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Noise of Electrical Machines

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Noise of electrical machines

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The control of noise in electrical machines may be effected by two separate and distinct methods. The first is the reduction of the noise at the source and the second the attenuation of this noise before it reaches the ears of the observer. Since noise reduction is costly, the problem is resolved into one of finding the most economic means of obtaining the required noise reduction, without interfering with the efficient functioning of the plant.

It is generally considered that a good machine should run smoothly and quietly, so from the earliest days the designers and manufacturers have put a great deal of effort into producing quiet machines and more recently considerable research effort has been expended in explaining the basic phenomena.

Table 1. Discrete analysis of noise of geared turbine-generator

frequency (Hz)	sound pressure level (dB re. 0·2 nbar)	probable source
125	70	turbine unbalance
199	75	generator fan blade passing frequency
1670	74	generator slot note
3340	61	brush and commutator note
3740	75	gear contact note
6000	71	gear phantom note

In many electrical machines it is found that several fundamental causes are responsible for the over-all noise. These include magnetostriction in the steel, variations of the magnetic pull across the air gap (which is often caused by the effects of slots) windage noise, and mechanical noise in bearings, gears and other parts. This is illustrated by the analysis shown in table 1 of a geared turbine generator where six separate major sources have been identified. The noise radiated by any of these sources may be magnified by resonances and it is indeed fortunate that the resonant magnification is usually low as a result of the methods of construction normally employed.

It is interesting to note that to reduce the loudness of a machine to one-half would require a reduction of 10 phon. Thus the radiated energy would require to be reduced to one-tenth, a very formidable task by direct methods when all the obvious precautions have already been taken, but comparatively easy by enclosure.

Large a.c. generators

In these machines the predominant source of noise is frequently the magnetic hum resulting from the rotating field. Magnetic noise generation similar to that in the electrical transformer takes place since the steel in the core laminations is subjected to cyclic reversal of magnetic flux. The result is radiation of noise at 100 Hz and its harmonics. Since, unlike the normal power transformer, the spaces between the laminations are not liquid filled there is a tendency for these to rattle unless precautions are taken to ensure tight axial clamping.

Mechanical and magnetic out-of-balance cause vibration of the core as a whole at a frequency corresponding to the running speed and although in many modern designs the core is suspended on some form of springs (see figure 1) these vibrations are transmitted to the outer shell and radiated as noise.

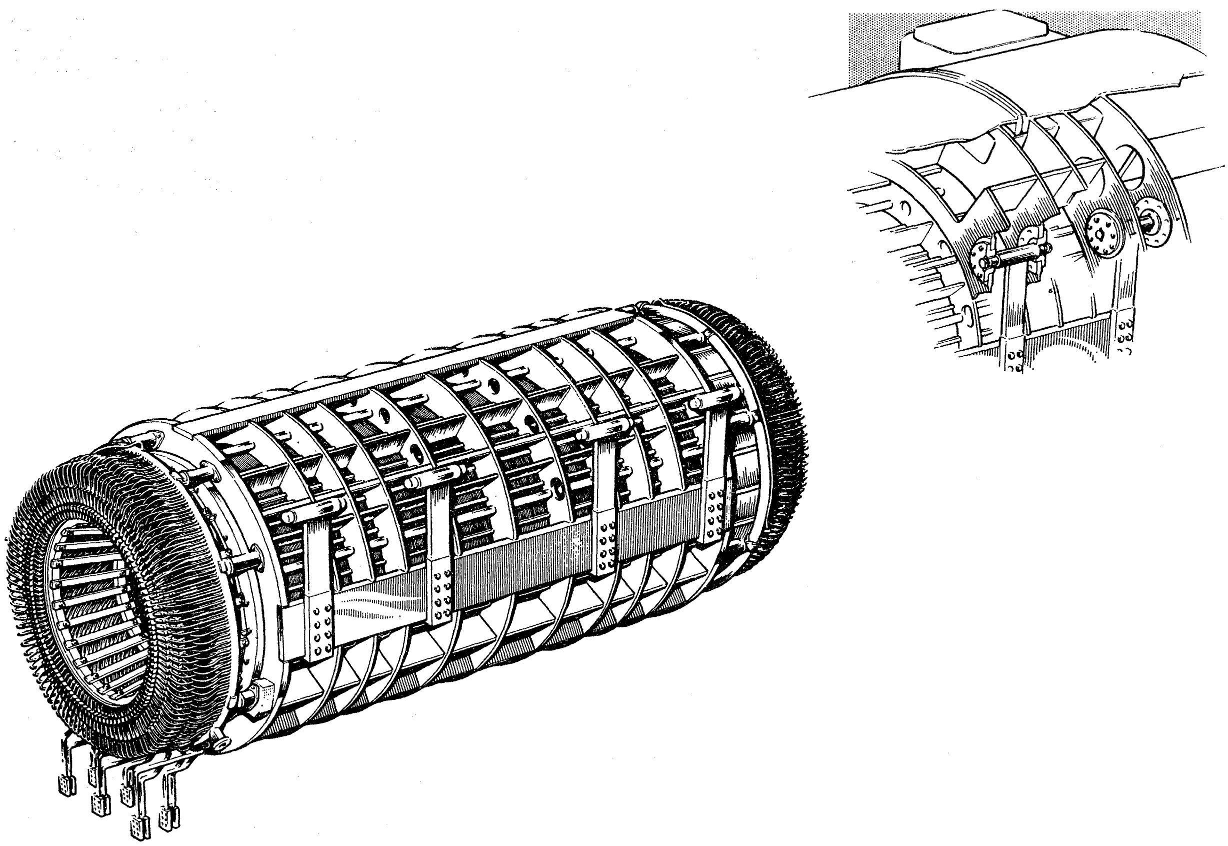

FIGURE 1. Stator core of large a.c. generator.

The noise produced by the fans and other aerodynamic sources is often a major contributor especially in air-cooled machines. Even in hydrogen-cooled machines, where the casings are thick and hermetically sealed, the aerodynamic noise resulting from the high surface velocity of the external shaft, slip rings and couplings may be appreciable. Ventilation noise from the exciters too is often a major contributor.

D.C. MOTOR NOISE

With a d.c. motor the main magnetic noise is caused by the variation in flux as the armature slots enter and leave the magnetic fields under the poles. Usually the most serious effect is the rocking of the pole pieces which in turn cause vibration of the magnetic yoke. This vibration may be magnified by resonance resulting in excessive noise at particular speeds. The noise can be reduced by skewing the armature slots, by tapering

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off the leading and trailing tips of the pole pieces, and by avoiding yoke resonances in the running range.

Commutator and brush noise is often troublesome in the d.c. machine. This is caused mainly by the impulsive excitation of the brush and its holder by the leading edges of the commutator segments, particularly where the commutator surface is not sufficiently smooth. Ventilation and bearing noise can also be troublesome as in the case of the induction motor.

By careful design and manufacture using total enclosure and sleeve bearings a great deal can be done to produce a quiet d.c. machine, but it will be considerably more expensive than the conventional motor, and some form of acoustic hood may provide a more economic solution.

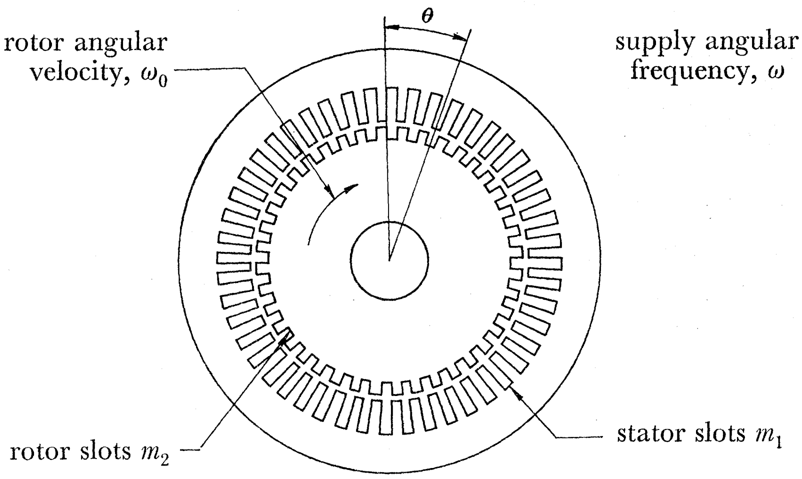

FIGURE 2. Three-phase induction motor ($2p$ poles). Net M.M.F. contains terms periodic in ωt, $p\theta$, $p\phi$, $m_1\theta$, $m_2\phi$; gap permeance contains terms periodic in $m_1\theta$, $m_2\phi$ (where $\phi = \theta - \theta_0 - \omega_0 t$).

INDUCTION MOTOR NOISE

The most commonly used electric motor in industry is the three-phase induction motor in which a rotating field is set up by the stator which reacts with currents in the rotor to produce mechanical power. Machines of this kind are used because of their general reliability of operation, and sizes range from fractional to thousands of horsepower, being most frequently encountered for the driving of pumps, fans, compressors, machine tools and other machinery.

Noise due to magnetic causes is again found to be the principal contributor and this source will be considered in more detail.

Ball and roller bearing noise can sometimes be troublesome and mechanical and electrical unbalance can introduce low-frequency components. In many cases forced draught ventilation used to cool the machine is a major source of noise.

Under constant speed running conditions a rotating field of angular velocity ω/p is set up by windings inserted in the slots on the stator (figure 2), and the interaction of this field upon the windings in the rotor slots (which in a squirrel-cage machine are simply conducting bars short-circuited together at their ends) imposes upon the rotor a torque, resulting in this rotating at angular velocity ω_0, which is always less than ω/p.

The principal causes of noise peculiar to induction motors are those due to the magnetic forces set up in the air gap, which act upon the stator annulus and cause this to vibrate radially. For example, the magnetomotive force (m.m.f.) distribution due to the applied stator currents is periodic not only in time but also spatially around the gap perimeter, being dependent primarily upon the machine pole number and the total number of stator slots. Similarly, because currents are induced in the rotor at slip frequency (Alger 1965) a magnetomotive force distribution due to these currents is set up which is periodic, relative to the rotor, at slip frequency with respect to time, and rotor slot number with respect to angular position.

Relative to the stator therefore the net m.m.f. due to the combined effects of stator and rotor currents contains terms periodic in ωt, $p\theta$, $p\phi$, $m_1\theta$ and $m_2\phi$ (see figure 2). The flux density in the air gap is dependent upon this m.m.f. and also upon the gap permeance, which because of the slotting contains terms periodic in $m_1\theta$ and $m_2\phi$. The resultant force in the air gap, acting upon the teeth of both stator and rotor, is proportional to the square of the flux density and contains a steady term plus terms periodic in both angular position and time.

The oscillatory force acting on any particular stator tooth may be resolved into tangential and radial components. The tangential component, which causes the tooth to bend as a cantilever about its root, while being contributory towards the motor torque, causes only localized deformations of the stator ring and is relatively unimportant so far as noise is concerned. The radial component, however, when compounded with the radial components of adjacent teeth, causes more general deformations of the stator ring, resulting in radiated noise occurring from the external surfaces.

The net radial force distribution around the gap perimeter may be shown to consist of a series of individual force waves (Lehmann 1961), each having its own particular frequency, wavelength and propagation velocity around the gap (Redfearn 1936; Carter 1932). The 'pole number', that is the number of maxima a given wave possesses around the gap perimeter defines the potency with which this wave can excite the stator ring into vibration. Generally, a wave with a low pole number will excite stator vibration more readily than a wave with a high pole number since in the former case the angular distance between maxima is greater.

The largest stator frame vibrations generally occur due to the force wave which is periodic at twice the supply frequency. This particular wave, which results from the main rotating field is common to both induction motors and alternators (Walker & Kerruish 1960), and its magnitude is not dependent upon how many slots there are on the stator and rotor. However, more troublesome components in induction motors are often those associated with rotor slot frequency. In table 2 the general forms of three of the most potent force waves of this nature are given. It can be seen that the frequencies of these occur at rotor slot minus twice supply frequency, rotor slot frequency and rotor slot plus twice supply frequency respectively. The pole numbers of these waves can be low if the rotor and stator slot numbers are nearly equal to each other. It is therefore desirable to so choose m_1 and m_2 that the resulting values of the pole numbers (as given in table 2) are as large as practicable.

Quiet machines are usually obtained when the rotor slot number is chosen so that

$m_2 = m_1 \pm 4p$, except for two-pole machines. In this case the best results are obtained when $m_2 = m_1 - 4p$, i.e. $m_2 = m_1 - 8$ (Alger 1965). With a squirrel cage machine an odd number of rotor slots must always be avoided (Jordan & Rothert 1953).

A typical example of the vibration spectrum of an induction motor is given in figure 3. In this case the radial vibration at a point on the side of a 200 h.p. 2-pole 3-phase motor was measured, with the machine running on no load. In this particular machine there were 48 stator and 40 rotor slots so that the pole numbers of the above three components were 10, 8 and 6 respectively.

Table 2. Three-phase induction motor: general forms of radial force distributions in gap; most potent slot-generated force waves

$$F_1 = \bar{F}_1 \cos[(m_1 - m_2 + 2p)\theta + (m_2\omega_0 - 2\omega)t]$$

$$\text{excitation frequency} = \frac{m_2\omega_0 - 2\omega}{2\pi}$$

$$\text{‘pole number’} = (m_1 - m_2 + 2p)$$

$$\text{‘propagation velocity’} = \frac{\partial\theta}{\partial t} = -\frac{m_2\omega_0 - 2\omega}{m_1 - m_2 + 2p}$$

$$F_2 = \bar{F}_2 \cos[(m_1 - m_2)\theta + m_2\omega_0 t]$$

$$\text{excitation frequency} = \frac{m_2\omega_0}{2\pi}$$

$$\text{‘pole number’} = (m_1 - m_2)$$

$$\text{‘propagation velocity’} = \frac{\partial\theta}{\partial t} = -\frac{m_2\omega_0}{m_1 - m_2}$$

$$F_3 = \bar{F}_3 \cos[(m_1 - m_2 - 2p)\theta + (m_2\omega_0 + 2\omega)t]$$

$$\text{excitation frequency} = \frac{m_2\omega_0 + 2\omega}{2\pi}$$

$$\text{‘pole number’} = m_1 - m_2 - 2p$$

$$\text{‘propagation velocity’} = \frac{\partial\theta}{\partial t} = -\frac{m_2\omega_0 + 2\omega}{m_1 - m_2 - 2p}$$

Since the rotor slot plus twice supply frequency component possessed the lowest pole number, i.e. 6, it was expected that the vibration of the stator ring would be higher for this than for the other two components. This is seen to be the case in curve *A*, the levels of the other two components being correspondingly smaller. A characteristic of this machine was the fact that a comparatively large component also occurred at rotor slot plus four times supply frequency. Non-uniformities of the gap and harmonics in the supply can both account for force waves of low pole number having this frequency.

It is usually found necessary when measuring the magnitudes of the individual slot-generated components to use a highly selective analyser. For example in figure 3, spectrum *A* was obtained by means of a heterodyne analyser having an effective pass band of 8 Hz, which was selective enough to discriminate between specific components. In a corresponding spectrum (*B*) obtained using a constant percentage bandwidth (6 %) analyser, such discrimination was not possible, the single peak at 2200 Hz consisting of the three principal slot-generated components plus a further component at 2400 Hz associated with the number of stator slots. None of these could be identified individually in spectrum *B*.

Two other factors influence the sound radiation due to air gap force waves. First, mechanical resonance of the stator ring at one or other of the exciting frequencies may occur, resulting in excessive radial vibration. Such conditions can be avoided by choosing the radial dimensions so that the ring resonant frequencies (Alger 1965; Carter 1932) are significantly different from the frequencies of any of the principal exciting components.

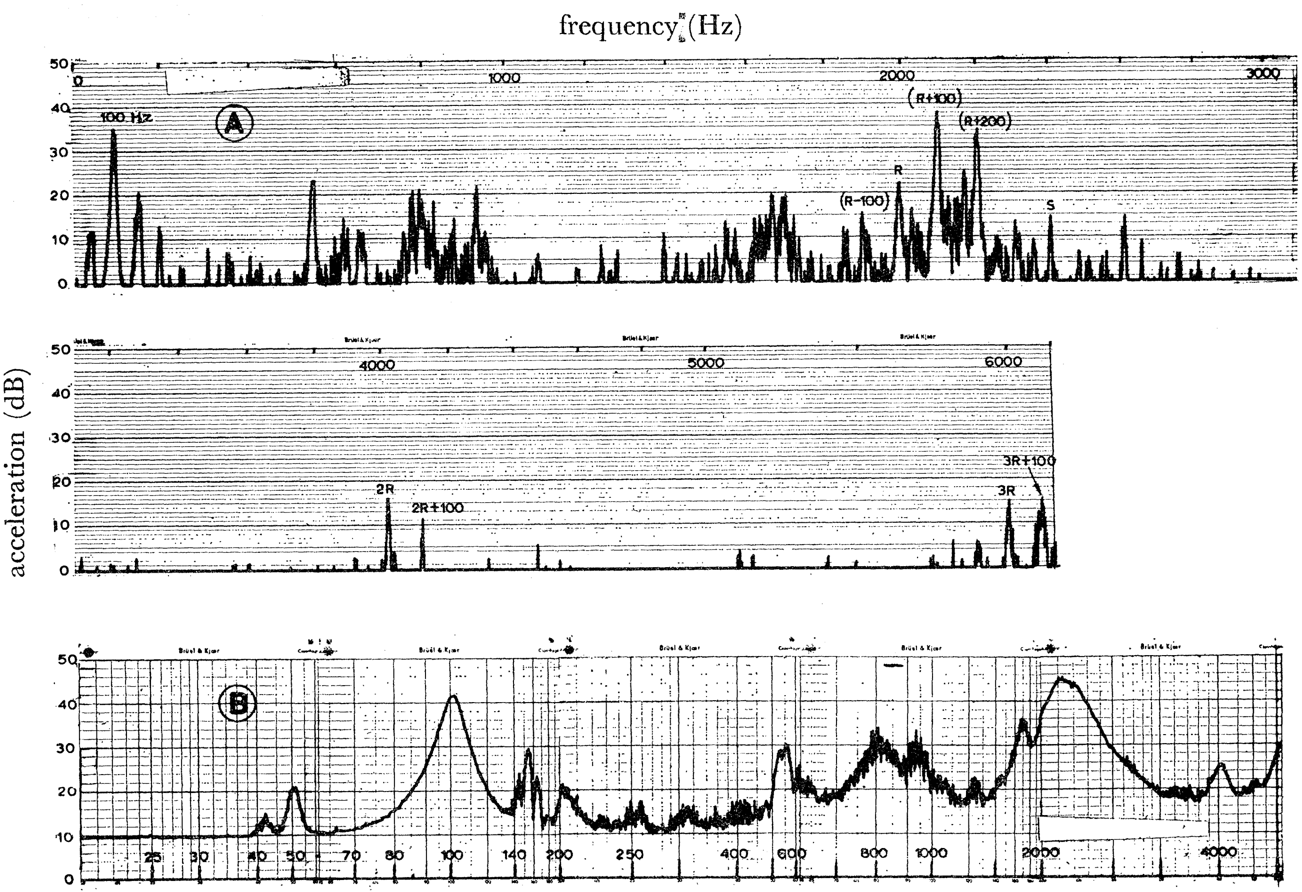

FIGURE 3. 200 h.p. 3-phase 2-pole induction motor, stator frame radial vibration. Comparison of frequency spectra obtained with analysers of differing selectivities: A, heterodyne analyser (Hz bandwidth). B, Feedback analyser (6 % bandwidth). Rotor slots = 40, stator slots = 48.

Secondly, the nature in which waves of radial displacement travelling around the outer perimeter radiate sound energy is a significant factor with larger machines. In this respect the propagational angular velocity of the particular imposed force wave is of importance. For example when this velocity is high enough and the radius large enough for the resultant peripheral velocity at the outer surface to exceed the velocity of sound in air, the machine is an effective sound radiator (Redfearn 1936; Carter 1932). On the other hand, when this velocity is much less than sonic, air vibrations are of a circulatory nature and do not radiate as sound so readily. A further design objective is thus to arrange for the propagational angular velocities to be as small as possible, and this again requires that the pole numbers shall be as large as possible (see table 2).

An interesting example (Brit. Pat. no. 761139) of a quiet design of induction motor is shown in figure 4. A rotor has been selected in which the normal shaft can be replaced by a hollow one of larger diameter. A separate drive shaft passes through this hollow shaft and is coupled to it at both ends by flexible couplings. The whole is then enclosed in an outer casing or hood having sleeve bearings to support the drive shaft. The weight of the motor body and its reaction forces are carried by conventional resilient mountings. In this way the airborne noise of the motor is attenuated by the outer casing and the structure-borne sound or vibration is attenuated by the resilient mountings and couplings. The result is a quieter motor which can be solidly bolted down and directly coupled without fear that the structure-borne noise will be radiated from the rest of the assembly. A prototype 2 h.p. motor-built on this principle gave a sound level of less than 35 dB (A) at one metre on no load.

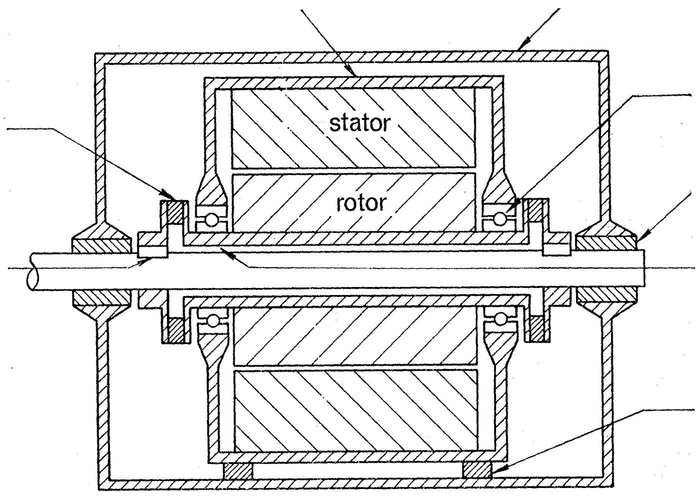

FIGURE 4. Quiet induction motor.

ENCLOSURE OF MACHINES BY SOUND ATTENUATING PANELS

In many respects the design of enclosures for machinery presents problems not generally encountered in the design of architectural partitions. The two most important of these are that, first, components of noise having discrete frequencies usually predominate, and secondly the source to enclosure spacing will generally be small, usually less than a wave-length of sound in air at the lower frequencies.

Because of the preponderance of discrete components, rather than broadband noise, design data obtained by methods suitable for architectural applications (B.S.I. 1956) are not always valid, since warble tones or broadband noise are used in the obtaining of such data. For example, it is possible to have two panels whose 'transmission losses' (Beranek 1949) are similar but which have widely differing performances when subjected to discrete tones.

Furthermore, because free space conditions do not prevail inside an enclosure, transmission loss data will again be invalid, since the total sound power radiated by a vibrating machine depends markedly upon the environment in which it is situated. This follows since the sound energy radiated by any surface having a given oscillatory motion is a function of the acoustical impedance into which it is feeding. An important consequence of this is that under many practical conditions the presence of a poorly designed enclosure placed around a source can result in an actual increase in the radiated sound energy compared with the untreated source, especially at low frequencies (Jackson 1962). For applications such as the enclosing of, say, power transformers where the major part of the radiated sound occurs at low frequencies, such factors are of considerable importance.

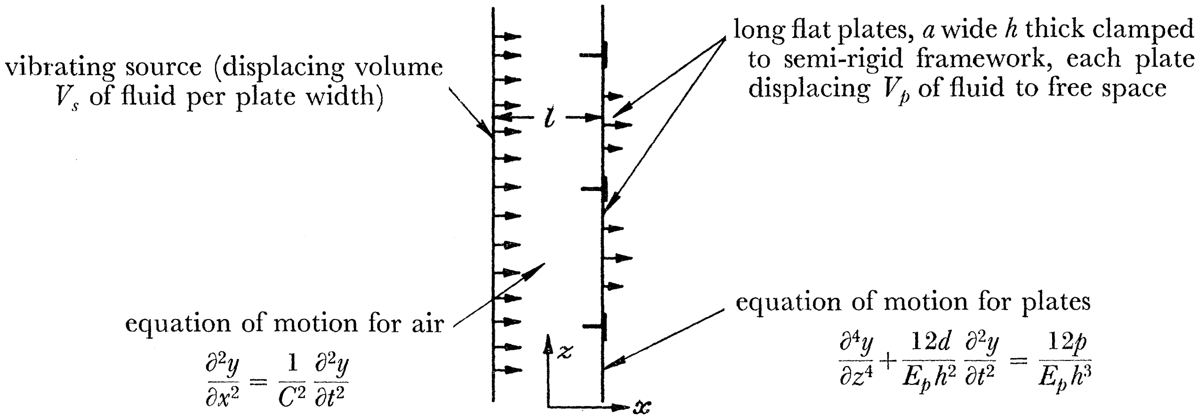

FIGURE 5. Section of a practical hood arrangement consisting of flat plates clamped in a semi-rigid framework. The solution is

$$\frac{V_p}{V_s} = \left[\cos\theta + \frac{1}{aNS_A}\left(\frac{\sin\theta}{\theta}\right) + \mathrm{j}\sin\theta\right]^{-1}$$

where $\theta = \dfrac{\omega l}{C}, \quad S_A = \dfrac{\rho C^2}{al}, \quad \mathrm{j}^2 = -1$

$$N = -\frac{1}{\omega^2 hd}\left\{1 - \frac{2\sin\psi\sinh\psi}{\psi\Delta} - \frac{2B}{aS_t\Delta}(\cosh\psi\sin\psi + \cos\psi\sinh\psi)\right\}$$

B = flexural rigidity of plate.

$$2\psi = na, \quad n^4 = \frac{\omega^2 hd}{B}$$

d = density of plate material.

$$\Delta = \left(\cos\psi - \frac{Bn^2}{S_x}\sin\psi\right)\left(\sinh\psi + \frac{Bn}{S_t}\cosh\psi\right) + \left(\cosh\psi - \frac{Bn^3}{S_x}\sinh\psi\right)\left(\sin\psi + \frac{Bn}{S_t}\cos\psi\right)$$

S_x, S_t = direct and torsional stiffnesses of supports.

The performance of enclosures in general can be demonstrated by considering the behaviour of thin flat sheets supported in a semi-rigid framework spaced a small distance away from a vibrating source. A section of such an arrangement is shown schematically in figure 5. The behaviour of this construction can be readily analysed mathematically (to be published) especially if the plates are considered to be infinitely long. What is required from the analysis is the ratio between the (oscillatory) volume displacement of a plate and the volume displacement of a corresponding area of source, since this forms a convenient criterion of the ability of a hood system to act as an isolator of airborne sound (Jackson 1962). The approximate expression for this ratio is included with figure 5.

In figures 6 and 7 curves showing the calculated sound attenuations provided by simple undamped flat plate systems of the above kind are given. The three curves of figure 7, which refers to 0·25 in. thick aluminium plates having built-in rigid supports, show the effects of varying the distance between source and hood. The resonances at approximately 70, 340, 800 and 1530 Hz are plate flexure resonances and are substantially unaffected by changes of source to hood plate spacing. The only means of reducing the heights of these peaks is to introduce mechanical damping into the plates themselves by the use of damping treatments (to be published; Jackson 1966).

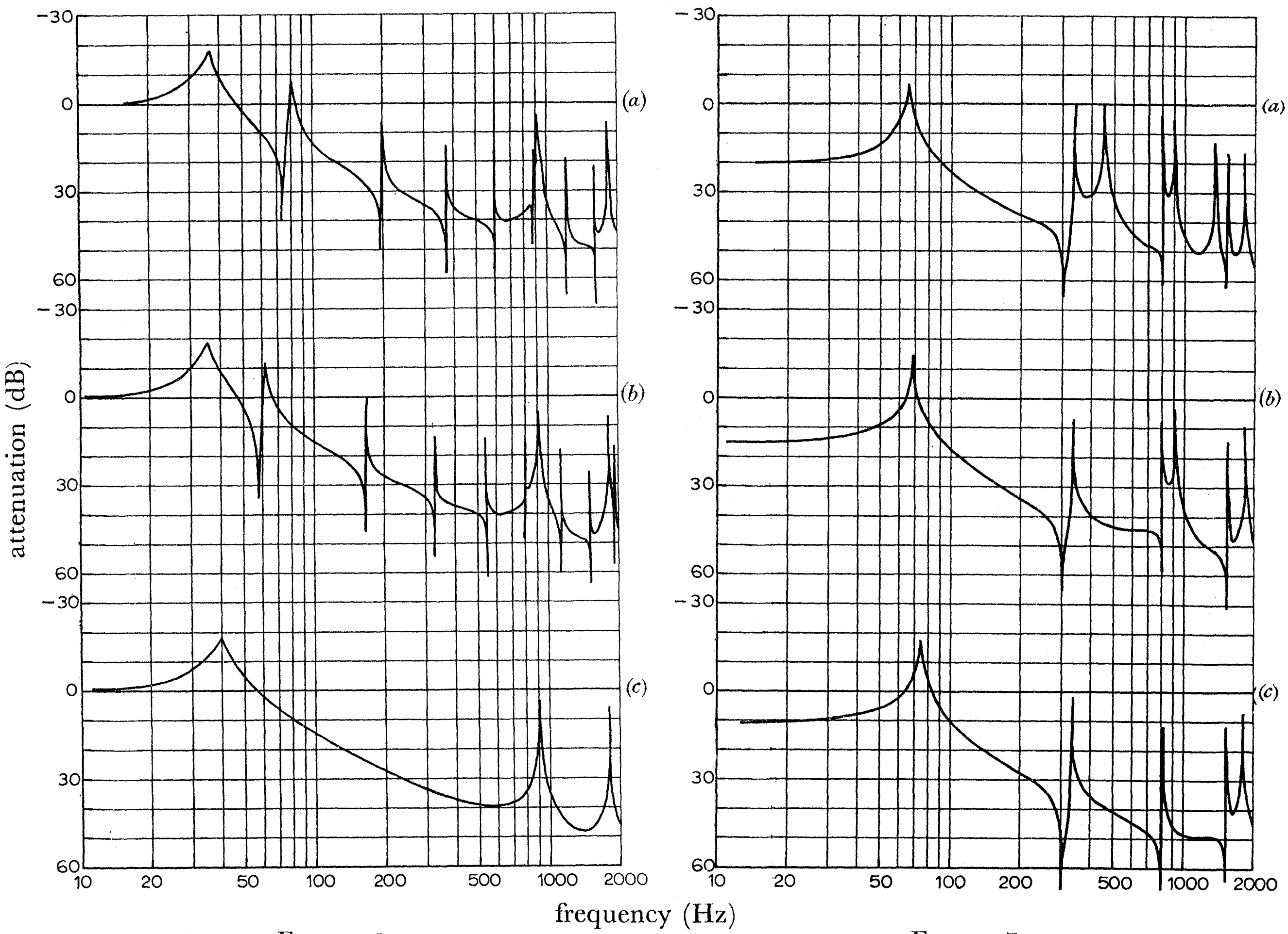

FIGURE 6

FIGURE 7

FIGURE 6. Flat steel plates, unlimited length, 30 in. wide, 0·06 in. thick. No mechanical or acoustical damping. Source plate to hood plate spacing = 7·5 in. (*a*) Built-in supports ($S_x \to \infty$, $S_t \to \infty$); (*b*) simple supports ($S_x \to \infty$, $S_t = 0$); (*c*) unconstrained at supports ($S_x = 0$, $S_t = 0$).

FIGURE 7. Flat aluminium plates, unlimited length, 30 in. wide, 0·25 in. thick. No mechanical or acoustical damping. Built in supports. (*a*) Source plate to hood plate spacing = 15 in.; (*b*) source plate to hood plate spacing = 7·5 in.; (*c*) source plate to hood plate spacing = 3·75 in.

On the other hand, the remaining peaks, e.g. at 450, 900, 1350 and 1800 Hz for curve (*a*), are standing wave resonances which occur when the source to hood spacing equals a multiple of a half wavelength of sound in air. These resonances can only be successfully reduced in height by introducing a sound absorbent layer into the intervening airspace.

The essential differences between the effects of acoustical and mechanical damping are illustrated by the curves of figure 8. If a sound-absorbent material such as glass wool is placed on the inner surface of a hood-plate system the air resonances will be substantially reduced in height, but little benefits will be derived at other frequencies. For example, the absorption coefficient at normal incidence (Kinsler & Frey 1962) of a typical absorbent layer is illustrated by figure 8 (*a*). The effect of placing this on the inner surface, when the

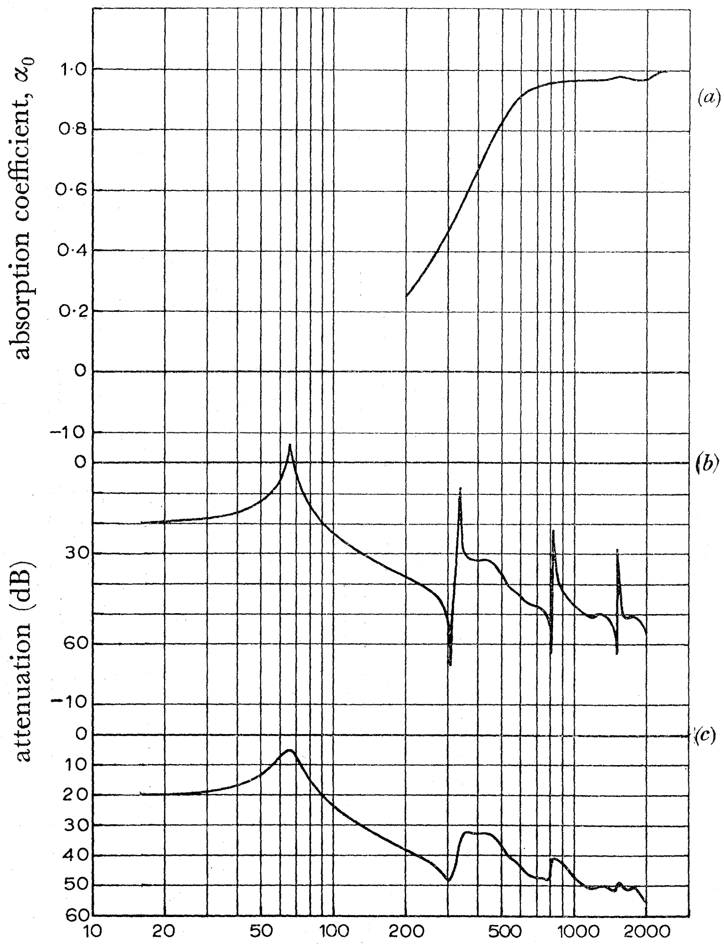

FIGURE 8

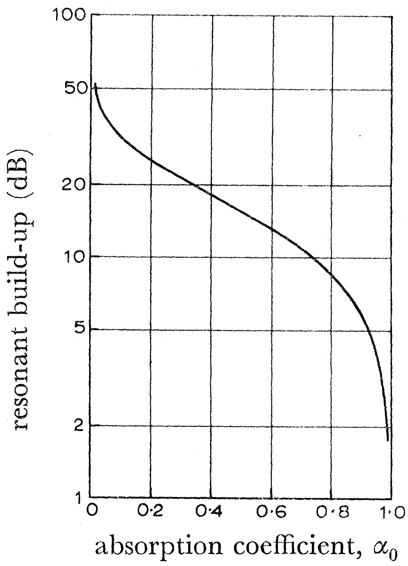

FIGURE 9

FIGURE 8. (*a*) Normal incidence absorption coefficient for 2 in. thick layer of crown 200 fibreglass faced with 0·5 in. thick layer of superfine fibreglass. (*b*) 0·25 in. thick aluminium plate, 30 in. wide, source to plate spacing = 15 in., absorbent layer on inner surface of plate. No mechanical damping; (*c*) as (*b*), but with mechanical damping applied to plate (effective Q of plate material = 10).

FIGURE 9. Effects of acoustical absorbent material on hood surface adjacent to source on standing wave resonances.

spacing is 15 in., is illustrated in figure 8 (*b*), which is directly comparable with curve (*a*) of figure 7, the untreated case. It is seen that, first, the plate resonances remain substantially unaltered, and secondly, only at or near to standing wave resonance frequencies are any benefits derived.

The absorbent treatment is thus seen to be little more than an effective means of reducing the resonant build-up due to standing waves, since propagation losses (Scott

1946) through such materials are not usually large enough to be of practical value. A relationship between acoustical build-up and absorption coefficient has been obtained (to be published) and the graph given in figure 9 shows how this relation behaves at a standing wave resonance. Acoustical build-up is defined as the ratio of the sound pressure at the inner wall of the enclosure to the sound pressure which would occur at this position if the enclosure were not present. This curve enables an assessment to be made of how much resonant build-up occurs for a given acoustical treatment, since it is usual for the elementary properties of absorbent linings to be specified in terms of absorption coefficient.

The effects of mechanical damping applied to the plates can be investigated by assuming their flexural rigidity, B, to be viscoelastic. A realistic value for the effective Q (Kinsler & Frey 1962) of a treated plate would be of the order of 10. Curve (*c*) of figure 8 has thus been calculated with an assumed value of $Q = 10$ for the plate material, and the graph shows the effects of this damping in addition to the acoustical absorption. The combined effects of the two forms of damping are thus seen to be necessary to achieve good over-all attenuation, although it is clear that only close to the resonances are these treatments really effective. Experiments on practical panels have confirmed this (Jackson 1962, 1966).

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