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ON ACOUSTIC RADIATION RESISTANCE OF PLATES

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It is reported that when both sides of a plate radiate into a reverberant room, the radiation resistance obtained experimentally is approximately half of the values obtained using theoretical expression. Hence, it was suggested to use plate area as the radiating area instead of using double the plate area as radiating area which is not logical. In this paper the reasons for the above anomaly are investigated. It is found that the error is not in the radiating area but in the expression for radiation resistance itself. New expressions for radiation resistance are suggested. Experimental results compare well with the theoretical results.

1. INTRODUCTION

Radiation resistance is a measure of the sound power radiated by a structure for a given vibration level. The response of a structure to acoustic excitation depends on the radiation resistance of the structure. Hence, it is an important parameter in studying the structure–acoustic interaction.

Analytical technique to obtain modal radiation characteristics of a structure, i.e., the sound radiation at a particular frequency with a particular mode shape, is well established [1–5]. This can be done either by the modal approach of the wave approach. Cremer *et al*. [3] and Fahy [4] have described both the methods in detail.

For the radiation resistance of a structure radiating into a reverberant room, an expression was derived by Lyon and Maidanik [6]. The above expression is based on the power flow between linearly coupled multi-modal systems for a white noise excitation. For Statistical Energy Analysis (SEA) applications expression derived in this fashion is quite suitable. Maidanik [7] applied the above theory for simply supported plates and obtained their radiation characteristics. Expressions for frequency averaged radiation resistance are also derived [7–10]. The above equations for radiation resistance are still in use today [11–13].

It is important to note that the above expressions give the radiation resistance of the panel kept in infinite, plane and rigid baffle and radiating into a reverberant room by one side of the panel. When both sides of the panel radiate sound, it is logical that the above value of radiation resistance is multiplied by a factor of 2·0. In other words one can say that the radiating area is twice the area of the plate.

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The above expressions are for the plate radiating into a reverberant room. If the room is very small, at low frequencies the above expressions are not valid. In such a situation one has to follow the method suggested by Fahy [14].

Consider a panel kept in a reverberant acoustic field. To obtain its response using Statistical Energy Analysis (SEA), the coupling loss factor is obtained from its radiation resistance. In such a situation it is logical to assume that both sides of the panel radiate and hence a factor of 2·0 is used to obtain radiation resistance. This argument/theory is followed by many authors [11, 15, 16].

Clarkson and Brown [11] conducted experiments to obtain acoustic radiation damping of panels. A reasonably good match between the measured radiation loss factor and theoretical estimates is reported. However the theoretical estimates are made without using the factor 2·0 though both sides of the panel radiate sound. This means that the measured radiation loss factor and theoretical estimates match only if the factor of 2·0 is not used which does not seem to be logical.

In this paper the anomaly mentioned above regarding the factor 2·0 for radiation resistance when both the sides of the panel radiate sound is investigated. New expressions are suggested for radiation resistance. They are verified by experiments.

2. EXPRESSION FOR RADIATION RESISTANCE

By definition the sound power radiated, W, by a panel having a spatial mean square value of velocity $\langle v^2 \rangle_x$, is related to the radiation resistance R_{rad} by the equation

$$W = R_{rad} \langle v^2 \rangle_x . \tag{1}$$

R_{rad} is dependent on the frequency of excitation and the properties of the structure and the medium into which the sound is radiated.

From power flow concepts Lyon and Maidanik [6] derived the following expression for the radiation resistance of a plate.

$$R_{rad} = (16/\pi)\rho c k^2 \int_{-\infty}^{\infty} \int_{-\infty}^{\infty} \Psi(x_1, x_2)\Phi(x_1, x_2)\, \mathrm{d}x_1\, \mathrm{d}x_2 . \tag{2}$$

In the above integral ρc is the characteristic impedance of the medium in which sound is radiated and k is the wavenumber. In equation (2), $\Psi(x_1, x_2) = \langle f_r(x_1) f_r(x_2) \rangle_r$ and $\Phi(x_1, x_2) = \langle f_m(x_1) f_m(x_2) \rangle_m$ where $f_r(x)$ and $f_m(x)$ are the mode shapes of the acoustic field and the structure, respectively.

Based on the above integral the frequency average radiation resistance of a plate into a reverberant room is derived as follows [7–10].

For $f < f_c$ and $ka, kb > 2\pi$,

$$R_{rad} = A\rho c\{(\lambda_c \lambda_a / A) 2 (f/f_c) g_1 + (p\lambda_c / A) g_2\},$$

where

$$g_1 = (4/\pi^4)\{(1 - 2\Psi^2)/[\Psi(1 - \Psi^2)^{(1/2)}]\} \quad \text{for } f/f_c < 0{\cdot}5$$

$$= 0 \quad \text{for } f/f_c > 0{\cdot}5$$

$$g_2 = (1/4\pi^2)\{(1 - \Psi^2) \ln[(1 + \Psi)/(1 - \Psi)] + 2\Psi\}\{1/(1 - \Psi^2)^{(3/2)}\}$$

$$\Psi = (f/f_c)^{(1/2)}.$$

For $f < f_c$ and $ka, kb < 2\pi$,

$$R_{rad} = A\rho c(4/\pi^4)\,(p\lambda_c\,/A)\,(f/f_c)^{(1/2)}.$$

For $f = f_c$,

$$R_{rad} = A\rho c\{(a/\lambda_c)^{(1/2)} + (b/\lambda_c)^{(1/2)}\}.$$

For $f > f_c$,

$$R_{rad} = A\rho c\{1 - (f_c\,/f\,)^{-(1/2)}. \tag{3}$$

The plate has dimensions a and b, area A, perimeter p and critical frequency f_c. λ_c is the wavelength corresponding to critical frequency and λ_a is the wavelength in air at the given frequency. The above expressions are for a simply supported plate kept in a rigid baffle and radiating by one side of the plate into a reverberant room.

Now consider the derivation of equation (2) based on which equation (3) is derived. This is based on power flow between two coupled multi-modal systems. The two multi-modal systems are the structure and the acoustic field.

The power flowing from the mth mode having a frequency of ω_m of one system to the rth mode having a frequency of ω_r of the other system, $\pi_{m,r}$ is given by [6, 17]

$$\pi_{m,r} = B_{m,r}\,[E_m - E_r], \tag{4}$$

where E_m and E_r are the respective modal energies.

$$B_{m,r} = b^2_{m,r}\,\{\beta_m\,\omega^2_r + \beta_r\,\omega^2_m\}/[\{\omega^2_m - \omega^2_r\}^2 + (\beta_m + \beta_r)\{\beta_m\,\omega^2_r + \beta_r\,\omega^2_m\}]. \tag{5}$$

In equation (5), β_m and β_r are the half-power bandwidths of the modes and $b^2_{m,r}$ depends on the mode shapes of the two systems.

The power flowing from the first system to the other, $\pi_{1,2}$, is the sum of the power flowing from all the modes of the first system to all the modes of the second system. Hence,

$$\pi_{1,2} = N_1\,N_2\langle B_{m,r}\rangle[E_{1m} - E_{2m}]. \tag{6}$$

N_1 and N_2 are the number of modes of the systems in the band of frequency $\Delta\omega$. E_{1m} and E_{2m} are the modal energies of each system and constant modal energy is assumed for a particular system. $\langle B_{m,r}\rangle$ is the average value of $B_{m,r}$ over different modes.

To obtain $\langle B_{m,r}\rangle$, let us examine the expression for $B_{m,r}$, i.e., equation (5). The expression for $B_{m,r}$ has two terms. The first term, $b^2_{m,r}$, depends on the mode shapes. The rest of the expression for $B_{m,r}$ is basically a function of the difference between the natural frequencies. Hence, if one uses $b^2_{m,r}$ averaged over different modes, denoted by $\langle b^2_{m,r}\rangle$, the averaging of $B_{m,r}$ reduces to the averaging of equation (5) after dropping the term $b^2_{m,r}$.

Lyon [17] carries out the above averaging (averaging of equation (5) after dropping the term $b^2_{m,r}$) in the following way. It is assumed that ω_m is a constant and ω_r is a random variable. Variable ω_r can have any value in the band $\Delta\omega$ with a uniform probability of occurrence. The integration is done in the entire frequency band $\Delta\omega$. Now assume that ω_m can have any value in the frequency band $\Delta\omega$. The integration yields

$$\langle B_{m,r}\rangle = (\pi/2)\langle b^2_{m,r}\rangle/\Delta\omega. \tag{7}$$

The procedure used for the above averaging is different in reference [6]. Here, $B_{m,r}$ is approximated by the expression

$$B_{m,r} = b^2_{m,r}\,\{\pi/(\beta_m + \beta_r)\}, \tag{8}$$

if the difference between the frequencies is less than half-power bandwidth and is zero otherwise. The averaging thus leads to

$$\langle B_{m,r} \rangle = \pi \langle B_{m,r}^2 \rangle / \Delta\omega. \tag{9}$$

Hence, in reference [17] the integration is performed on the exact expression which gives a very high value when $\omega_m - \omega_r$ is small and a very low value when $\omega_m - \omega_r$ is large. In reference [6] the integration is carried out on the expression which has a constant value in the half-power bandwidth. One can see the difference in the results obtained by the two methods of averaging. The value of $\langle B_{m,r} \rangle$ as per reference [6] is twice that of reference [17].

Maidanik obtained the expression for radiation resistance using the integral given by equation (2). The above integral is derived based on $\langle B_{m,r} \rangle$ given by equation (9). Since equation (7) is based on the exact integral, it is suggested to use equation (7) for $\langle B_{m,r} \rangle$. Hence, the integral for radiation resistance becomes

$$R_{rad} = (8/\pi)\rho c k^2 \int_{-\infty}^{\infty} \int_{-\infty}^{\infty} \Psi(x_1, x_2) \Phi(x_1, x_2) \, \mathrm{d}x_1 \, \mathrm{d}x_2. \tag{10}$$

From equations (10) and (2) it can be seen that the suggested radiation resistance values are half the values given by the existing expressions.

Using integral (10) the expressions for radiation resistance for a plate become as follows.

For $f < f_c$ and $ka, kb > 2\pi$,

$$R_{rad} = A\rho c\{(\lambda_c \lambda_a / A) 2 (f/f_c) g_1 + (p\lambda_c / A) g_2\}/2,$$

where

$$\begin{aligned} g_1 &= (4/\pi^4)\{(1 - 2\Psi^2)/[\Psi(1 - \Psi^2)^{(1/2)}]\} \quad \text{for } f/f_c < 0{\cdot}5 \\ &= 0 \qquad\qquad\qquad\qquad\qquad\qquad\quad \text{for } f/f_c > 0{\cdot}5 \\ g_2 &= (1/4\pi^2)\{(1 - \Psi^2) \ln[(1 + \Psi)/(1 - \Psi)] + 2\Psi\} \{1/(1 - \Psi^2)^{(3/2)}\} \\ \Psi &= (f/f_c)^{(1/2)}. \end{aligned}$$

For $f < f_c$ and $ka, kb < 2\pi$,

$$R_{rad} = A\rho c (4/\pi^4)(p\lambda_c / A)(f/f_c)^{(1/2)}/2.$$

For $f = f_c$,

$$R_{rad} = A\rho c\{(a/\lambda_c)^{(1/2)} + (b/\lambda_c)^{(1/2)}\}/2.$$

For $f > f_c$,

$$R_{rad} = A\rho c\{1 - (f_c/f)\}^{-(1/2)}. \tag{11}$$

The difference between equation (11) and equation (3) is that a factor of 0·5 is added for radiation resistance when $f \leqslant f_c$. Radiation resistance for $f > f_c$, is the same as that of an infinite plate and is not derived based on the integrals discussed above and hence the factor 0·5 is not used. But if one uses equation (10) and derives the above expression, it will be half of the value given by equation (2).

Therefore, it is suggested that equation (11) be used for calculating radiation resistance of panels. Also, when both sides of the panel radiate, the above expressions should be multiplied by a factor of 2·0.

As discussed earlier, Clarkson and Brown [11] used equation (3) to obtain the radiation resistance of a panel when both sides radiate into a reverberant room. The experimental results were lower than the theoretical estimates. They suggested using the plate area as radiating area instead of using twice the plate area as the radiating area by which the results match well. It is now clear that the differences seen in the experimental and the theoretical estimates of radiation resistance are not due to the error in the factor used for radiating area but the error in the expression for radiation resistance itself.

The above expressions for radiation resistance are for simply supported boundary conditions. Boundary conditions do not affect radiation resistance at higher frequencies. But at low frequencies boundary conditions do influence radiation resistance. In such cases equation (11) can be multiplied by a factor depending on the boundary condition. These factors are derived by Nikiforov [18] and a more general formulation is given in reference [19]. This factor is equal to 1·0 for simply supported boundary conditions and 2·0 for clamped boundary conditions except at very low frequencies. It is nearly zero for free edge conditions. The above effects are dominant at low frequencies. The frequency range in which these effects are significant are debatable. The logic could be that if the plate contains a few bending waves, i.e., $\lambda_b < a$, the effect of boundary conditions will not be significant.

Equation (11) is for the radiation resistance of a plate with a plane baffle. For the radiation resistance of plates which have other types of nearby structures the results obtained by equation (11) shall be multiplied by a factor. This factor can be obtained in a way suggested by Lyon [20] and Price and Crocker [9] which is as follows. The sound power radiated by a source kept on a rigid floor is twice the sound power radiated by the same source when kept at the centre of the room. It is to be noted that the radiating area is the same in both cases, although the sound power radiated is different depending on the presence of rigid walls nearby. Discussion on the above factor, called the directivity factor, for different configurations for the nearby rigid walls is available in the literature [16, 21]. A similar logic can be used to obtain the radiation resistance of unbaffled panels. For example, the above factor for a panel, which is a side wall of an enclosure, radiating into the enclosure, is 2·0. Similarly if the panel is hanging in the chamber without any baffle, the radiation resistance will be half the radiation resistance of the baffled panel.

The above effect is expected to be dominant for corner modes. In a frequency band both corner modes and edge modes will be present at frequencies below critical frequency. At low frequencies most of the modes will be corner modes and at higher frequencies most of the modes will be edge modes. It can be shown that corner modes do not occur at frequencies above $0{\cdot}5f_c$. Hence, it is recommended to use the baffling effect up to $0{\cdot}5f_c$. Experimental results, which are given later, confirm the above behaviour.

For an unbaffled plate other than the factor suggested above the "short-circuiting" acoustic field around the free edges affects the radiation resistance. The free edge means the edges without any baffle and not the boundary condition which could be simply supported, clamped, free or any other type. Short-circuiting is due to the flow of air from one side of the plate to the other without getting compressed. Consequently it reduces the radiation resistance. The effect is more obvious at low frequencies when it gets more time to escape from compression. The effect becomes very significant when the acoustic wavelength is larger than the plate dimensions. Oppenheimer and Dubowsky [13] have derived expressions for taking into account the short-circuiting effects. They have suggested that for $f < f_c$

$$R_{rad} = F_{plate}\,\{F_{corner}\,R_{rad,c} + F_{edge}\,R_{rad,e}\}. \tag{12}$$

In equation (12) F_{corner} and F_{edge} are the correction factors for corner as well as edge modes, respectively. F_{plate} is the correction factor for the flow around the plate and F_{edge} and F_{corner}

are for the flow near the edges. $R_{rad,c}$ and $R_{rad,e}$ are the radiation resistance of corner and edge modes, respectively. These correction factors are given below [13].

$$F_{corner} = 13(f/f_c)/\{1 + 13(f/f_c)\}, \tag{13}$$

$$F_{edge} = 49(f/f_c)/\{1 + 49(f/f_c)\}, \tag{14}$$

$$F_{plate} = 53f^4A^2/c^4/\{1 + 53f^4A^2/c^4\}. \tag{15}$$

The corner and edge correction factors are significant only at low frequencies and the plate correction factor is significant only when the acoustic wavelength is larger than the plate dimensions.

In summary, it is suggested that equation (11) be used with a factor for boundary conditions, effect of neighbouring structure and effect of inertial flows for obtaining radiation resistance of plates.

3. EXPERIMENTAL RESULTS

To validate the above results, radiation resistance is obtained experimentally. A thin plate kept in a reverberation chamber is excited using a shaker and the sound pressure level (SPL) in the chamber is measured. Radiation resistance is obtained from the measured SPL and the plate vibrations.

3.1. THEORY

Consider a panel kept in a reverberation chamber excited mechanically. For SEA modelling the acoustic field is taken as subsystem 1 and the vibration field of the plate is taken as subsystem 2. The power balance of subsystem 1 becomes

$$\pi_1 = \omega(\eta_1 + \eta_{12})E_1 - \omega\eta_{21} E_2, \tag{16}$$

where π_1 is the power input and η_1 is the dissipation loss factor of subsystem 1. η_{12} and η_{21} are the coupling loss factors and E_1 and E_2 are the total mean energies of the subsystems.

Since the power input to subsystem 1 is zero, from equation (16) the energy of subsystem 1 is obtained as

$$E_1 = \{\eta_{21} /(\eta_1 + \eta_{12})\}E_2. \tag{17}$$

The energy of the plate is given by

$$E_2 = mA\langle v^2\rangle_x, \tag{18}$$

where m is the mass of the plate per unit area. The energy of the acoustic field having a volume of V_1 and mean square value of acoustic pressure p^2_{rms} is

$$E_1 = (p^2_{rms} /\rho c^2)V_1. \tag{19}$$

The dissipation loss factor of the room can be shown to be

$$\eta_1 = sc\bar{\alpha}/(8\pi fV_1), \tag{20}$$

where

$$\bar{\alpha} = \{\alpha + Md - (M^2d^2/2)\}/\{1 - [\alpha + Md - (M^2d^2/2)]\}. \tag{21}$$

In the above equation α is the sound power absorption coefficient of the walls. Sound absorption due to air is represented by Md in which d is the mean free path of the room and M depends on humidity and temperature of the air. The term $M^2d^2/2$ is significant at very high frequencies, say, 8000 Hz.

By definition

$$\eta_{21} = R_{rad}/(\omega m A). \quad (22)$$

From the reciprocal relationship

$$\eta_{12} = \eta_{21}(n_2/n_1). \quad (23)$$

The modal density of the room is given by

$$n_1 = 4\pi f^2 V_1/c^3. \quad (24)$$

Substituting equations (18)–(24) into equation (17) and with suitable mathematical operations one can show that

$$R_{rad} = p_{rms}^2 s\bar{\alpha}/\{4\rho c\langle v^2\rangle_x - (n_2 c^2/\pi f^2 m A)p_{rms}^2\}. \quad (25)$$

Radiation resistance is obtained experimentally using equation (25).

3.2. TEST SET-UP

The test set-up is shown in Figure 1. The plate is hung in a reverberation chamber. The plate is made of aluminium having dimensions 2·19 × 1·22 m. The thickness of the plate is 4·95 mm. Young's modulus of aluminium is taken as $7{\cdot}2 \times 10^{10}$ N/m^2 and the density as 2800 kg/m^3. Modal density of the plate is calculated as 0·176 lHz. Critical frequency of the plate is estimated to be 2512 Hz.

The reverberation chamber has dimensions of 10·33 × 8·2 × 13 m. The chamber has a surface area of 651·2 m^2. For the convenience of testing, the door of the chamber was kept open for a length of 0·68 m. The medium of the chamber is air. The temperature of the air was 25°C and the relative humidity was 51%. The above parameters are important in obtaining the sound power absorption due to air. For the above conditions the speed of sound in air is taken as 346 m/s. The density for the air is assumed to be 1·21 kg/m^3.

Excitation at a single point does not produce statistically independent modes [22]. Bies and Hamid [23] have suggested, from experimental results, to use a minimum of the three randomly selected driving points to overcome the above problem. In the present experiment the plate is excited at five randomly selected locations which are shown in Figure 2. An electromagnetic shaker is used to excite the structure. The excitation scheme is shown in Figure 3. The output of the random noise generator is fed through an amplifier before being applied to the shaker. An aluminium block is bonded on the plate at the point of excitation. The shaker is connected to the block through a stringer and impedance head.

Vibration responses are measured at six randomly selected locations shown in Figure 2. The mass of each accelerometer is approximately 0·5 g. The resonant frequency of the accelerometer is 32 kHz with a useful frequency range of (±5%) of 5–8000 Hz. The estimated average impedance of the plate is 844 N s/m. Since the impedance due to accelerometer mass is only 31·4 N s/m at 10 000 Hz, the mass loading of the accelerometer on the measured response is very much negligible. Acceleration measurement set-up is shown in Figure 3. The output signal of the accelerometer is conditioned using a charge amplifier and the charge amplifier output is recorded on an FM tape recorder.

Sound pressure levels in the room are measured at three locations using condenser type microphones. The sensitivity of these microphones is approximately 12·0 mV/Pa. The resonant frequency of the microphone is about 15 000 Hz. This causes an errror of 0·2 dB at 5000 Hz and 1 dB at 10 000 Hz. These correction factors are applied in obtaining the correct sound pressure level. The microphone is fitted with suitable pre-amplifier and 200 V polarization voltage is applied. The output of the microphone is recorded on an FM tape

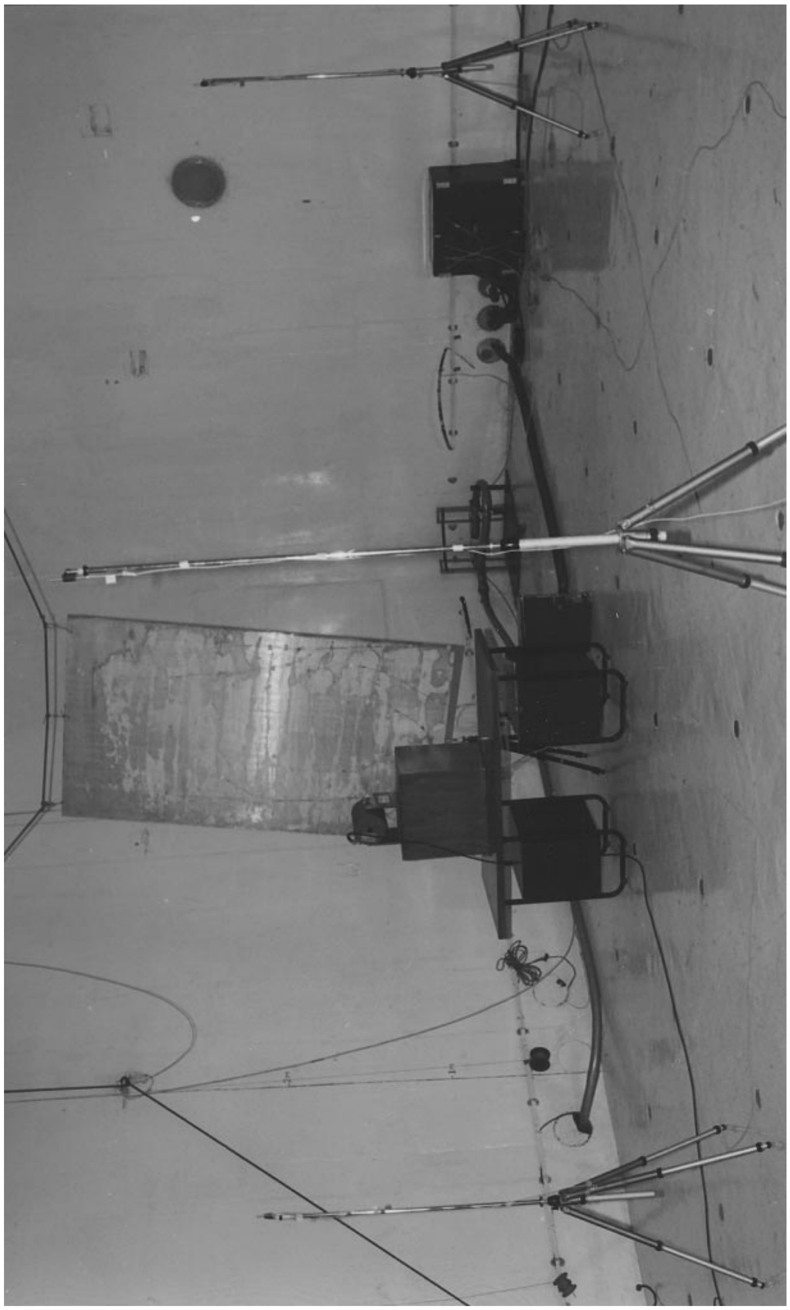

Figure 1. A view of the test set-up.

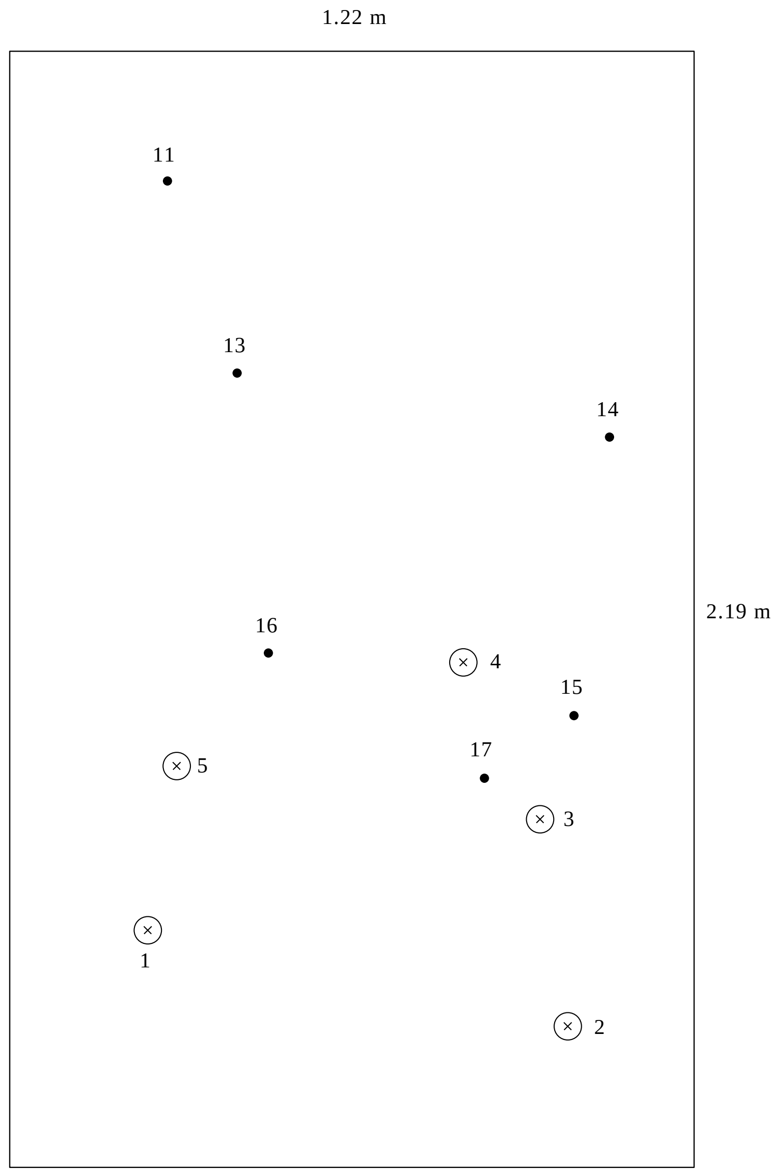

Figure 2. Driving point (⊗) and response (●) measurement locations.

recorder. The block diagram showing the measurement of sound pressure level is given in Figure 3.

The above data is analysed off-line using FFT based signal analysers. It can also be seen that required care is taken in obtaining accurate data up to 10 000 Hz.

3.3. ROOM ABSORPTION

As can be seen from equation (25), the exprerimental determination of radiation resistance is largely dependent on the sound power absorption by the chamber. This

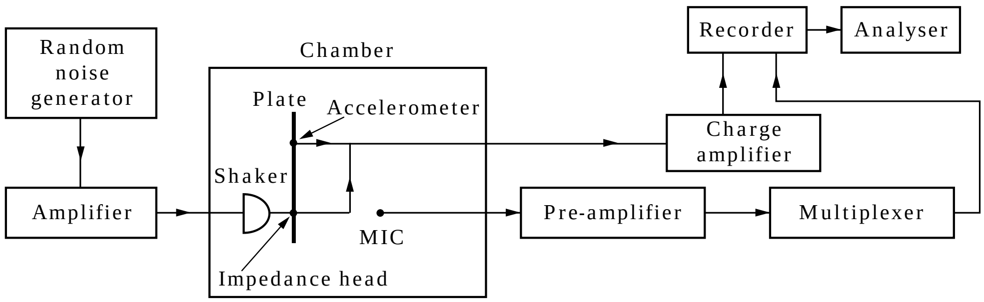

Figure 3. Block diagram of the measurement set-up.

absorption has two components, one the absorption by the wall and the other by air (refer to equation (21)).

Absorption due to air depends on its temperature and relative humidity and these are represented by M. These values for some standard conditions are available in the literature [16, 21]. In this case, values of M for a temperature of 25°C and 50% relative humidity are used. Absorption due to air is significant only at values above 2000 Hz. The mean free path of acoustic wave in this chamber is 6·76 m.

Absorption due to walls is obtained experimentally. For this, the reverberation time of the room is found experimentally using standard procedure. For brevity the details of the experiments are not given here. The results are shown in Figure 4. The reverberation time of the room, T, is given by

$$T = 55{\cdot}26(\mathrm{V/cs})/\{\ln(1-\alpha)^{-1} + Md\}. \tag{26}$$

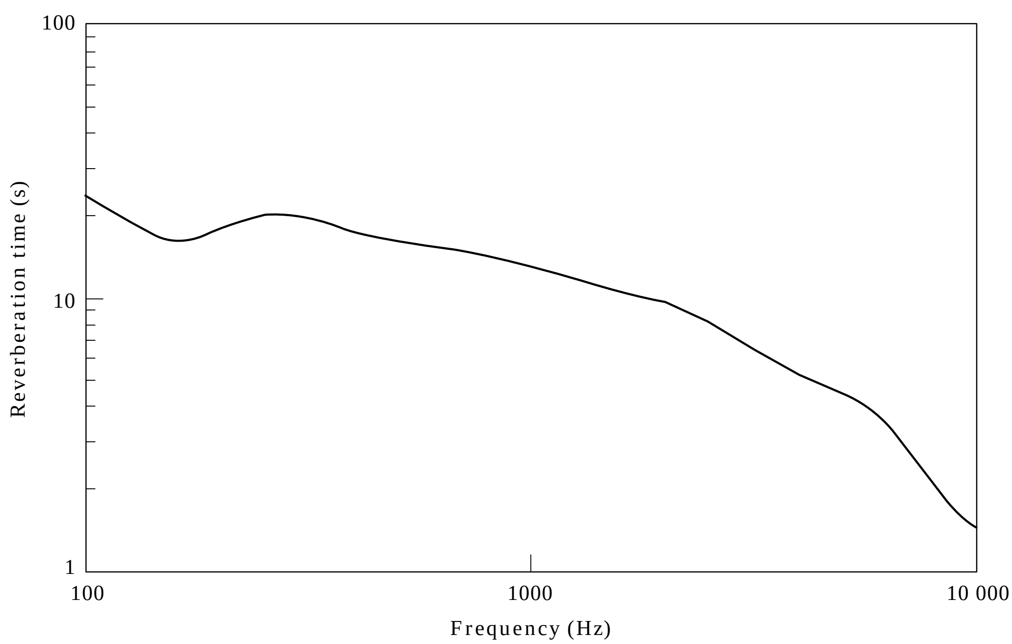

Figure 4. Reverberation time of the chamber.

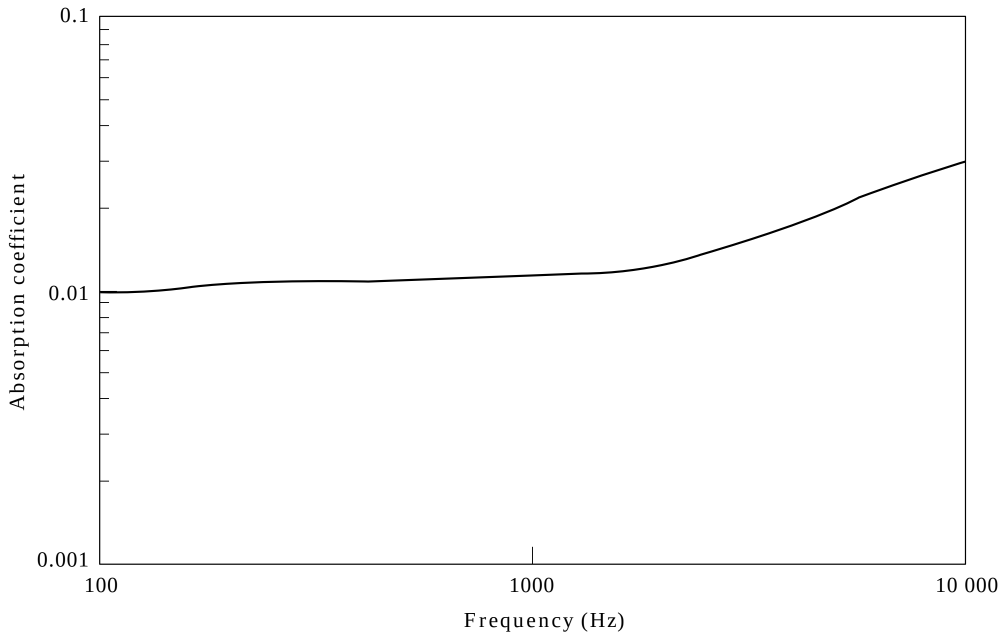

Figure 5. Sound power absorption coefficient of the walls.

From the known values of reverberation time and the temperature and humidity of the air, the sound power absorption coefficient of the walls is obtained. The results are shown in Figure 5.

From the above results, room absorption, i.e., $s\bar{\alpha}$, is calculated. It is to be remembered that during the present test the chamber is open for a length of 0·68 m and a height of

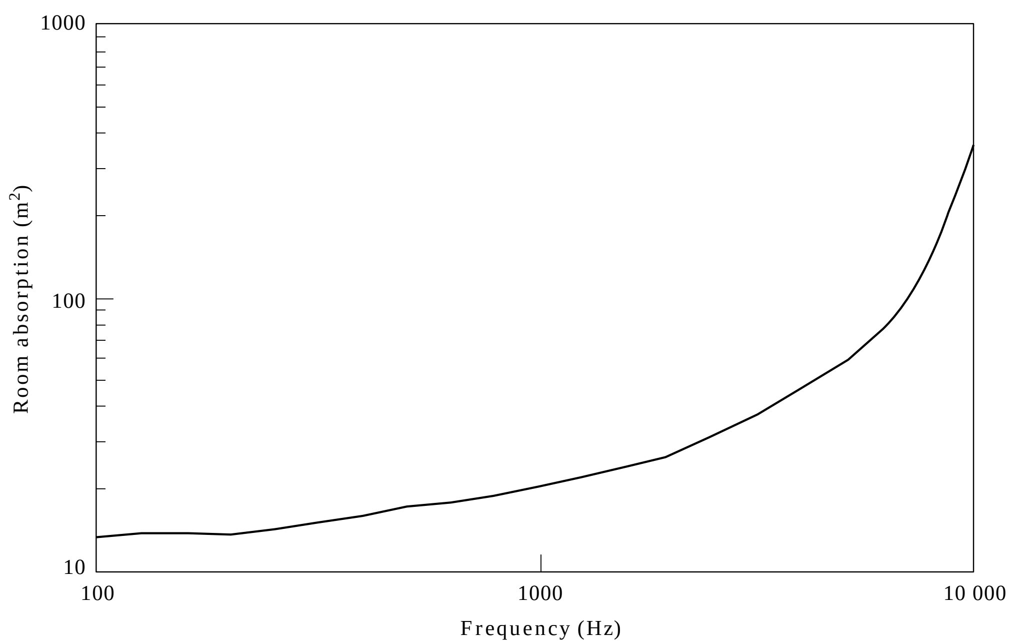

Figure 6. Room absorption of the chamber.

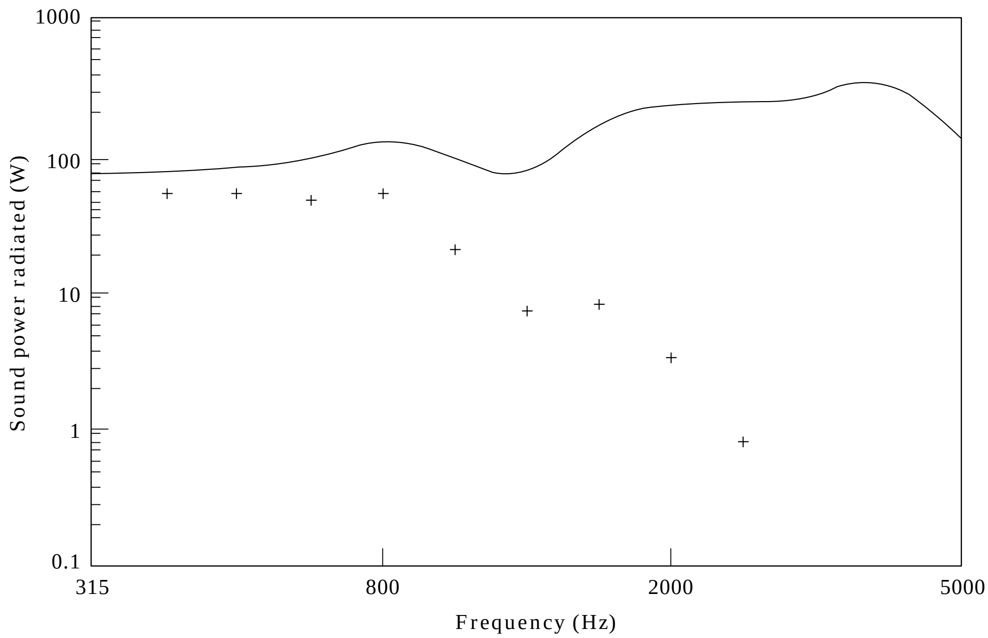

Figure 7. Sound power radiated by the plate: ——, total power; +, near field radiation.

10 m to facilitate some cable connections. This area (6·8 m^2) will have an absorption coefficient of unity at higher frequencies. This is taken into consideration in evaluating room absorption. This has a significant effect at low frequencies where absorption by wall and air is very low. Room absorption thus obtained is shown in Figure 6. The opening has an absorption coefficient of unity only at higher frequencies, in this case above 250 Hz.

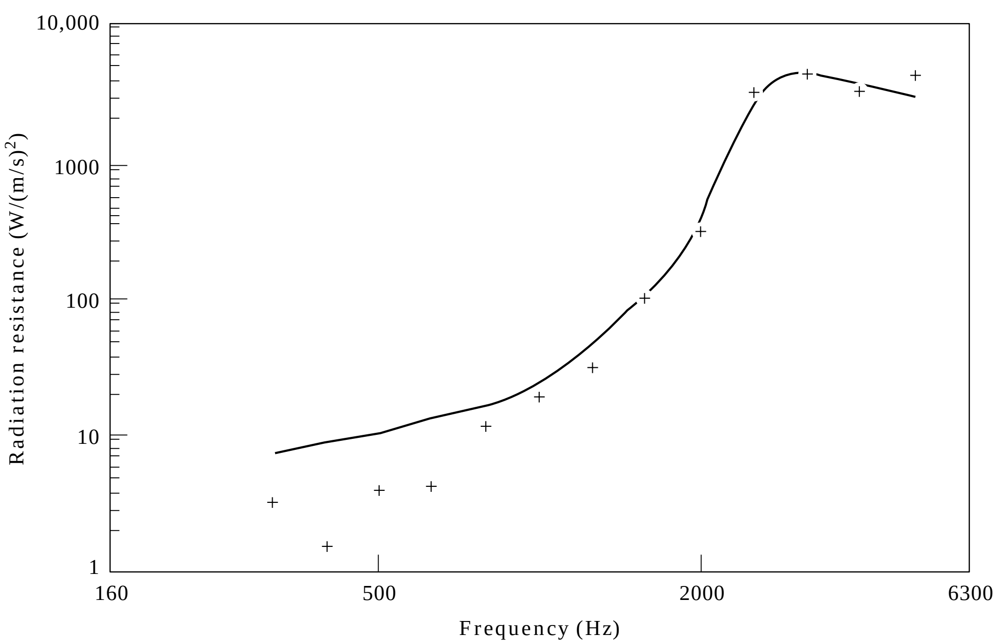

Figure 8. Radiation resistance of the plate: ——, theory; +, experiment.

Table 1

Radiation resistance of a typical plate

1/3 Octave band centre frequency (Hz)	Radiation resistance (W/(m/s)^2)			
	Experiment	Present theory	Equation (3)	Equation (3) with plate area as radiating area
315	3·13	7·49	35·62	7·49
400	1·54	8·94	40·96	8·94
500	3·95	10·81	48·03	10·81
630	4·24	13·26	57·61	13·26
800	11·57	16·77	71·53	16·77
1000	18·39	22·21	93·42	22·21
1250	32·14	50·67	132·4	50·67
1600	108·4	107·5	221·9	107·5
2000	314·9	298·6	612·1	298·6
2500	3339	3978	4268	2134
3150	4531	5000	5000	2500
4000	3496	3675	3675	1838
5000	4290	3158	3158	1579

At low frequencies it is lower than unity. Hence the results given here are higher than the actual absorption at low frequencies.

3.4. Near field radiation

Sound radiated by the near field for a mean square value of force $F^2(t)$ is estimated using the equation [3, 4, 16]

$$W = \rho F^2(t)/(2\pi c m^2) \qquad \text{for } f < f_c. \tag{27}$$

Total power radiated by the panel can be obtained from equations (25) and (1). Figure 7 gives a comparison of the total power radiated and the power radiated by the near field. Power radiated by the near field is significant at low frequencies.

3.5. Radiation resistance

Sound power radiated by the far field is obtained from the total power radiated and the near field radiation. The total power radiated is from the experiment and the near field radiated power is obtained using theoretical expression. Experimental radiation resistance obtained from the far field sound radiation is given in Figure 8 and Table 1. Results are obtained in 1/3 octave bands.

4. DISCUSSION OF RESULTS

Radiation resistance estimated using the present expression can be compared with the results obtained experimentally. They are given in Figure 8 and Table 1.

Theoretical estimates are made based on equation (11), considering twice the area of the plate as radiating area. Since the panel is unbaffled, a factor of 0·5 is applied on the radiation resistance obtained by equation (11) for frequencies up to half the critical frequency. Factors for short-circuiting effects are used as per equations (12) to (15). The boundary of the panel is assumed to be simply supported.

Experimental radiation resistance above 5000 Hz is not given here since it is expected to be in error. This is because the measured vibration levels of the plate are very small in 8000 Hz and 10000 Hz frequency bands. Since $\langle v^2 \rangle_x$ is in the denominator of equation (25), the radiation resistance calculated is in large error. Also, the room absorption used for calculating the radiation resistance seems to be in error at higher frequencies. It can be seen from Figure 6 that the room absorption suddenly increases at higher frequencies. Results are not expected to be very accurate at frequencies below 160 Hz since the number of modes in those bands are less than six. The experimental radiation resistance below 250 Hz tends to be over-estimated since the room absorption estimated is higher than the actual value. Hence, the results are given for 315 Hz and above only.

It can be seen that the experimental results and the theoretical estimates match very well except at very low frequencies. In this frequency range the boundary conditions are expected to play a significant role. For a plate with free edges this factor is very near to zero and hence the measured radiation resistance is very small at low frequencies. The experimental results validate the factor 0·5 suggested for unbaffled plate.

It is interesting to compare the experimental results with the theoretical estimates obtained using the existing expression for radiation resistance (equation (3) without the factor for baffle and short-circuiting effect). Above the critical frequency they match well (above the critical frequency both equations (3) and (11) are the same). But below the critical frequency radiation resistance values estimated using the existing expression are very much higher than the experimental radiation resistance values.

If, as Clarkson and Brown [11] suggested, plate area is taken as the radiating area, the experimental results match well with the theoretical estimates (equation (3) with effect of baffle and short-circuiting) only up to the critical frequency. Above the critical frequency experimental radiation resistance is twice that of theoretical estimates meaning that radiating area should be twice the area of the plate.

5. CONCLUSIONS

A new expression for estimating the radiation resistance of panels is suggested. Radiation resistance estimates using this expression are half the values estimated using the existing expression up to and including the critical frequency. Above the critical frequency they are the same. When both sides of the panel radiate, the radiating area is twice the plate area. The effect of the presence of other structural elements should be considered in the calculation of radiation resistance up to half the critical frequency. This factor is 0·5 for unbaffled panels. Apart from the above, factors for the effect of boundary conditions and short-circuiting inertial flow from one side of the plate to the other side should be used. The above results match very well with the experimental results. At low frequencies the sound power radiated by the near field of the excitation is very significant.

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APPENDIX A: LIST OF SYMBOLS

a, b	dimensions of the panel
A	area of the panel
c	speed of sound in air
d	mean free path
E_1, E_2	mean energy of sybsystems 1 and 2
E_{1m}, E_{2m}	mean modal energy of subsystems 1 and 2
f	frequency in Hz
f_c	critical frequency
$f_m(x)$	mode shape of the structure
$f_r(x)$	mode shape of the acoustic space
F_{corner}	correction factor for corner mode
F_{edge}	correction factor for edge mode
F_{plate}	correction factor for plate
k	wavenumber
m	mass per unit area of the panel
n_1, n_2	modal density of subsystems 1 and 2

N_1, N_2	number of modes of subsystems 1 and 2 in a frequency band
p	perimeter
p^2_{rms}	mean square value of pressure
R_{rad}	radiation resistance
$R_{rad,c}$	radiation resistance of corner mode
$R_{rad,e}$	radiation resistance of edge mode
s	surface area of the chamber
v	velocity of the plate
V_1	volume of subsystem 1
W	sound power radiated
α	sound power absorption coefficient
η_1, η_2	dissipation loss factor of subsystems 1 and 2
$\eta_{1,2}$	coupling loss factor for subsystems 1 to 2
$\Delta\omega$	frequency band
λ_a	wavelength in air
λ_b	wavelength of the bending wave in the plate
λ_c	wavelength at critical frequency
ω	circular frequency in rad/s
π_1	mean power input to subsystem 1
$\pi_{1,2}$	mean power flowing from subsystem 1 to 2
ρ	density of air
$\langle\rangle$	ensemble average