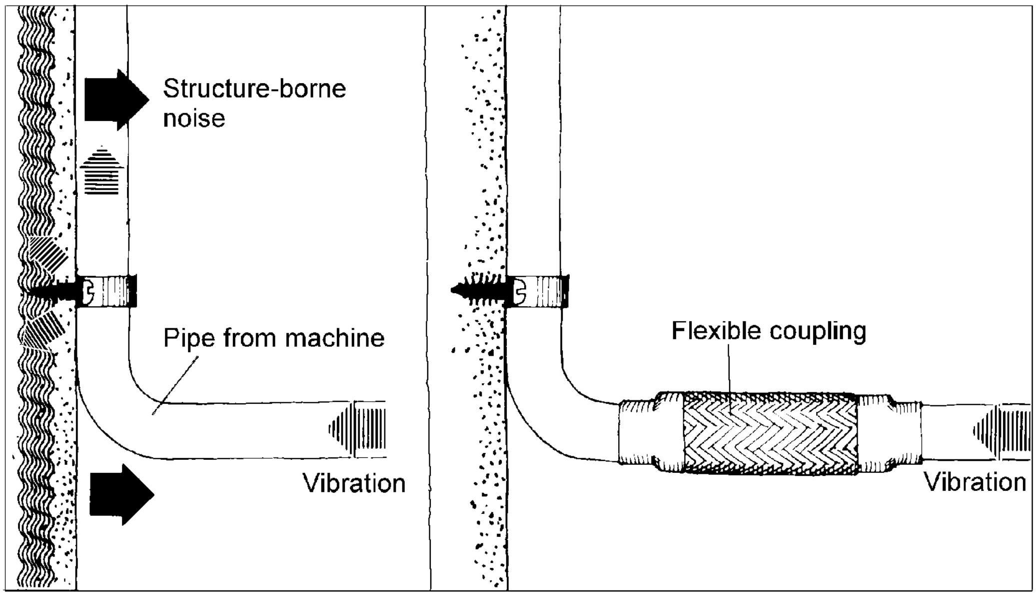

Figure 10.19. Reduction of vibration transmission along piping systems (ASF, 1977).

10.4. CONTROL OF NOISE PROPAGATION *(See ISO 11690-2)*

With regard to control of the noise during its propagation from the source to the receiver (generally the worker) some or all of the following actions need to be considered.

- Use of barriers (single walls), partial enclosures or full enclosure of the entire item of equipment.
- Use of local enclosures for noisy components on a machine.
- Use of reactive or dissipative mufflers; the former for low frequency noise or small exhausts, the latter for high frequencies or large diameter exhaust outlets.
- Use of lined ducts or lined plenum chambers for air handling systems
- Reverberation control - the addition of sound absorbing material to reverberant spaces to reduce reflected noise fields. Note that care should be taken when deciding upon this form of noise control, as direct sound arriving at the receiver will not be affected. Experience shows that it is extremely unusual to achieve noise reductions in excess of 3 or 4 dB(A) using this form of control which can be exorbitantly expensive when large spaces or factories are involved. In flat rooms the spatial sound distribution is of interest, see ISO 11690-1,-2.
- Active noise control, which involves suppression, reflection or absorption of the noise radiated by an existing sound source by use of one or more secondary or control sources.

To understand how best to design the propagation path controls mentioned above, the following concepts will be discussed: the determination of whether the problem arises from airborne or structure-borne transmission of the energy; noise absorption, reflection and reverberation; and transmission loss and isolation. This will be followed by a discussion of some of the controls which can be designed and implemented.

10.4.1. Airborne vs structure-borne noise

Very often in existing installations 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 is due to 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 that they are 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. 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 which should be implemented. In this case, the best control that could be expected from engine isolation would be the difference in noise levels in the space of concern with the speaker excited and 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, the 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 of an intense tonal noise in the cockpit of a fighter airplane 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. The frequency of strong cockpit resonance coincided with the 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.

10.4.2. Isolation of noise and transmission loss

The noise generated by a source can be prevented from reaching a worker by means of an obstacle to its propagation, conveniently located between the source and worker. This is the concept of sound isolation. Although one would ideally like the obstacle to isolate the noise completely, in practice, some of the noise always passes through it and the amount by which the noise is reduced by the obstacle, in dB, is dependent on the noise reducing properties of the material (its "transmission loss") and the acoustic properties of the room into which the noise is being transmitted. The transmission loss is defined as $TL = 10 \log_{10}\tau$, where the transmission coefficient, τ, is defined as the ratio of transmitted to incident energy (on the obstacle). If the receiving space is outdoors in a "free" field, the noise reduction is equal to the transmission loss (ignoring, for now the transmission of sound around the edges of the partition). If the receiving space is indoors, the noise reduction is given by

$$NR = TL - 10\log_{10}\left(\frac{A_{wall}}{S\bar{\alpha}}\right) \tag{2}$$

where

A_{wall} is the surface area of the partition and
$S\bar{\alpha}$ is the absorption of the receiving space

It can be seen from the preceding equation that the performance of the partition in reducing noise levels is improved as the amount of absorption in the receiving room is increased.

Example: The sound pressure level on one side of a 3m x 5m wall is measured 95 dB in the 500 Hz octave band. If the transmission loss of the wall is 35 dB in this band and in the receiving room $S\bar{\alpha}$ = 100 m^2, what will be the sound pressure level in the receiving room?

$$NR = 35 - 10\log_{10}\left(\frac{3\times 5}{100}\right) = 35 + 8.2 = 43.2 \text{ dB}$$

Transmission loss through a partition depends on the type of material of which it is made and it varies as a function of frequency. For usual industrial noise, the transmission loss through a partition increases by about 6 dB for each doubling of its weight per unit of surface area. Therefore, the best sound isolating materials are those which are compact, dense, and heavy.

The transmission loss achieved by a single isotropic partition can be estimated from its weight per unit area, for each frequency, from the graph presented in Figure 10.20. The straight portion of the curves is referred to as the "mass law" range because in this range the transmission loss of the partition is proportional to its mass; that is, every doubling of the mass per unit area of the partition results in a 6 dB increase in transmission loss.

The equation describing the transmission loss in the mass law range is given by:

$$TL = 20\log_{10}(fm) - 47 \quad \text{dB} \tag{3}$$

where f is the one third octave band centre frequency (Hz) and m is the mass per unit area of the partition (kg/m^2). The equation is only valid up to a maximum frequency of half the critical frequency, f_c, which is defined as the frequency at which the wavelength of sound waves in air is equal to the wavelength of the bending waves in the partition and is given by,

$$f_c = c^2(1 - \nu^2)^{1/2}/1.81\,c_L t \tag{4}$$

where c is the speed of sound in air (see chapter 1, equation (1)), t is the partition thickness (m) and c_L is the longitudinal wave speed in the panel given by $c_L = (E/\rho)^{1/2}$, where E, ρ and ν are Young's modulus of elasticity, density and Poisson's ratio respectively for the partition material. The lower frequency limit of validity of equation (3) is twice the lowest resonance frequency of the partition which is usually well below the audio frequency range for most constructions used for machine enclosures. Means for estimating the transmission loss of partitions at frequencies outside of the mass law range are discussed in the literature (Bies and Hansen, 1996, Ch. 8).

The term "isotropic" refers to a panel with uniform stiffness and mass properties and does not include ribbed or corrugated panels which are referred to as "orthotropic". The transmission loss of these latter panels is much less than the corresponding value for an isotropic panel and means for calculating the transmission loss of orthotropic panels are discussed in the specialised literature (Bies and Hansen, 1996, Ch. 8).

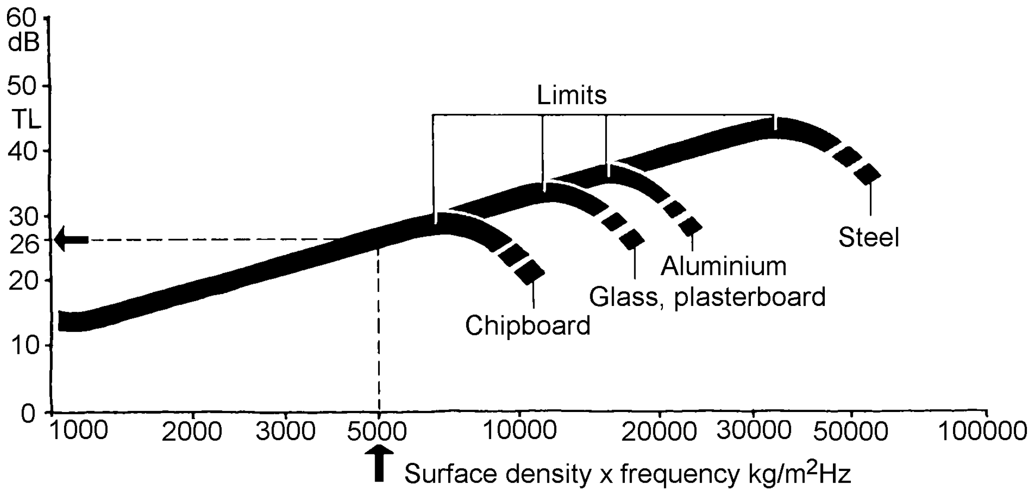

Figure 10.20. The transmission loss (TL) of a single wall is estimated from its weight per unit area (or surface weight) and frequency of the incident sound, (ASF, 1977).

Openings, even small holes or cracks greatly limit the noise reduction characteristics of a partition. The quantitative effect of openings and cracks in partitions is discussed in detail in Bies and Hansen (1996, Ch. 8). Including windows and doors in noise reducing walls also can have a large effect on the performance (Bies and Hansen, 1996, Ch. 8).

Noise reduction due to a partition can be substantially increased by constructing it as two panels separated by an air gap containing sound absorbing material. Details of these constructions and means for estimating their transmission loss are discussed in the literature (Bies and Hansen, 1996, Ch. 8).

Whenever planning for the isolation of a noise source, its characteristics (in terms of noise levels produced and frequency distribution) must be determined so that the appropriate material and construction can be selected. Tables giving transmission loss of usual construction materials and construction types as a function of frequency, may be found in the specialised literature and standards.

10.4.3. Enclosures *(See ISO 15667, ISO 11546-1, -2)*

The first task to consider in the design of an acoustic enclosure of a noise source is to determine the transmission paths from the source to the receiver and order them in relative importance. For example, on close inspection it may transpire that, although the source of noise is readily identified, the important acoustic radiation originates elsewhere, from structures mechanically connected to the source. In this case structure-borne sound is more important than the airborne component. In considering enclosures for noise control one must always guard against such a possibility; if structure-borne sound is the problem, an enclosure to contain airborne sound can be completely useless.

The wall of an enclosure may consist of several elements, each of which may be characterised by a different transmission loss. For example, the wall may be constructed of panels of different materials, it may include permanent openings for passing materials or cooling air in and out of the enclosure, and it may include windows for inspection and doors for access.

Each such element must be considered in turn in the design of an enclosure wall, and the transmission loss of the wall determined as an overall area weighted average of all of the elements.

For this calculation the following equation is used:

$$TL = -10\log_{10}\left(\frac{\sum_{i=1}^{q} S_i 10^{-TL_i/10}}{\sum_{i=1}^{q} S_i}\right) \tag{5}$$

where

S_i is the surface area (one side only), and

TL_i is the transmission loss of the ith element.

Example: Calculate the overall transmission loss at 125 Hz of a wall of total area 10 m^2 constructed from a material which has a transmission loss of 30 dB, if the wall contains a panel of area 3 m^2 of a material having a transmission loss of 10 dB .

The overall transmission loss is:

$$TL_{av} = 10 \log_{10}\left(\frac{7\times10^{-30/10} + 3\times10^{-10/10}}{10}\right) = 15.1 \text{ dB}$$

The noise reduction (NR) due to an enclosure may be calculated in terms of the transmission loss of the walls using the following equation.

$$NR = TL - C \tag{6}$$

where the quantity C is dependent on the enclosure internal conditions and may be estimated using Table 10.1.

Doors to give access to the enclosed equipment are usually needed and it must be possible to close them against rubber seals so that they are airtight. The transmission loss of both **doors and windows** should be as close as possible to that of the enclosure walls so that the presence of these items does not degrade the enclosure acoustic performance significantly. In the case of windows, this usually means double glazing and good rubber seals.

Many enclosures require some form of **ventilation** as illustrated in Figure 10.21(a) and (b). They may also require access for passing materials in and out. Such necessary permanent openings must be treated with some form of silencing to avoid compromising the effectiveness of the enclosure. In a good design, the acoustic performance of access silencing will match the performance of the walls of the enclosure. Techniques developed for the control of sound propagation in ducts may be employed for the design of silencers (see later section on mufflers).

Table 10.1. Values of constant *C* (dB) to account for enclosure internal acoustic conditions.

Enclosure internal acoustic conditions*	Octave band centre frequency (Hz)							
	63	**125**	**250**	**500**	**1,000**	**2,000**	**4,000**	**8,000**
live	18	16	15	14	12	13	15	16
fairly live	16	13	11	9	7	6	6	6
average	13	11	9	7	5	4	3	3
dead	11	9	6	5	3	2	1	1

*Use the following criteria to determine the appropriate acoustical conditions inside the enclosure:

live: *all enclosure surfaces and machine surfaces hard and rigid*
fairly live: *all surfaces generally hard but some panel construction (sheet metal or wood)*
average: *enclosure internal surfaces covered with sound-absorptive material, and machine surfaces hard and rigid*
dead: *as for "average", but machine surfaces mainly of panels*

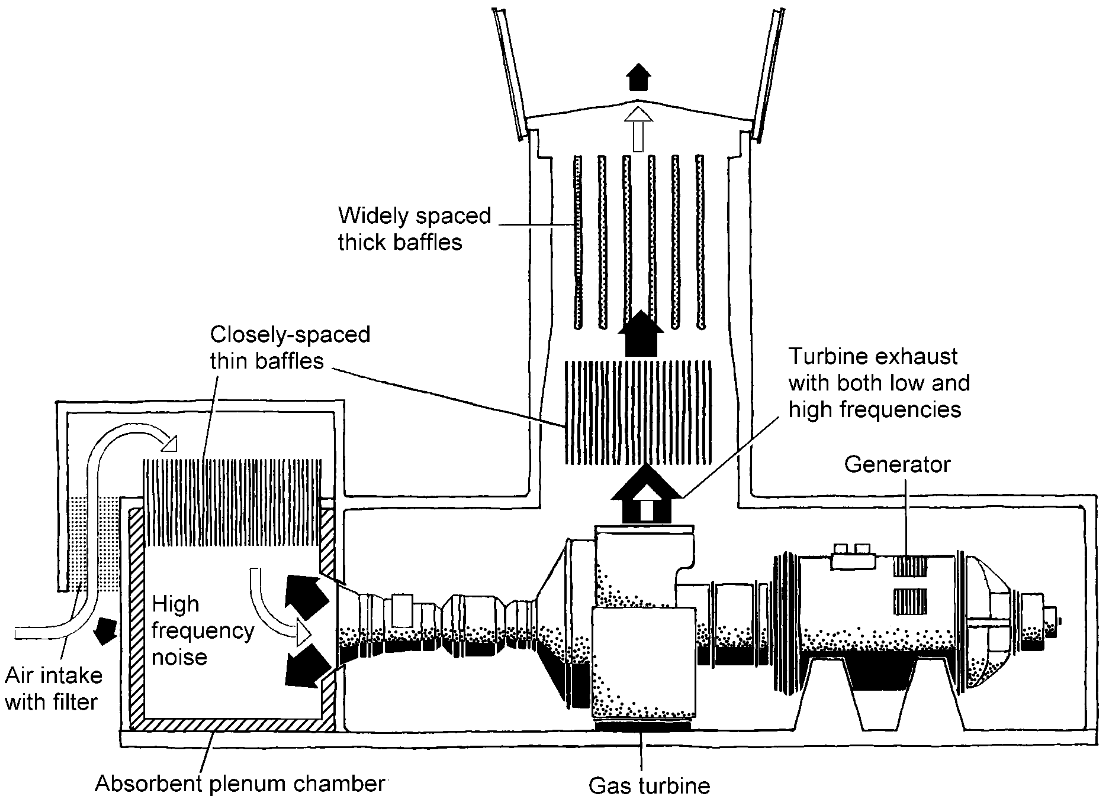

Figure 10.21(a). Example of a ventilated gas turbine enclosure (ASF, 1977).

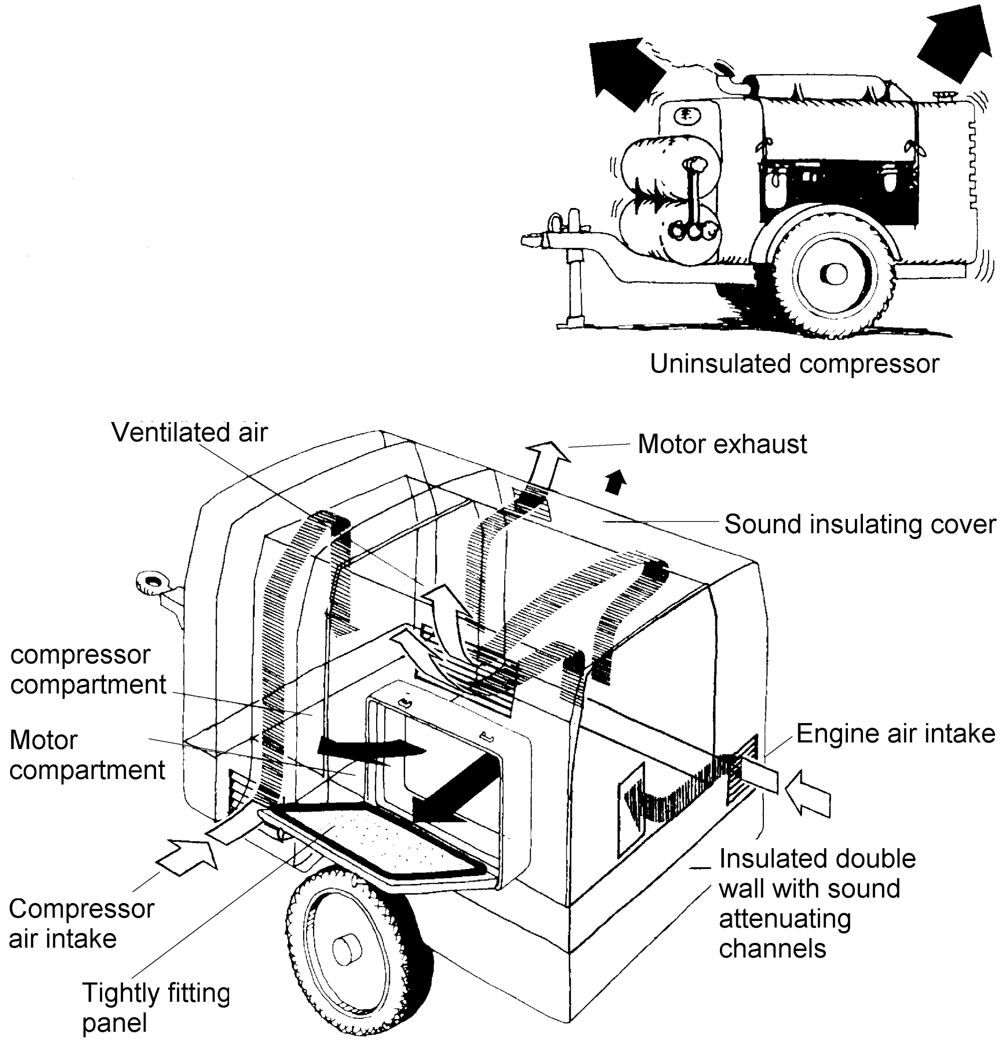

Figure 10.21(b). Example of a ventilated compressor enclosure (ASF, 1977).

If ventilation for heat removal is required but the heat load is not large, then natural ventilation with silenced air inlets (usually ducts lined in the inside with fiberglass or rockwool) at low levels close to the floor and silenced outlets at high levels, well above the floor, will be adequate. If forced ventilation is required to avoid excessive heating then the approximate amount of air flow needed can be determined using the following equation.

$$\rho C_p V = H/\Delta T \tag{7}$$

where

V is the volume ($m^3\ s^{-1}$) of airflow required,
H is the heat input (W) to the enclosure,
ΔT is the temperature differential (°C) between the external ambient and the maximum permissible internal temperature of the enclosure,
ρ is the gas (air) density ($kg\ m^{-3}$), and
C_p is the specific heat of the gas (air) in SI units (1,010 $m^2\ s^{-2}\ °C^{-1}$).

If a fan is provided for forced ventilation, the silencer will usually be placed external to the fan so that noise generated by the fan will be attenuated as well. When high volumes of air flow are required, the noise of the fan should be considered very carefully, as this noise source is quite often a cause of complaint. As fan noise generally increases with the fifth power of the blade tip speed, large slowly rotating fans are always to be preferred over small high-speed fans.

Any rigid connection between the machine and enclosure must be avoided. If at all possible, all pipes and service ducts passing through the enclosure wall should have flexible sections to act as vibration breaks; otherwise, the pipe or duct must pass through a clearance hole surrounded by a packing of mineral wool closed by cover plates and mastic sealant.

It is usually advisable to mount the machine on vibration isolators, particularly if low-frequency noise is the main problem. This ensures that little energy is transmitted to the floor. If this is not done there is the possibility that the floor surrounding the enclosure will re-radiate the energy into the surrounding space, or that the enclosure will be mechanically excited by the vibrating floor and act as a noise source. Sometimes it is not possible to mount the machine on vibration isolators. In this case, excitation of the enclosure can be avoided by mounting the enclosure itself on vibration isolators, while at the same time avoiding an air gap at the base of the enclosure by using heavy rubber flaps. Note that great care is necessary, when designing machinery vibration isolators, to ensure that the machine will be stable and that its operation will not be affected adversely. For example, if a machine must pass through a system resonance when running up to speed, then "snubbers" (which are usually horizontally mounted flexible elements) can be used to prevent excessive motion of the machine on its isolation mounts.

Two types of enclosure resonance are important and should be considered. The first is mechanical resonance of the enclosure panels, while the second is acoustic resonance of the air space between an enclosed machine and the enclosure walls. At the frequencies of these resonances the noise reduction due to the enclosure is markedly reduced from that calculated without regard to resonance effects.

The lowest order enclosure panel resonance is associated with a large loss in enclosure effectiveness at the resonance frequency. Thus the enclosure should be designed so that the resonance frequencies of its constituent panels are not in the frequency range in which appreciable sound attenuation is required. Only the lowest order, first few, panel resonances are of concern here. The panels may be designed such that their resonance frequencies are higher than or lower than the frequency range in which appreciable sound attenuation is required. Additionally, the panels should be well damped or "dead" which generally requires treatment with some form of visco-elastic damping material.

If the sound source radiates predominantly high-frequency noise, then an enclosure with low resonance frequency panels is recommended, implying a massive enclosure. On the other hand, if the sound radiation is predominantly low frequency in nature then an enclosure with a high resonance frequency is desirable, implying a stiff but not massive enclosure.

The resonance frequency of a panel may be increased by using stiffening ribs, but the increase that may be achieved is generally quite limited. For stiff enclosures with high resonance frequencies, materials with large values of Young's modulus to density ratio, E/ρ are chosen for wall construction, and for massive enclosures with small resonance frequencies, small values of E/ρ are chosen. In practice, stiff enclosures will generally be restricted to small enclosures.

If a machine is enclosed, reverberant build-up of the sound energy within the enclosure will occur unless adequate sound absorption is provided. The effect will be an increase of soundpressure at the inner walls of the enclosure over that which would result from the direct field of the source. A degradation of the noise reduction expected of the enclosure is implied.

In close-fitting enclosures, noise reduction may be degraded by yet another resonance effect. At frequencies where the average air spacing between a vibrating machine surface and enclosure wall is an integral multiple of half wavelengths of sound, strong coupling will occur between the vibrating machine surface and the enclosure wall, resulting in a marked decrease in the enclosure wall transmission loss.

The effect of inadequate absorption in enclosures is very noticeable. Table 10.2 shows the reduction in performance of an ideal enclosure with varying degrees of internal sound absorption. The sound power of the source is assumed constant and unaffected by the enclosure. "Percent" refers to the fraction of internal surface area which is treated.

Table 10.2. Enclosure interior noise increase as a function of percentage of internal surface covered with sound-absorptive material (referenced to 100% coverage).

Percent sound absorbent	10	20	30	50	70
interior noise increase (dB)	10	7	5	3	1.5

For best results, the internal surfaces of an enclosure are usually lined with glass or mineral fibre or open-cell polyurethane foam blanket. Typical values of absorption coefficients are given in Table 10.3.

Since the absorption coefficient of absorbent lining is generally highest at high frequencies, the high-frequency components of any noise will suffer the highest attenuation. Some improvement in low-frequency absorption can be achieved by using a thick layer of lining. However the liner should, in many cases, be protected from contamination with oil or water, to prevent its acoustical absorption properties from being impaired. This may be done by enclosing the liner in a polyethylene bag, about 20 µm thick. If mechanical protection is also required, then a perforated metal sheet of at least 25% open area may be added, provided that it does not contact the polyethylene bag lining. This latter condition can be achieved by placing an open wire mesh between the polyethylene and perforated metal. If this is not done, the performance of the sound absorbing material will be severely degraded.

The cost of acoustic enclosures of any type is proportional to size; therefore there is an economic incentive to keep enclosures as small as possible. Thus, because of cost or limitations of space, a close-fitting enclosure may be fitted directly to the machine which is to be quietened, or fixed independently of it but so that the enclosure internal surfaces are within, say, 0.5 m of major machine surfaces.

When an enclosure is close-fitting, the panel resonance frequencies will be somewhat increased due to the stiffening of the panel by the enclosed air volume. Thus an enclosure designed to be massive with a low resonance frequency may not perform as well as expected when it is close-fitting. Furthermore, system resonances will occur at higher frequencies; some of these modes of vibration will be good radiators of sound, producing low noise reductions, and some will be poor radiators, little affecting the noise reduction. The magnitude of the decrease in noise reduction caused by these resonances may be controlled to some extent by increasing the mechanical damping of the wall. Thus, if low-frequency sound (less than 200 Hz) is to be attenuated, the close-fitting enclosure should be stiff and well damped, but if high-frequency sound is to be attenuated the enclosure should be heavy and highly absorptive but not stiff.

Doubling of the volume of a small enclosure will normally lead to an increase in noise reduction of 3 dB at low frequencies, so that it is not desirable to closely surround a source, such as a vibrating machine, if a greater volume is possible.

Generally, if sufficient space is left within the enclosure for normal maintenance on all sides of the machine, the enclosure need not be regarded as close-fitting. If, however, such space cannot be made available, it is usually necessary to upgrade the transmission loss of an enclosure by up to 10 dB at low frequencies (less at high frequencies), to compensate for the expected degradation in performance of the enclosure due to resonances.

In many situations where easy and continuous access to parts of a machine is necessary, a complete enclosure may not be possible, and a partial enclosure must be considered (Alfredson & Seow, 1976). However, the noise reductions that can be expected at specific locations from partial enclosures are difficult to estimate and will depend upon the particular geometry. An example of a partial enclosure is shown in Figure 10.22. Estimates of the sound power reduction to be expected from various degrees of partial enclosure are presented in Figure 10.23.

Figure 10.23 shows fairly clearly that the enclosure walls should have a transmission loss of about 20 dB, and the most sound power reduction that can be achieved is about 10 dB. However, noise levels may in some cases be more greatly reduced, especially in areas immediately behind solid parts of the enclosure.

Some other practical considerations which should be taken into account are:

- who and what needs to be in the enclosure during operation of the noisy equipment (personnel should be excluded if possible);
- number and location of doors and windows (minimum possible);
- method of door closure (manual, automatic) and type of latch to ensure a tight seal around the door perimeter;
- automatic machine stop when doors are not closed properly;
- ease of cleaning inside the enclosure;
- ease of maintenance of enclosure and enclosed equipment;
- resistance of sound absorbing material to oil, dust, water or other chemicals; and
- attractiveness of finished enclosure

In many instances where there are a large number of noise sources and a few personnel in one or two localised areas, it may be preferable to enclose the people rather than the machines. In this case, many of the enclosure design principles outlined above still apply and the enclosure performance can be calculated using equation (6).

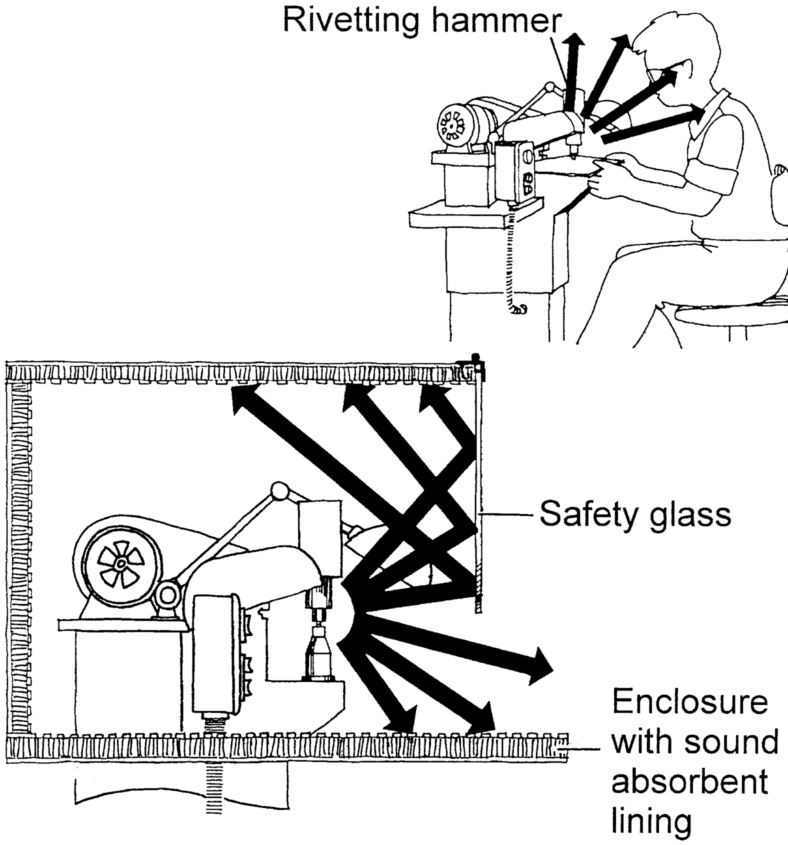

Figure 10.22. Example of a partial enclosure (ASF, 1977)

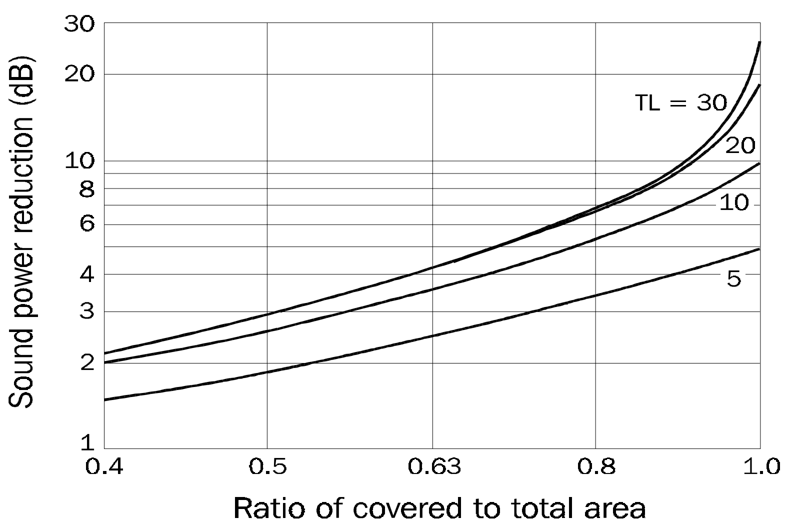

Figure 10.23. Estimate of sound power reduction due to a partial enclosure (see text for explanation).

10.4.4. Acoustic barriers *(See ISO 11821)*

Since detailed information on the calculation of the Insertion Loss of a single barrier (indoors and outdoors) is available in the literature (Bies and Hansen 1996, ISO 9613-2, ISO 10847, ISO 11821) this clause is concerned with the basic rules for the use of indoor barriers.

Barriers are placed between a noise source and a receiver as a means of reducing the direct sound observed by the receiver. In rooms, barriers suitably treated with sound-absorbing material may also slightly attenuate reverberant sound field levels by increasing the overall room absorption.

Barriers are a form of partial enclosure usually intended to reduce the direct sound field radiated in one direction only. For non-porous barriers having sufficient surface density, the sound reaching the receiver will be entirely due to diffraction around the barrier boundaries.

Now we will consider the effect of placing a barrier in a room where the reverberant sound field and reflections from other surfaces cannot be ignored.

In estimating the Insertion Loss of a barrier installed in a large room the following assumptions are implicit:

(1) The transmission loss of the barrier material is sufficiently large that transmission through the barrier can be ignored. A transmission loss of 20 dB is recommended.
(2) The sound power radiated by the source is not affected by insertion of the barrier.
(3) The receiver is in the shadow zone of the barrier; that is, there is no direct line of sight between source and receiver.
(4) Interference effects between waves diffracted around the side of the barrier, waves diffracted over the top of the barrier and reflected waves are negligible. This implies octave band analysis.

Barriers are ineffective in a highly reverberant environment. The performance of an indoor barrier is always improved by hanging absorptive baffles from the ceiling or by placing sound absorbing material directly on the ceiling (see Figure 10.24).

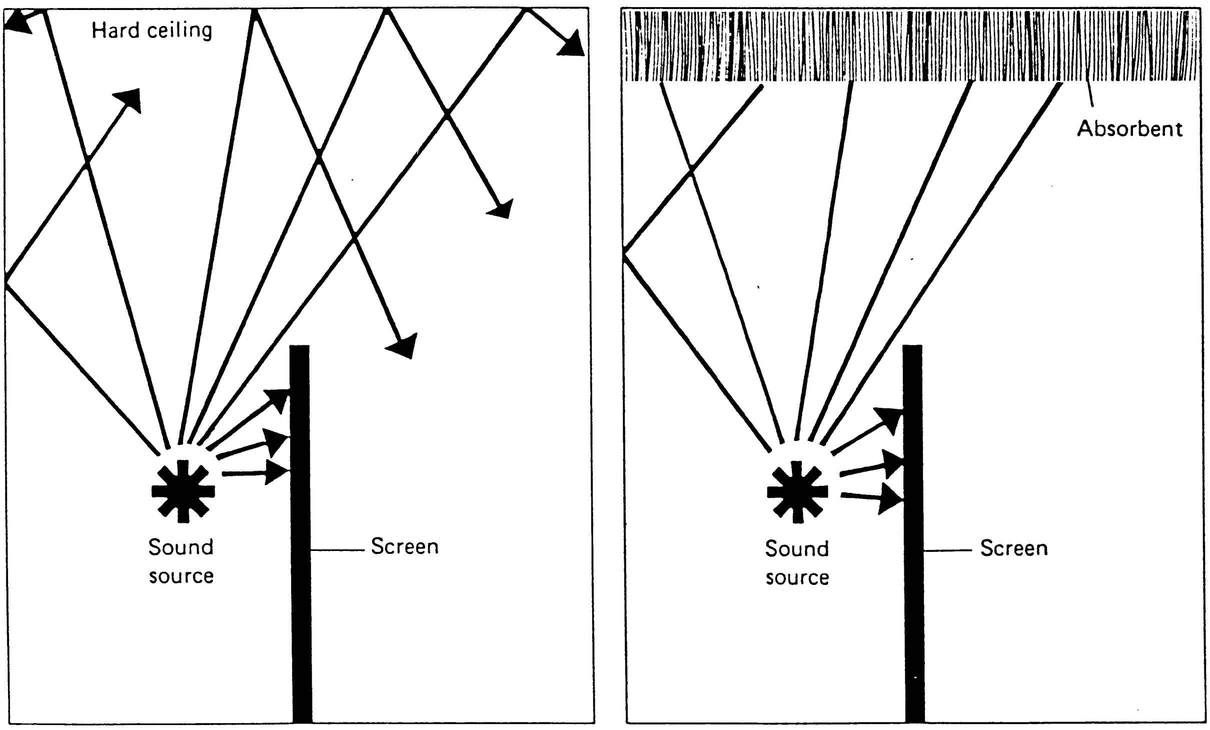

Figure 10.24. Use of a barrier indoors (ASF, 1977)

When multiple barriers exist, as in an open-plan office, experimental work (West & Parkin, 1978) has shown the following statements to be true in a general sense (test screens were 1.52 m high by 1.37 m wide):

(1) No difference in attenuation is obtained when a 300 mm gap is permitted between the base of the screen and the floor.

(2) When a number of screens interrupt the line of sight between source and receiver, an additional attenuation of up to 8 dB(A) over that for a single screen can be realised.

(3) Large numbers of screens remove wall reflections and thus increase the attenuation of sound with distance from the source.

(4) For a receiver immediately behind a screen, a local shadow effect results in large attenuation, even for a source a large distance away. This is in addition to the effect mentioned in (2) above.

(5) For a screen less than 1 m from a source, floor treatment has no effect on the screen's attenuation.

(6) A maximum improvement in attenuation of 4-7 dB as frequency is increased from 250 to 2 kHz can be achieved by ceiling treatment. However, under most conditions, this greater attenuation can only be achieved at the higher frequencies.

(7) Furnishing conditions are additive; that is, the attenuations measured under two different furnishing conditions are additive when the two furnishing conditions coexist.

10.4.5. Mufflers and lined ducts *(See ISO 14163, ISO 11820)*

Muffling devices are commonly used to reduce noise associated with internal combustion engine exhausts, high pressure gas or steam vents, compressors and fans. These examples lead to the conclusion that a muffling device allows the passage of fluid while at the same time restricting the free passage of sound. Muffling devices might also be used where direct access to the interior of a noise containing enclosure is required, but through which no steady flow of gas is necessarily to be maintained. For example, an acoustically treated entry way between a noisy and a quiet area in a building or factory might be considered as a muffling device.

Muffling devices may function in any one or any combination of three ways: they may suppress the generation of noise; they may attenuate noise already generated; and they may carry or redirect noise away from sensitive areas. Careful use of all three methods for achieving adequate noise reduction can be very important in the design of muffling devices, for example, for large volume exhausts.

Two terms, insertion loss, IL, and transmission loss, TL, are commonly used to describe the effectiveness of a muffling system. The insertion loss of a muffler is defined as the reduction (in decibels) in sound power transmitted through a duct compared to that transmitted with no muffler in place. Provided that the duct outlet remains at a fixed point in space, the insertion loss will be equal to the noise reduction which would be expected at a reference point external to the duct outlet as a result of installing the muffler. The transmission loss of a muffler, on the other hand, is defined as the difference (in decibels) between the sound power incident at the entry to the muffler to that transmitted by the muffler.

Muffling devices make use of one or the other or a combination of the two effects in their design. Either sound propagation may be prevented (or strongly reduced) by reflection (generally as the result of using orifices and expansion chambers), or sound may be dissipated, generally by the use of sound absorbing material. Muffling devices based upon reflection are called reactive devices and those based upon dissipation are called dissipative devices. A duct lined with sound absorbing material on its walls is one form of dissipative muffler.

A type of combined reactive/dissipative muffler is a plenum chamber which is a large volume chamber which connects two ducts. The interior of the chamber is lined with sound absorbing material, and thus part of the high frequency sound energy which enters the chamber is absorbed due to multiple reflections within the unit, while the low frequency energy is reflected or suppressed because of the sudden expansion and contraction in effective duct cross-sectional area as a result of the presence of the chamber. An example of a reactive muffler is shown in Figure 10.25, dissipative mufflers are shown in Figures 10.26(a)-(d), and a combined reactive / dissipative muffler is shown in Figure 10.27.

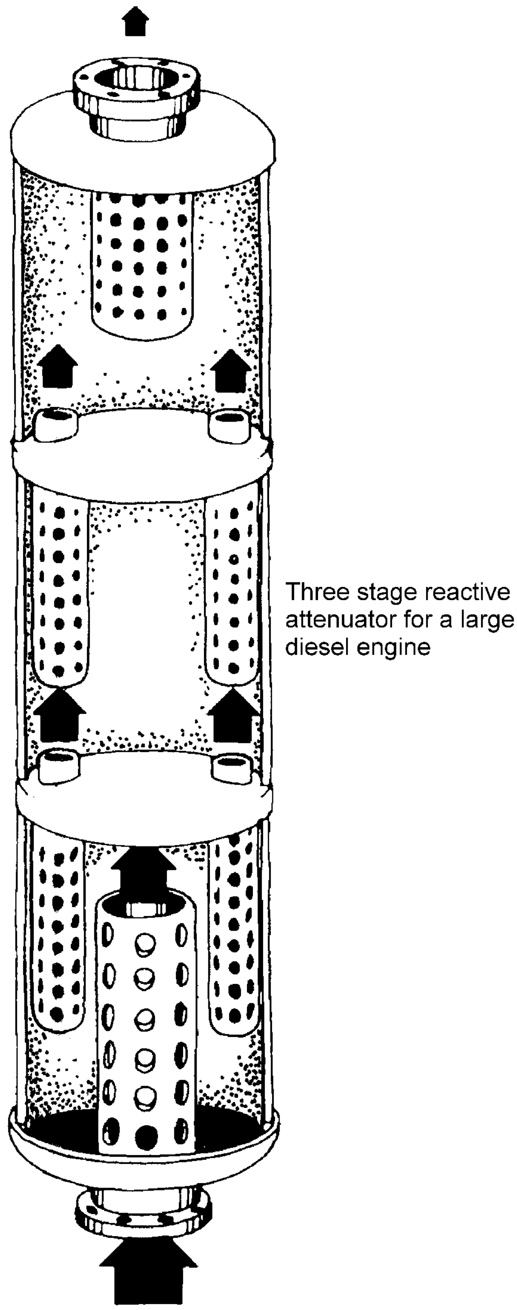

Figure 10.25. An internal combustion engine reactive muffler (ASF, 1977).

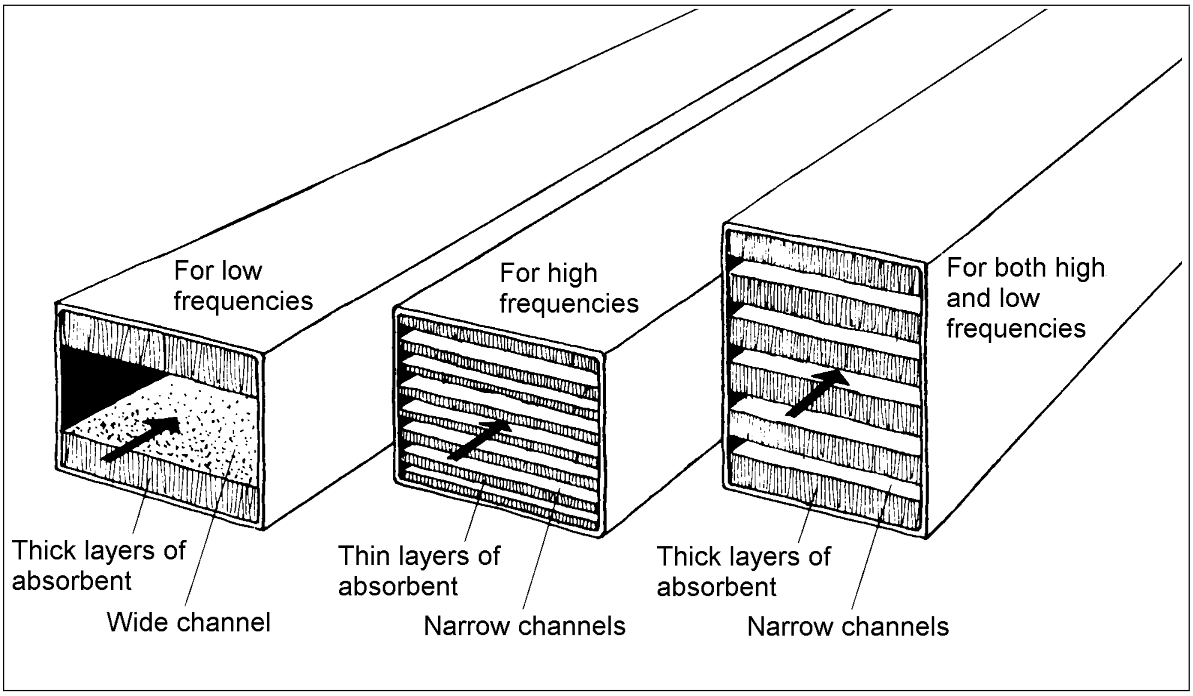

Figure 10.26(a). Examples of general dissipative mufflers (ASF, 1977).

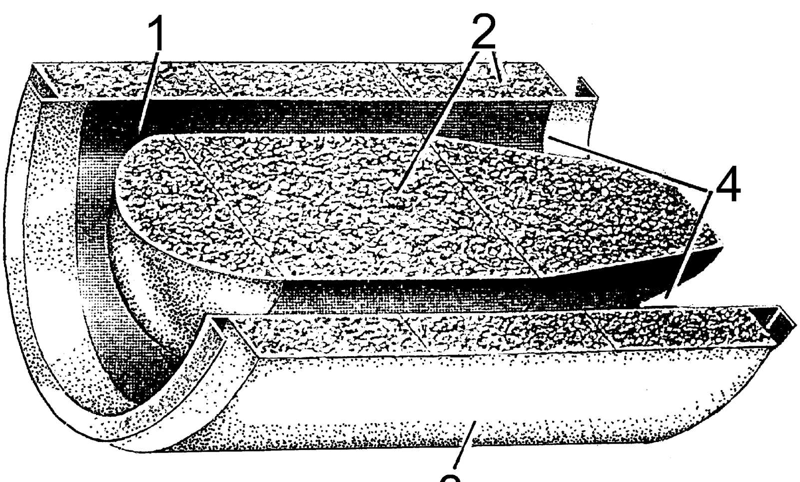

Figure 10.26(b). Circular silencer. (1) perforated, galvanized steel; (2) fiberglass or mineral wool acoustic fill; (3) steel casing; (4) low turbulence air passages.

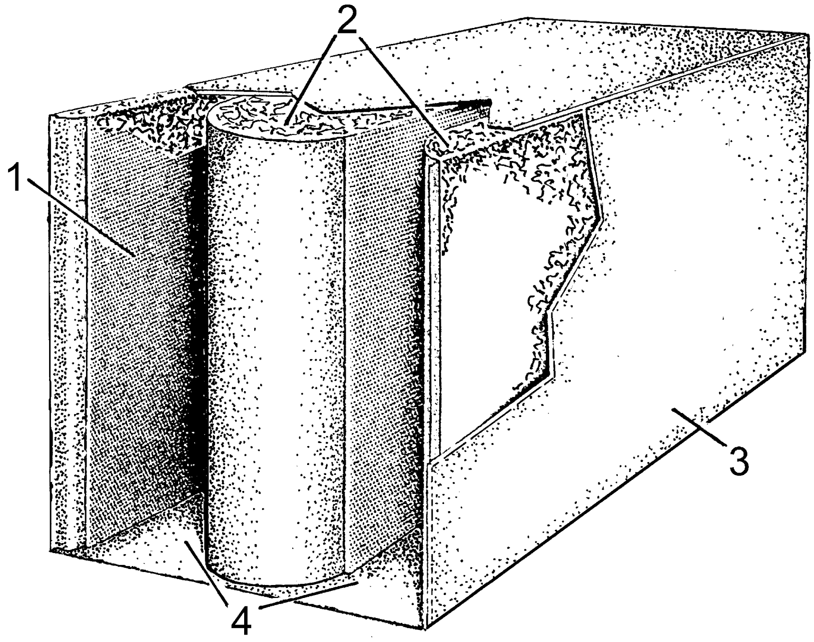

Figure 10.26(c). Parallel baffle rectangular silencer. (1) perforated, galvanized steel; (2) fiberglass or mineral wool fill; (3) sheet metal casing; (4) streamline inlet (Bell, 1982).

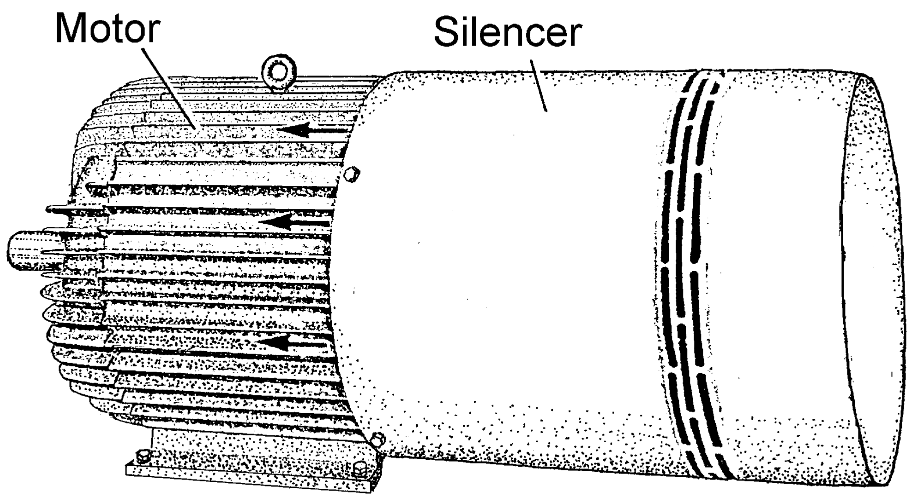

Figure 10.26(d). Electric motor with dissipative muffler (Bell, 1982).

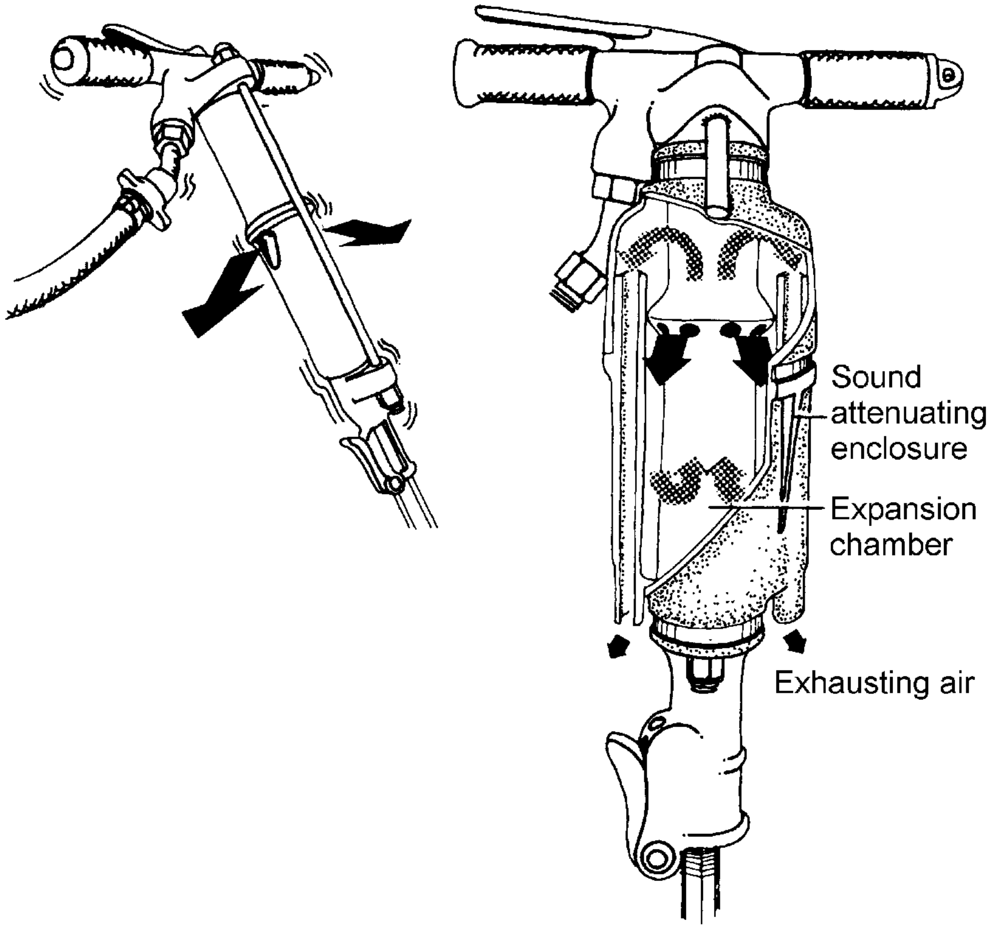

Figure 10.27. An example of a combined reactive/dissipative muffler

The performance of reactive devices is dependent upon the impedances of the source and termination (outlet). In general, a reactive device will strongly affect the generation of sound at the source. This has the effect that the transmission loss and insertion loss of reactive devices may be very different. As insertion loss is the quantity related to noise reduction, it should be used here to describe the performance of reactive muffling devices in preference to transmission loss which unfortunately is commonly used. Insertion Loss values for reactive mufflers vary depending on the design, but are generally in the range of 10 to 30 dB over several octaves.

The performance of dissipative devices, on the other hand, by the very nature of the mode of operation, tends to be independent of the effects of source and termination impedance. Provided that the transmission loss of a dissipative muffler is at least 5 dB it may be assumed that the insertion loss and the transmission loss are the same. This assertion is justified by the observation that any sound reflected back to the source through the muffler will be reduced by at least 10 dB and is thus small and generally negligible compared to the sound introduced. Consequently, the effect of the termination impedance upon the source must also be small and negligible. The Insertion Loss of dissipative silencer increases with the length and varies with the design of the silencer. It can range from 5 dB at low frequencies to 50 dB at high frequencies in typical installations. It is always best to consult manufacturer's data and to remember that larger Insertion Losses usually translate to large pressure drops imposed on any gas flowing through the muffler.

It is possible to design both reactive and dissipative mufflers to achieve desired noise reductions in specific applications and procedures for doing this are outlined by Bies and Hansen (1996, Ch. 9), where the design of sound reducing plenum chambers is also discussed. The procedures are relatively complex and are not discussed here.

The greater the sound attenuating performance of a muffler, the greater will be the pressure drop any gas flowing through it will experience. This pressure drop can be extremely important in some applications and must be considered in any design. In dissipative mufflers containing sound absorbing material, care must be taken to ensure that the material is not eroded by gas flowing through the muffler. In many cases, a thin plastic skin sprayed on to the face of the material is sufficient, but where relatively high speed flows exist, a higher level of protection may be necessary. This usually takes the form of a lightweight impervious layer (of about 10-20 grams per square metre) against the sound absorbing material and a perforated metal sheet between the impervious material and the duct airway, as illustrated in Figure 10.28.

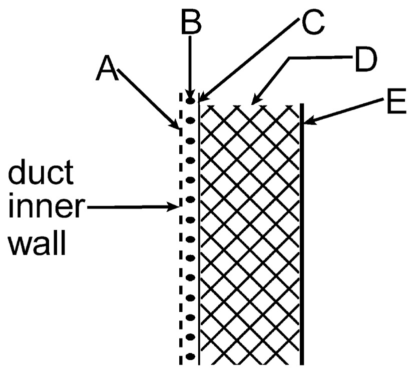

Figure 10.28. Protective facings for duct liners. The elements of the liner are: A, 20 gauge (≈1mm) perforated facing, minimum 25% open area; B, wire mesh spacer, minimum mesh size 12mm, wire diameter minimum 1mm; C, light plastic sheet or fibreglass cloth or fine mesh metal screen; D, fibrous material of specified flow resistance, or unbonded but contained, as in a light plastic bag; E, rigid wall or air cavity backing. Maximum flow speeds up to 8 m s^{-1} do not require A or C. Speeds up to 10 m s^{-1} require that the fibrous material of C be coated or replaced with plastic. Speeds up to 25 m s^{-1} require B and C, while speeds up to 90 m s^{-1} require A, B, C and D. Higher speeds are not recommended. The gap between A and C is ensured by using an open steel mesh spacer, B, (e.g. 2mm wire on 20 mm centres).

10.4.6. Sound absorption and reflection

Sound absorption is the phenomenon by which sound is absorbed by transformation of acoustic energy into ultimately thermal energy (heat). Although some absorption always happens when a sound wave encounters an obstacle, it happens in an appreciable manner when the sound wave is incident on a sound absorbing material.

Sound absorbing materials are fibrous, lightweight and porous, possessing a cellular structure of intercommunicating spaces. It is within these interconnected open cells that acoustic energy is converted into thermal energy. Thus the sound-absorbing material is a dissipative structure which acts as a transducer to convert acoustic energy into thermal energy. The actual loss mechanisms in the energy transfer are viscous flow losses caused by wave propagation in the material and internal frictional losses caused by motion of the material's fibres. The absorption characteristics of a material are dependent upon its thickness, density, porosity, flow resistance, fibre orientation, and the like.

Common porous absorption materials are made from vegetable, mineral or ceramic fibres (the latter for high temperature applications) and elastomeric foams, and come in various forms.The materials may be prefabricated units, such as glass blankets, fibreboards, or lay-in tiles; the material may also be sprayed or trowelled on the surface; or it may be a foam or open-cell plastic. Each type of material has its inherent advantages and disadvantages, and quite often the particular application dictates which form of absorbent material to use. For example, the aesthetics of the environment often prove to be the factor that governs the choice of material. In addition to the acoustical efficiency of the material, one must also consider its cost, installation, maintenance, and resistance to wear and environmental factors.

If fibrous materials such as fibreglass and mineral wool are used it is important to ensure that other health problems arising from human contact with the fibres are avoided. This often means that the material should be enclosed in a thin plastic bag. If the plastic bag is sufficiently thin (20 µm thick polyethylene), the acoustic properties of the acoustic material will be unaffected. As the material is made heavier, the high frequency absorption ability of the material will be degraded, although there will be some improvement at low frequencies. Some manufacturers supply fibrous material which has been sprayed on the surface with a plastic or resin coating which may take the place of the plastic containment bag.

Often there is also a need for mechanical protection to ensure that the thin plastic containment bag remains undamaged. The usual form of protection is a thin sheet of perforated metal or wood. To ensure no effect on the acoustic properties of the material being protected, the perforated sheet should have a ratio of open area (holes) to solid area of greater than 25%. Open area ratios less than this will result in reduced sound absorption at high frequencies, although the absorption at low frequencies will be increased a little.

An important mistake often made in the installation of acoustic materials is to place the perforated sheet in contact with the plastic bag protection. This causes a very severe degradation in performance of the acoustic material and must always be avoided. The simplest way of avoiding the problem is to insert a spacer (usually thin wire mesh with holes at least 15mm in size) between the plastic bag and the perforated sheet (see Figure 10.28).

Acoustic absorbing materials can be rated by their sound absorption coefficients which are frequency dependent and defined as the fraction of incident energy which is absorbed when the incident sound field is diffuse. This is discussed in detail in the specialised literature (NIOSH, 1980; AIHA, 1975; Beranek, 1971; Beranek and Ver, 1992; Bies and Hansen, 1996) and data are usually provided by manufacturers of special acoustic materials. Because different materials have different absorption coefficients for different frequencies, a frequency analysis of the noise to be controlled should be made so that the most suitable materials can be selected.

Table 10.3 presents examples; however, it is always preferable to consult manufacturer's data when using absorptive materials to control reverberant sound fields. Note that the absorption coefficients listed in Table 10.3 are "Sabine" absorption coefficients as distinct from "statistical" absorption coefficients. The difference lies in the way in which the coefficients are measured. "Sabine" absorption coefficients are measured in a reverberation room, while "statistical" absorption coefficients are calculated from the normal incidence absorption coefficient measured in an impedance tube. Because of inaccuracies in the inherent assumptions involved in the measurement of "Sabine" absorption coefficients, values greater than the theoretical maximum of unity are often obtained. When using these values in practice, better sound prediction results are obtained if they are rounded down to one. On the other hand, "statistical" absorption coefficients are never greater than 0.94. For the purposes of predicting the effect of sound absorbing treatment on noise levels in an industrial space, it is probably better to use the "Sabine" absorption coefficient, and as it is always larger than the "statistical" absorption coefficient, it is the one usually quoted by manufacturers of sound absorbing materials. Further details on the measurement and use of absorption coefficients may be obtained from Bies and Hansen (1996, Ch. 7 and App. 3).

Table 10.3. Examples of Sabine absorption coefficients of general building materials
(Collected from the published literature and manufacturer's data)

	Octave-band centre frequency (Hz)					
	125	**250**	**500**	**1000**	**2000**	**4000**
Brick, unglazed	0.03	0.03	0.03	0.04	0.05	0.07
Brick, unglazed, painted	0.01	0.01	0.02	0.02	0.02	0.03
Carpet on foam rubber	0.08	0.23	0.57	0.69	0.71	0.73
Carpet on concrete	0.02	0.06	0.14	0.37	0.60	0.65
Concrete block, coarse	0.36	0.44	0.31	0.29	0.39	0.25
Concrete block, painted	0.10	0.05	0.06	0.07	0.09	0.08
Floors, concrete or terrazzo	0.01	0.01	0.015	0.02	0.02	0.02
Floors, resilient flooring on concrete	0.02	0.03	0.03	0.03	0.03	0.02
Floors, hardwood	0.15	0.11	0.10	0.07	0.06	0.07
Glass, heavy plate	0.18	0.06	0.04	0.03	0.02	0.02
Glass, standard window	0.35	0.25	0.18	0.12	0.07	0.04
Gypsum board 1/2 in.	0.29	0.10	0.05	0.04	0.07	0.09
Panels, fibreglass, 1.5 in.	0.86	0.91	0.80	0.89	0.62	0.47
Panels, perforated metal, 4 in. thick	0.70	0.99	0.99	0.99	0.94	0.83
Panels, perforated metal with fibre-glass insulation 4 in. thick	0.21	0.87	1.52	1.37	1.34	1.22
Panels, perforated metal with mineral fibre insulation, 4 in. thick	0.89	1.20	1.16	1.09	1.01	1.03
Panels, plywood, 3/8 in.	0.28	0.22	0.17	0.09	0.10	0.11
Plaster, gypsum or lime, rough finish on lath	0.02	0.03	0.04	0.05	0.04	0.03
Plaster, gypsum or lime, smooth finish on lath	0.02	0.02	0.03	0.04	0.04	0.03
Polyurethane foam, 1 in. thick	0.16	0.25	0.45	0.84	0.97	0.87
Tile, ceiling mineral fibre	0.18	0.45	0.81	0.97	0.93	0.82
Tile, marble or glazed	0.01	0.01	0.01	0.01	0.02	0.02
Wood, solid, 2 in. thick	0.01	0.05	0.05	0.04	0.04	0.04

One of the indices used to describe a sound-absorbing material is the noise-reduction coefficient (NRC). This is defined to be the arithmetic average of the material's sound absorption coefficients at 250, 500, 1000, and 2000 Hz:

$$NRC = \frac{\bar{\alpha}\,250 + \bar{\alpha}\,500 + \bar{\alpha}\,1000 + \bar{\alpha}\,2000}{4}$$

As such, the NRC is an index of the sound-absorbing efficiency of the material. Absorbing materials by their very nature are effective at reducing reflected sound fields but have little effect on noise transmitted through them, except at very high frequencies.

It should be kept in mind that walls covered with sound absorbing materials do not have the ability to reduce noise from a source. The maximum effect possible in covering walls with absorbing materials is to avoid reflected noise (which at best is equivalent to having no walls), and this measure therefore has little effect when the operator is close to the source or when reflected noise is not an important component of the total noise to which workers are exposed.

In practice, the importance of the reflected noise component can be estimated by measuring noise levels close to the source and then by making successive measurements at increasing distances. If the level does not appreciably drop, the reflected noise component is important; if the noise level drops appreciably with distance, then it is not efficient to recover walls with sound absorbing material. In the absence of any reflecting surfaces, the noise level should drop by 6 dB for each doubling of the distance from the noise source.

10.4.7. Reverberation

When sound reflects within boundaries, it "accumulates" as a result of the addition of the reflected sound to the original sound. Sound may continue even after the original source stops - this is called "reverberation".

Thus a reverberant field is one which is characterised by sound which has been reflected from at least one surface in a particular room or enclosure. When the enclosure boundaries are hard and reflective, the reverberant field can easily dominate the sound arriving directly (without reflection) from a particular sound source and this will become increasingly likely as the distance from the sound source is increased. The reduction of noise with distance from the source in a reverberant field is illustrated in Figure 10.29 (with distance, r in metres) for varying degrees of reverberation characterised in terms of a "room constant" which is a measure of the sound absorbing characteristics of a room and is expressed by the following equation:

$$R = \bar{\alpha} S / (1 - \bar{\alpha}) \tag{8}$$

where:

S = the total area of the boundaries of the room (m^2)

$\bar{\alpha}$ = the average absorption coefficient of the surfaces of the room at a given frequency. At high frequencies, absorption due to the air in the room must also be included in the calculation (see Bies & Hansen, Ch. 7, 1996).

In practice, all boundaries do not have the same acoustical absorption characteristics and the average absorption coefficient $\bar{\alpha}$ for i surfaces must be computed using

$$\bar{\alpha} = \sum_{i=1} S_i \alpha_i / S \tag{9}$$

where:

α_i = the absorption coefficient of surface i and

S_i = the area of surface i in the room (with corresponding coefficient α_i).

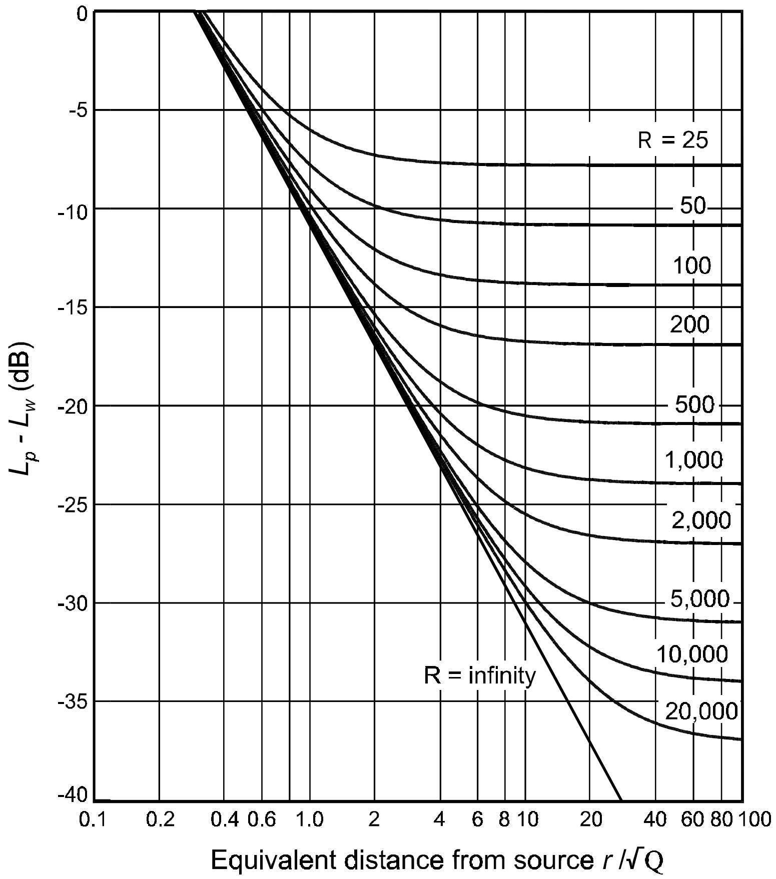

Figure 10.29. Sound attenuation in a reverberant field

The product $S\bar{\alpha}$ is called the absorption of the room and has the units m². The units are sometimes called sabines (m^2).

In a reverberant field (which exists in most indoor situations), the mean square sound pressure at a given point is the sum of the mean square sound pressures of the direct sound waves and of all reflected sound waves. At a distance r from the source, the mean square sound pressure squared can be mathematically expressed by the following simplified equation (note that the equation is approximate only and more accurate analyses of factory noise can be made using finite element or boundary element analyses):

$$p^2 = W\rho c\left(\frac{Q}{4\pi r^2} + \frac{4}{R}\right) \qquad (10)$$

where:

- Q = the directivity factor of the source towards the measuring point (see Ch. 1)
- r = distance from the source (in meters)
- R = room constant
- ρ = density of air (1.21 kg/m^3)
- c = speed of sound in air (343 m/s at 20°C).

The preceding expression can be written approximately (to within 0.2 dB), in terms of levels, as follows:

$$L_p = L_w + 10\log_{10}\left(\frac{Q}{4\pi r^2} + \frac{4}{R}\right) \tag{11}$$

This relation is shown graphically in Figure 10.29. This graph allows the determination of attenuation in sound pressure level, with increasing distance from the source (in meters). There are different curves for different room constants, R.

In the case of a room for which the length width and height dimensions are not different by more than a factor of 10, the relative strength of the reverberant sound field may be compared with the direct field produced by a machine at a particular location by comparing the direct and reverberant field terms (the bracketed quantities of equation (11)); that is, $4/R$ and $Q/4\pi r^2$. When these two terms are equal, the direct and reverberant sound fields are equal (see Chapter 1, section 1.1.1 for definitions of direct and reverberant fields. The case of flat rooms or long rooms is a little different and is discussed in detail in Bies and Hansen (1996, Ch. 7).and is part of ISO 11690. Besides the reverberation time in this standard other parameters are defined and related to the spatial sound distribution curve which can be determined and verified with affordable means (ISO 14257). Some national legal provisions require acoustic quality of workrooms specified by parameters given by ISO 11690-1,-2 and -3.The prescribed values are related to the typical values of ISO 11690-2.

Example *(approximately cubic room)*:

Consider a room 10 meters long, 8 meters wide and 4 meters high, which has a ceiling covered with a material having an absorption coefficient of 0.7, while it is 0.2 and 0.05 for the floor and the walls respectively (at 500 Hz).

The products $S_i\alpha_i$ for ceiling, floor and walls are then respectively 56, 16 and 7 m^2. The equivalent absorption area A is the sum of these three quantities and is equal to 79 m^2. The total room surface area is 2(10×8 + 10×4 + 8×4) = 304 m^2. Using equation (9) then gives the average absorption coefficient $\bar{\alpha}=0.26$. Equation (8) then gives the room constant R = 107 m^2. If the sound power level of a source is 100 dB and the source is placed on a reflecting surface (Q = 2), use of equation (11) gives the sound pressure level, L_p, at 3 meters from the source in that room as 87.5 dB. In free field, R in equation (11) is equal to infinity, Q is equal to 1 and equation (11) gives L_p = 82.5 dB. In the purely reverberant field, r in equation (11) is sufficiently large that the first term in brackets of equation (11) can be ignored with the result that L_p = 86 dB.

The calculation of the room constant, R, in real occupational situations, is usually not very accurate, particularly in the industrial environment, as the absorption of the machines, pipes, etc., is almost impossible to take into account. However, it can be measured by measuring the reverberation time, T_{60}, of the indoor work area.

Reverberation time is the time (in seconds) required for the sound pressure level in an enclosed space to decay by 60 dB when the sound source is switched off. It is related to the Sabine absorption coefficient as follows:

$$T_{60} = \frac{55.25\,V}{Sc\bar{\alpha}} \tag{12}$$

where:

V is the room volume in m^3

S is the surface area of all surfaces and objects in the room.

If it is too difficult to estimate S, then the room constant is often approximated by substituting R for $S\bar{\alpha}$ in the above equation.

c=speed of sound in air as given for equation (10)

NOTE: The equation T60 = 0.163 V/A with A=S$\bar{\alpha}$ as given in ISO 11690-2, Annex F, is accurate to within 2% for temperatures between 15 and 25 °C.

If the reverberant sound field dominates the direct field, then the sound pressure level will decrease if absorption is added to the room or factory. The decrease in reverberant sound pressure level ΔL_p to be expected for a particular increase in sound absorption expressed in terms of the room constant R (see equation (11)) may be calculated by using equation (11) with the direct field term set equal to zero. The following equation is obtained where R_i is the initial room constant and R_f is the room constant after the addition of sound absorbing materials.

$$\Delta L_p = 10 \log_{10}\left[\frac{R_f}{R_i}\right] \tag{13}$$

It can be seen from the preceding equation that if the original room constant R_i is large then the amount of additional absorption to be added must be very large so that $R_f \gg R_i$ and ΔL_p is significant and worth the expense of the additional absorbent. Clearly it is more beneficial to treat hard surfaces such as concrete floors which have small Sabine absorption coefficients, because this will have greatest effect on the room constant.

Remember that when calculating the increase in room constant due to fixing absorbing material to an existing surface, the difference in absorption coefficient between the existing and new surface should be used together with equations (8) and (9). In many cases best results are obtained by increasing the room constant by hanging sound absorbing panels from the ceiling.

It is highly undesirable to use hard reflective materials as boundaries for a space where there are noise sources and occupants. In fact, even if there are no occupants, it may not be desirable to let such noise build up as this will increase the noise which escapes through the enclosing walls. So, even if noise is "closed in", there is interest in decreasing it. In fact, it should be kept in mind that even if isolation is used, reduction at the source should not be overlooked.

The same concept applies to the isolation of a noisy area inside a workplace, in which case workers must wear ear protection as part of a hearing conservation program. In this case, it is particularly important to treat the insides of the isolating walls with sound absorbing materials (see below) to avoid or reduce the reflected noise component which would add to the exposure of the workers inside the area.

10.4.8. Active noise control

Active control of noise is the process of reducing existing noise by the introduction of additional noise by means of one or more secondary (or control) noise sources. The introduced noise may achieve the required noise reduction by way of any one or combination of three different physical mechanisms.

One mechanism which is often used to describe the active control of noise in the popular press is that of sound field cancellation; that is, the introduced control sound is anti-phase to the original sound and cancellation results. This mechanism characterises cases where noise reduction is achieved in small local areas surrounding a control source; however, local areas of cancellation are always balanced by other areas of reinforcement where the sound level is increased. This type of control mechanism, which may be called "local cancellation", characterises the process involved in the control of noise around a passenger's head in an aircraft or motor vehicle using a loudspeaker embedded in the head rest of the seat or by use of a headset or earmuff containing a loudspeaker.

A second mechanism, which will be called suppression of sound generation, is possible and may be understood on the basis of the following considerations. If it were possible to make the entire control sound field (or almost all of it) 180° out of phase with the original (primary) field, then the sound radiated by the primary source would be effectively "cancelled" leaving one to wonder where all the energy had gone. The answer is that in this case, the control mechanism is not really cancellation; the sound field generated by the control sources has effectively "unloaded" the primary source, changing its radiation impedance so that it radiates much less sound (even though the motion of the physical source such as a vibrating surface may remain unchanged). In this case, the control sources act to suppress the sound power radiated by the primary source by making its radiation impedance reactive with only a negligible real part.

To achieve effective suppression of the primary source output by presenting a purely reactive impedance to it, the control sources must be large enough and located such that they are capable of presenting the required impedance to the primary source. In one dimensional wave guides, such as air conditioning ducts, these constraints are relatively easy to satisfy and the distance between the control and primary sources is not too important. However, in 3-D space, the control source in general will need to be close to the primary source to affect its radiation impedance significantly. It will also need to be of similar size with a similar volume velocity output.

A third mechanism of active noise control is that of absorption by the control sources. In this case, the primary sound field energy is used to assist in driving the control source (for example the speaker cone if the control source is a loudspeaker). However, the acoustical efficiency of loudspeakers and other artificial noise generators is so poor, that electrical energy is still needed to drive the source with sufficient amplitude and at the correct phase to enable it to absorb energy from the sound field. Except for plane wave sound propagation in ducts, this mechanism is likely to result only in areas of reduced noise close to the control source.

Feedforward and feedback control are the two main approaches which have been used in the past for active noise control. A feedforward controller requires a measure of the incoming disturbance sufficiently far ahead in time that it can be used to generate the required control signal for the control source. This type of control is ideal for periodic noise or for random noise propagating in ducts. An example of such a system is illustrated in Figure 10.30. Note that for the controller to remain stable, a measure of the cancellation path electroacoustic transfer function (from loudspeaker input to microphone output) is necessary. This is typically done on-line using low level random noise as illustrated in Figure 10.30.

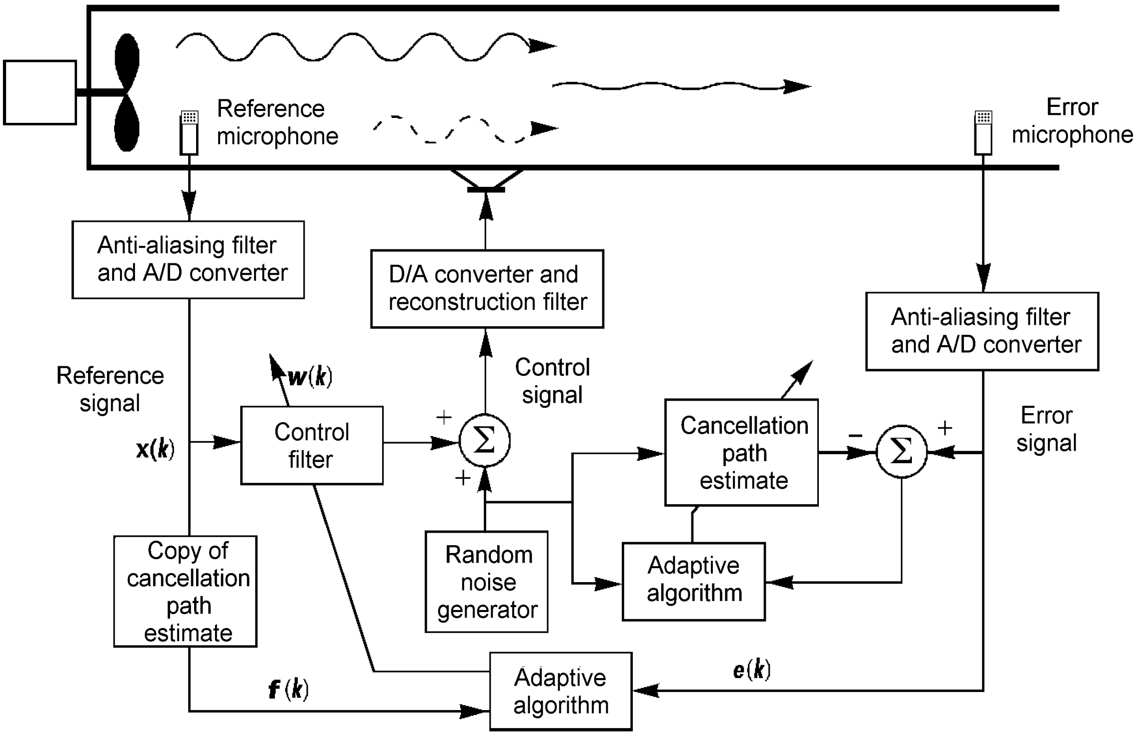

Figure 10.30(a). Configuration of a feedforward active noise control system to attenuate noise propagation along a duct (after Eriksson and Allie, 1989).

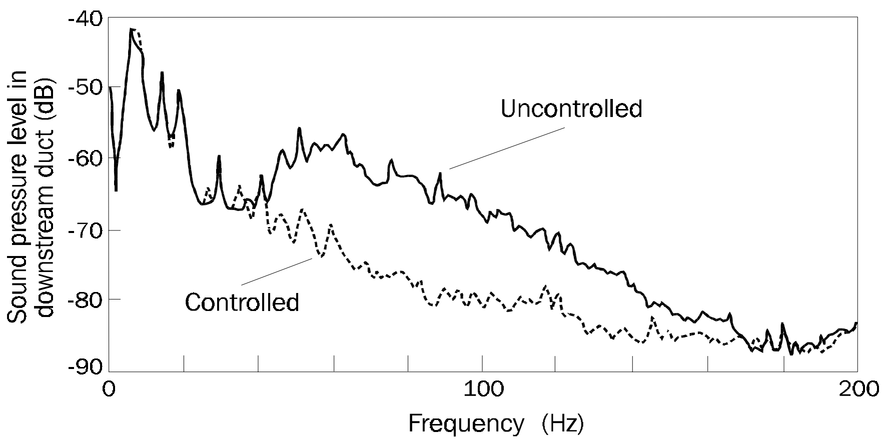

Figure 10.30(b). Attenuation of the feedforward systemof figure 10.30(a) achieved for broadband sound (after Eriksson and Allie, 1989).

Practical implementation of the system is much more complex than shown in the figure because allowance must be made for quantisation errors associated with the digital nature of the controller and the electro-acoustic delay between the controller signal input to the control sourceand the signal output from the error microphone. This is discussed in more detail in specialist books on the subject (Nelson and Elliott, 1992; Hansen and Snyder, 1996).

Feedforward controllers generally use a digital filter to act as an inverse model of the system to be controlled, with the measure of the incoming disturbance being passed through the digital filter and then to the control source. Practical systems are adaptive so that they can cope with changes over time of physical parameters such as temperature, speed of sound, and transducer contamination. Adaptation is achieved by using an error sensor, which detects the residual sound field after control, to provide a signal to a control algorithm which adjusts the weights of the digital filter.

When sound propagating in ducts is to be attenuated, the elements of the active system are usually sufficiently small to be mounted in the duct wall, thus minimising air flow pressure losses. Disadvantages of active attenuators are associated with their cost (although this is rapidly decreasing), the need for regular maintenance (speaker replacement every three to five years) the requirement for custom installation and testing by experts, the reduction in performance at mid to high frequencies and a requirement for a separation between the reference microphone and control loudspeaker of a minimum of 1m at 150 Hz to 10m at 20Hz.

A feedback controller requires no knowledge of the incoming disturbance and acts to change the system response by changing the system resonance frequencies and damping. To be effective, relatively high gains in the feedback loop are necessary which makes this type of controller prone to instability if any parameters describing the physical system change slightly. However, this type of controller is ideal in cases where it is not possible to sample the incoming disturbance or for random noise. To minimise acoustic delays and thus maximise system stability, the physical locations of the control source and error sensor should be as close together as possible.

Examples of the practical use of a feedback controller include active ear muffs (or active head sets - see Figure 10.31), active vehicle suspension systems and active control of structural vibration. Feedback controllers, however, are unsuitable for controlling travelling acoustic waves in ducts (where reflection from the end is negligible) or flexural waves in structures where no reflections are involved. However, in cases where reflections are involved, the damping introduced by the feedback controller minimises the transient or reverberant response of the acoustic or structural system and as such can be quite effective. An example of a feedback system to control noise propagating in a duct is illustrated in Figure 10.32.

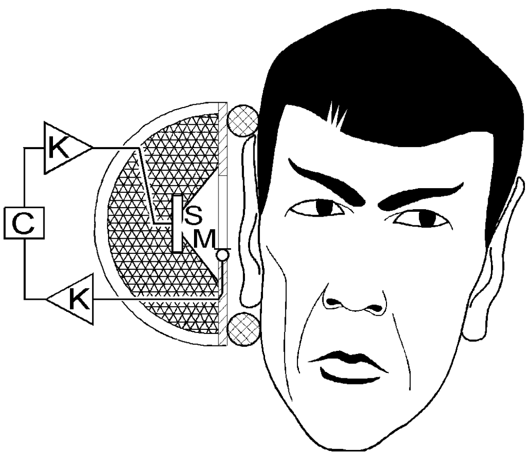

Figure 10.31. Feedback control system applied to a headset.

C = digital filter;
K = amplifier;
M = microphone;
S = loudspeaker.

It is important to discuss limitations on applications of active noise control. It is very cost effective and beneficial in some very specific applications, but it is definitely not the all encompassing answer to a wide range of noise problems which will become available just as soon as the cost of the electronic hardware falls low enough. Unfortunately active noise control is limited in application for physical reasons, and not because of limitations in electronic processing power.

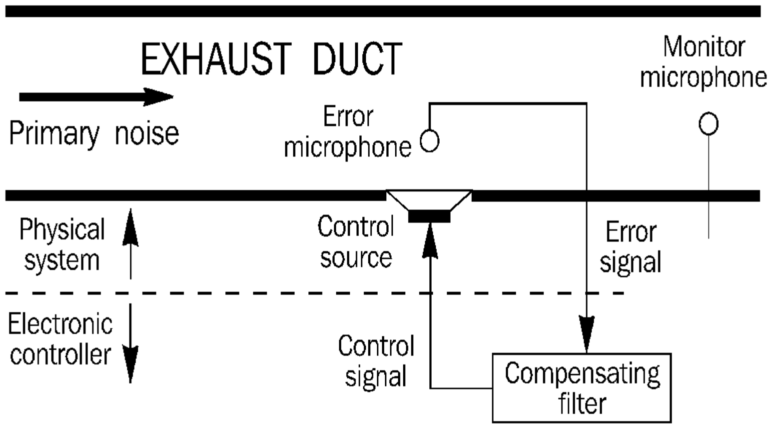

Figure 10.32. Feedback control system applied to sound propagating in a duct.

Active noise control is most suitable for low frequency tonal sound fields such as radiated by electrical transformers or exist in a propeller driven aircraft cabin. Even for tonal sound fields, a large number of control sources and error microphones are needed to make the systems effective. When the sound is confined to a duct and propagates as plane waves, broadband noise can be controlled actively as well using one or two control sources and error sensors, provided that an acoustic reference signal can be obtained sufficiently far upstream of the control source for the control system to generate the required control signal. In this case about 15 to 20 dB of noise reduction may be expected over 2-3 octave bands. However performance is usually reduced in the presence of large air flow speeds. In very small enclosures (smaller than a wavelength of sound at the highest frequency of interest), broadband and pure tone noise can both be controlled. In larger enclosures (and in free space or outdoors), the control of random noise is not practical. Thus active earmuffs are useful for frequencies below about 1500 Hz and a number of systems are commercially available. However it is not practical to use active noise control to reduce general broadband factory noise in the vicinity of workers.

A more detailed discussion of active noise control can be found in specialist books on the subject (Hansen and Snyder, 1996).

10.4.9. Separation of source and receiver

Another type of noise propagation control is the separation, which can be by distance or in time. As the direct field radiated by a source generally decreases by 6 dB for each doubling of the distance from it (after the initial 1 metre), separating the source and receiver by distance is beneficial.

Noisy operations can also be separated in time; that is, they are performed out of the usual shift.

10.5. RECEIVER CONTROL

Receiver control in an industrial situation is generally restricted to providing headsets and/or ear plugs for the exposed workers, see chapter 11. It must be emphasised that this is a last resort treatment and requires close supervision to ensure long term protection of workers' hearing. The main problems lie in ensuring that the devices fit adequately to provide the rated sound attenuation and that the devices are properly worn. Extensive education programs are needed in this regard. Hearing protection is also uncomfortable for a large proportion of the workforce; it can lead to headaches, fungus infections in the ear canal, a higher rate of absenteeism and reduced work efficiency. It is worth remembering that the most protection that a properly fitted headset/earplug combination will provide is 30 dB, due to conduction through the bone structure of the head. In most cases, the noise reduction obtained is much less than this.

Another option which is sometimes practical for receiver control is to enclose personnel in a sound reducing enclosure (see ISO 11957). This is often the preferred option in facilities where there are many noisy machines, many of which can be operated remotely. In this case, the enclosure design principles outlined in section 10.3.5 may be used and the enclosure performance may be calculated using equation (6) and the appropriate wall material and construction selected after the required noise reduction has been established. Guidelines which should be followed during design and construction are:

- doors, windows and wall panels should be well sealed at edges;
- interior surfaces of enclosure should be covered with sound absorptive material;
- all ventilation openings should provided with acoustic attenuators.

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INTERNATIONAL STANDARDS

Titles of the following standards related to or referred to in this chapter one will find together with information on availability in chapter 12:

ISO 4871; ISO 9613-2; ISO 10846; ISO 10847; ISO 11654; ISO 11689; ISO 11690-1, -2, -3; ISO 11820, ISO 11821; ISO 11957; ISO 14163; ISO 14257; ISO 15667; ISO/TR 11688.

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