Z = ρ x c

where z is the acoustic impedance (for air =415 rayls) and ρ is the density of the medium. For other types of waves eg standing waves the relationship is more complicated and is best described mathematically in terms of complex numbers.

For absorption the wave must be able to get into the material. The pores must be big enough to let in the wave but not too large to reduce friction. The former is a way of saying that air and the material must have similar acoustic impedances. A large impedance mismatch ensures that the sound wave is mainly reflected. In practice an absorber with an acoustic impedance of two or three times that of air is used which for mineral fibre is obtained with a density between 40 - 100 kg/m^3.

Previously it was described how a partial standing wave is formed against a wall. For a hard surface and a sound wave incident normally to the surface , there is a particle velocity node against the wall and an antinode at a distance of one quarter wavelength from the wall. This means that there is zero movement of air at the wall surface (no friction, no absorption) and maximum movement of the air at a quarter of a wavelength from the wall (maximum friction, good absorption). Therefore the absorber should be at least as thick as a quarter of a wavelength to get the best absorption and relatively thin layers would have very poor absorption. Of course we have only considered normally incident waves and ignored other effects but the principle applies. Thin materials only become good absorbers when placed away from walls. That is why acoustic tiles in suspended ceilings are much better absorbers than when stuck directly on the ceiling.

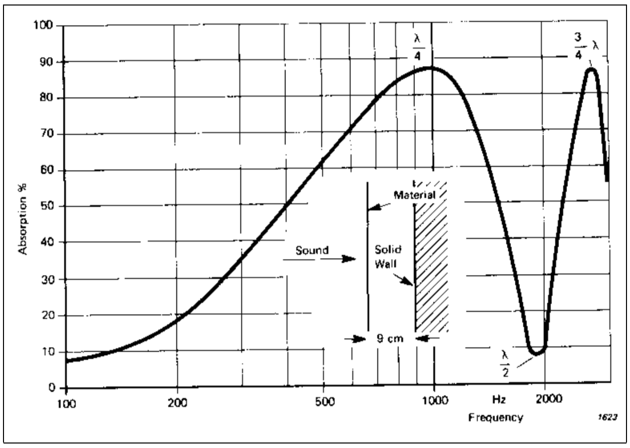

Fig 7 Absorption
characteristics of an absorptive material mounted 90 mm from a solid wall (Source ref 2)

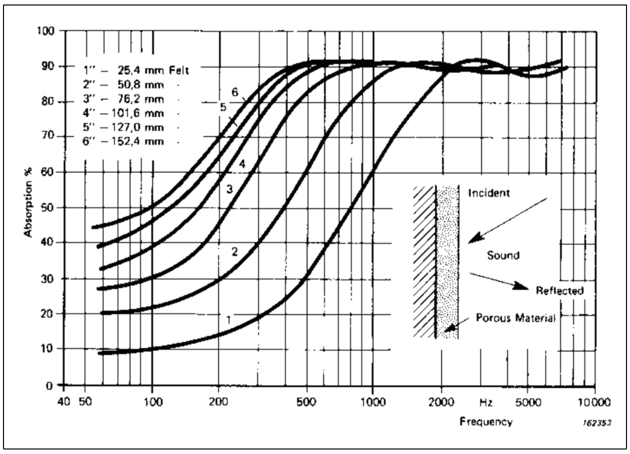

Fig 8 Dependence of the absorption of a porous materail on the thickness of the material (Source ref 2))

Panel or membrane Absorbers

A wood panel with an air space between it and the wall will act as a resonance system absorbing sound. The air space is the spring and the wood panel the mass attached to it so it resembles a weight hanging on the end of a spring and has a natural frequency of oscillation. Sound is absorbed at this natural frequency setting the system into resonance. The narrower the air gap the stiffer the spring and the higher the natural frequency of the system. The resonant frequency is given by the expression: $f = 60/(md)^{½}$ where

m is the superficial mass or surface density of the panel in kg/m2
and d is the air gap in m.

This is a reduction of the expression for a double panel resonating system :

$$f = 60\ \{(m_1 + m_2)/m_1m_2d\}^{1/2}$$

where m_1 and m_2 are the masses of the panel. This frequency is important in the sound insulation of multiple partitions and is known as the mass-air-mass resonant frequency.

For a typical plywood panel (10 mm) with a 50mm air gap this resonant frequency works out about 100 Hz. Practical panel absorbers absorb around the resonant frequency, with in the absence of friction, theoretical perfect absorption at the resonant frequency. However stiff panels reradiate sound limiting the absorption coefficient to about 0.3 at the resonant frequency. This can be slightly improved by placing sound absorber in the air space to damp the air vibrations but it is more effective to damp the panel by addition of heavily internally damped materials such as bitumen roofing felt. Note that for damping to be successful the damping material must be heavy compared to the material being damped. Successful damped materials for machinery casings have been made however using a thin visco-elastic layer between two alloy sheets. These are known as sandwich materials.