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PRACTICAL EXPERIENCE IN THE MODELLING OF FACTORY SOUND FIELDS

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1. INTRODUCTION

This paper describes the development and application of computer models used for the prediction of noise levels in Unilever factories. Computer simulation of noise levels in packing halls was first considered by SEAC Manufacturing in 1978 when Unilever was planning to build a new food production/packaging facility in Italy. In order to conduct a cost benefit analysis of alternative acoustic building treatments and source specifications, a computer model was constructed. Since the development of the first modelling programme the techniques have been developed and applied in around 20 projects both at the design stage of new facilities and to evaluate alternative noise control strategies in existing factories. A variety of approaches have been used dependant upon the complexity of the application. Before modelling a new or an existing installation, a critical review of the reasons for the model and the availability data has to be conducted to ensure that the complexity of the model is commensurate with the application.

2. DEVELOPMENT OF COMPUTER MODELLING TOOLS

2.1 The First Model

It is now well established that diffuse field theory is only accurate for prediction of steady state sound pressure levels in relatively small, regular enclosures. In fact, in a paper presented in 1952 H. J. Sabine (1) noted that, in rooms with floor dimensions which are large in comparison to the ceiling height, the level of reflected noise does not remain constant but drops off continually with increasing distance from the source. Even now some practitioners still use diffuse field theory to predict sound levels in workrooms where true diffuse field conditions will not exist.

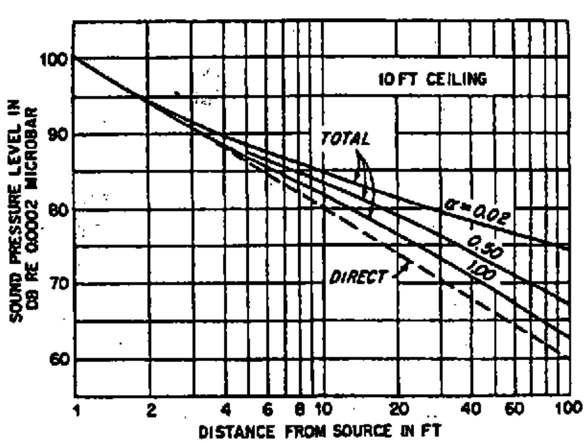

FIGURE 1. Sound Propagation Curves

In his paper Sabine proposed a family of propagation curves (figure 1) which present the reduction in sound pressure with distance from a single source for different ceiling absorption coefficients. The curves are plotted for a ceiling height of 10 feet (3.3m) but can be used for different ceiling heights by diagonally shifting the curves parallel to the line representing the direct level. These curves were derived from geometric analysis of multiple reflections between an infinitely extended floor and ceiling with no wall reflections. It was assumed that the source was midway between the floor and ceiling and that, for an absorptive ceiling, at large distances from the source, the reflected noise is due to the single reflection from the floor. The concept of scattering from fittings was not included in this model hence the limiting slope of the propagation curve is 6dB per doubling of distance (dB/dd).

It was these curves that formed the basis of the first Unilever computer model. The concept of the model was very simple. The distance from the centre of each source to points 1m apart across the floor of the building was calculated. Source sound powers were estimated from field measurements and the appropriate corrections were deducted to determine the 1m sound pressure level. The slope of the relevant propagation curve (figure 1) was computed by linear regression and this was used in the computation of the reduction in sound pressure between the source and the receiver position. The contribution of each source was then logarithmically summed to determine the combined effect of sources throughout the building. This approach was first applied to estimate the likely effects of acoustic treatment of the ceiling of a new factory in Italy. Unfortunately, at that time, an appropriate hygienic acoustic treatment which could be applied directly to the underside of the concrete ceiling could not be found hence the model was never validated.

2.2 Model Development

An alternative method of estimating propagation curves was subsequently implemented by utilising the empirical equations published by Friberg (2). This approach enables the slope of the propagation curve to be estimated from a linear equation whose coefficients are defined in tables. The values of the coefficients depend upon the room shape and the density of fittings. However, no information is provided regarding the absolute levels of the propagation curves. This option, together with the option to input measured propagation slopes, formed the basis of the modelling tool which subsequently evolved. For reference, calculation of noise levels has always been conducted in dBA as the spectral content of sources has been found to be similar and concentrated in the 500 to 2000 Hz octave bands.

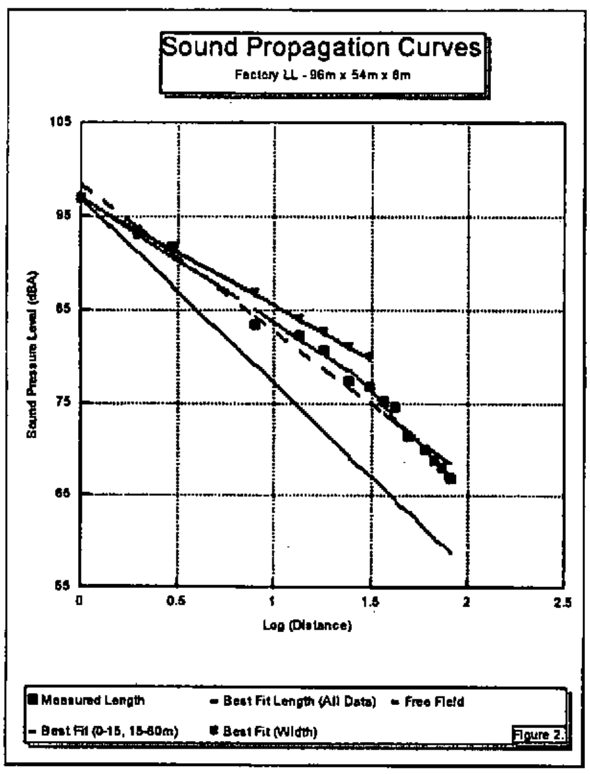

Figure 2.

In the model it was assumed that the sound propagation curves do not vary with source / receiver position and that they can be approximated by one or two straight lines, characterised by their slope and intercepts. In applications such as production / packing halls where the sources are of fairly uniform height and distribution this assumption has been validated by measurement.

Examples of the propagation curves measured in a production hall 96m long by 54m wide by 8m high are shown in figure 2. In this factory there are 10 parallel filling / packaging lines with a spacing of approximately 5m. Typical machine heights were approximately 2m. The floor of the building was painted epoxy, the walls painted block and the roof multi-layer decking supported by steel trusses.

On the graph the noise levels measured (in dBA) at distances from a test source are plotted together with the best fit curves computed by linear regression. The twin regression lines computed from the data for 0 to 15m and 15 to 80m gave the best fit to the measured data in this case.

The key input options of the calculation programme are as follows:

i) source position, sound power;

ii) source directivity (in 2 dimensions defined as corrections to the sound power in each of six segments);

iii) a constant background noise level (if required e.g. for ventilation noise);

iv) room dimensions;

v) receiver grid spacing;

vi) co-ordinates of a ceiling zone having a different acoustic absorption to the rest of the ceiling;

vii) the rate of decrease of sound with distance as either a single or double slope inside or outside the zone;

viii) input of environmental correction factors for sources inside or outside the zone.

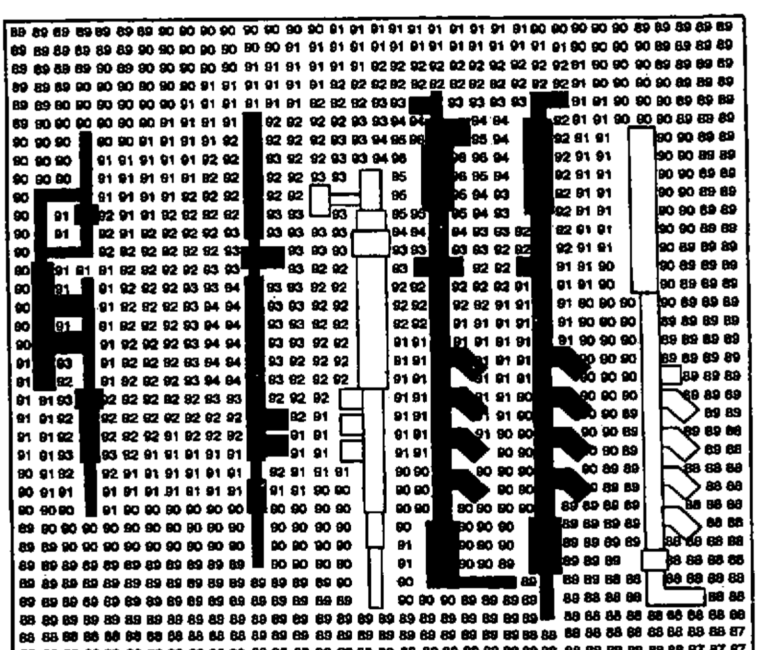

Figure 3

The output of the programme is a matrix of numbers (see figure 3) representing the calculated sound pressure level at points across the floor of the building. When necessary, for presentation purposes, colour contour plots are produced with the machine layout overlaid.

A case study discussing the application of this modelling tool is presented in reference 3.

2.3 Ray Tracing

Since Friberg published his paper in 1975, several workers have investigated various analytical methods for predicting the sound propagation in fitted rooms such as factories. Hodgson (4) has evaluated the accuracy and applicability of seven models. The models were the method of images models of Jovicic (5), Lindqvist (6), Hodgson(7), Kurze (8) and of Lemire and Nicholas (9), the empirical formula of Friberg (2) and Ondet and Barbry's ray-tracing model (10). In each case he compared the predicted sound propagation curves with those measured in a 1:50 scale model and with data measured in a warehouse when empty and containing parallelepipedic, isotropically distributed fittings Several of the models were found to be accurate under certain circumstances. However the ray tracing model of Ondet and Barbry was found to be the most accurate method of predicting the propagation curves in all applicable cases. A key feature is that it is applicable to non-parallelepipedic rooms and rooms with non-isotropic fittings.

Ray-tracing techniques require careful preparation of input data and can involve long computation times, particularly if large numbers of sources are involved. In some cases a full ray tracing simulation can be justified. However, for duct shaped buildings with a uniform distribution of machine sources, it can be unnecessary. A compromise approach, which has been used in practice (11,12) is to use the ray tracing algorithm to compute the expected noise level at distances down and across a building from a single source. The propagation curves for the building can be plotted from these data and the slopes determined. These slopes can then be input into the calculation programme to compute the combined effect of the 'real' machine sources.

If the building exists then the computed propagation curves can be compared with measured data and if necessary the fitting density, fitting absorption and surface absorptions can be adjusted until a good agreement is achieved. The effects of changing surface absorption can then be simulated with some degree of confidence.

The advantage of this approach is greatly reduced computation time with little reduction in accuracy. This is particularly important when conducting sensitivity analyses of different scenarios. A similar approach is used in certain commercial models used in Germany which implement the DIN standard. However, these models utilise a method of images approach for computing the sound propagation curves which have been found to be less accurate than ray tracing.

With any analytical approach there is potential uncertainty when defining the input parameters. Hodgson has investigated the criteria for defining input data for ray tracing models (13). Criteria have been defined for surface absorption, fitting absorption coefficients and scattering cross sections . Although certain assumptions are still required when assigning values, prediction of the sound propagation using ray tracing is the preferred approach in critical or complex applications.

3. PRACTICAL CONSIDERATIONS

3.1 Choice of Model

The output of any computer model is only as good as the input data. Accurate estimation of source sound powers is clearly important. The European Council Machinery Directive 89/392/EEC has an explicit requirement (Annex B 1.74(f)) for suppliers to provide instructions which give information regarding the noise emission of equipment. However, in practice, many suppliers do not provide this information. This is a key area where action is required. The following practical questions need to be considered when deciding upon the complexity of modelling to be conducted:

i) What is the purpose of the model;

ii) What can be decided without a model and what accuracy is required to influence a decision;

iii) Is the building geometry so simple that theoretical methods add little value compared with empirical methods ?

iii) Is accurate noise data available for all significant sources ?

iv) What are the potential cost implications of the scenarios being modelled. Is the effort/cost associated with refining a model justified ?

In duct shaped packing halls the initial slope of the propagation curves has been found to vary over a limited range of values (3.0 to 5.5 dB/dd) and simplified rules have been derived for the environmental correction which defines the intercept of the curves (14). In applications where there is a uniform distribution of sound power, inaccuracies in predicting the slopes of the propagation curves are less important as the noise level at a particular point is dominated by the contributions of sources within approximately 10m. In such cases, and when the purpose of the model is to evaluate the 'order of magnitude' effects of changes, empirical models can be appropriate.

The most recent empirical model is that proposed by Hodgson (15) and the accuracy of this model has been compared with seven other models (16). The conclusion from this study was that the Kutruff (diffuse) model (17) and the Hodgson model were the most accurate. It is not surprising that the Hodgson model performed well in this study as it was derived from the same database as that used to test the other models. This model is currently being evaluated as it appears to provide the appropriate level of accuracy and functionality.

3.2 Design Guidelines

Over the last twenty years computer models have been used, in Unilever, at the design stage of around 15 projects to identify the appropriate machine noise specifications and the effects of acoustic treatment. In addition models have also been constructed to assist in the evaluation of options of alternative noise control strategies in several production facilities.

Unilever has a design target of 83 dBA LAeq in the workplace for new manned facilities to ensure compliance with an employee exposure level of 85 dBA LAeq (8Hr) (allowing for 12 hour shifts). Experience with modelling has enabled the following design guidelines to be established for packing halls:

Background noise target for ventilation systems - 65 dBA. At this level the ventilation system will not contribute to operator exposure levels.

Combined background noise target for background sources (e.g. conveyors, vacuum systems) - 73 dBA. These sources can often be overlooked and contribute significant sound power when extended throughout the building.

Packaging machine noise 'alarm level'- 75 dBA at 1m.

The 'alarm level' is intended to be the basis for decision as to whether noise is likely to be a problem and whether more detailed noise data is required. If these specifications cannot be met then a level of modelling would be conducted to establish appropriate machine noise specifications and to evaluate the effect of other options to reduce noise exposure.

4. CONCLUSIONS

To assess the requirements to meet a given hearing conservation criterion the acoustic practitioner needs both modelling tools and data. The sophistication of the tools and the accuracy of the data has to be commensurate with the scale of the project and the consequences of not achieving the design target. In collaboration with UBC Vancouver, predictions based on empirical methods, method of images and ray tracing have been compared. In general, for the simple building geometries, empirical methods have proved adequate particularly when there is a fairly uniform distribution of machinery noise sources within the building.

Fundamental to any noise prediction is machine sound power data. Tighter enforcement of the requirements of the machinery directive (for machines manufactured inside and outside the EC) to provide noise data, and ideally the provision of a central EU noise database, would be of considerable benefit.

Consideration should be given to developing a 'noise screening model', for use by non-experts, which would enable basic information on the building geometry, machine numbers and sound power to be input. The output would then be an indication as to whether employee exposure would be likely to exceed 85 dBA LAeq or whether employee noise exposure should be acceptable. The ultimate aim is to reduce employee noise exposure, to the lowest level reasonably practicable. Modelling helps to define the requirements, it does not solve the problem. Models should be 'user friendly' and not require prohibitive data preparation and computation times.

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AN ENGINEERING APPROACH FOR THE ACOUSTIC CHARACTERISATION OF APERTURE DEVICES

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1. INTRODUCTION AND BACKGROUND

There are many engineering applications that require transfer of fluid from one spatial domain to another via smaller or larger openings. One such obvious application is ventilation in living and working spaces where air is to be transported either between rooms or between the outside world and a room with the hope of getting fresh oxygen to its inhabitants. In many cases the airflow has a low mean velocity but especially for working spaces forced ventilation might be required and the flow velocity is significantly elevated. Another related area is cooling of machinery or plants such as in transportation vessels and chemical or energy plants. Again, the mean flow velocity varies from that of self-circulation to forced. Common for all applications is that some kind of device separates two fluid domains and with the device having several different objectives. Among such objectives, sound reduction or in more recent terms "sound styling" is frequently prominent. Regarding the acoustic aspects, the devices generally establish the interface between two differing acoustic fields such as the free, outdoor sound field and the reverberant indoor field. In addition, the device can be situated in a shorter or longer duct involving yet more field parameters.

A review of the pertinent literature [1] indicates that there is a wealth of knowledge related to specific applications. Unfortunately, the different motives for the work as well as varying frequency ranges and experimental methodology make the combination of data and comparisons of results rather problematic. By considering the interplay between the size of an aperture and the wavelength, a Helmholtz number, based on the wavenumber multiplied by some characteristic dimension of the aperture, was introduced in an attempt to obtain some overview of earlier work. Even so a lucid picture is difficult to establish owing to the different controlling parameters and mechanism underlying the attenuation.

The very wide range of applications for aperture devices with possibly conflicting objectives and increasing requirements regarding the acoustic aspects allowing for an arbitrary mean flow makes it technologically relevant to attempt a unified, generic treatment of the associated acoustic effects.

2. APPROACH

It is well known that a simple hole in a partitioning wall can, depending on its size, offer some acoustic reduction at certain frequencies. A way that the end user might view or, perhaps, should view the problem is indicated in Figure 1. Optimisation in this context incorporates the matching of the device to the surrounding fields as well as accounting for any effect of elevated mean flow velocity.

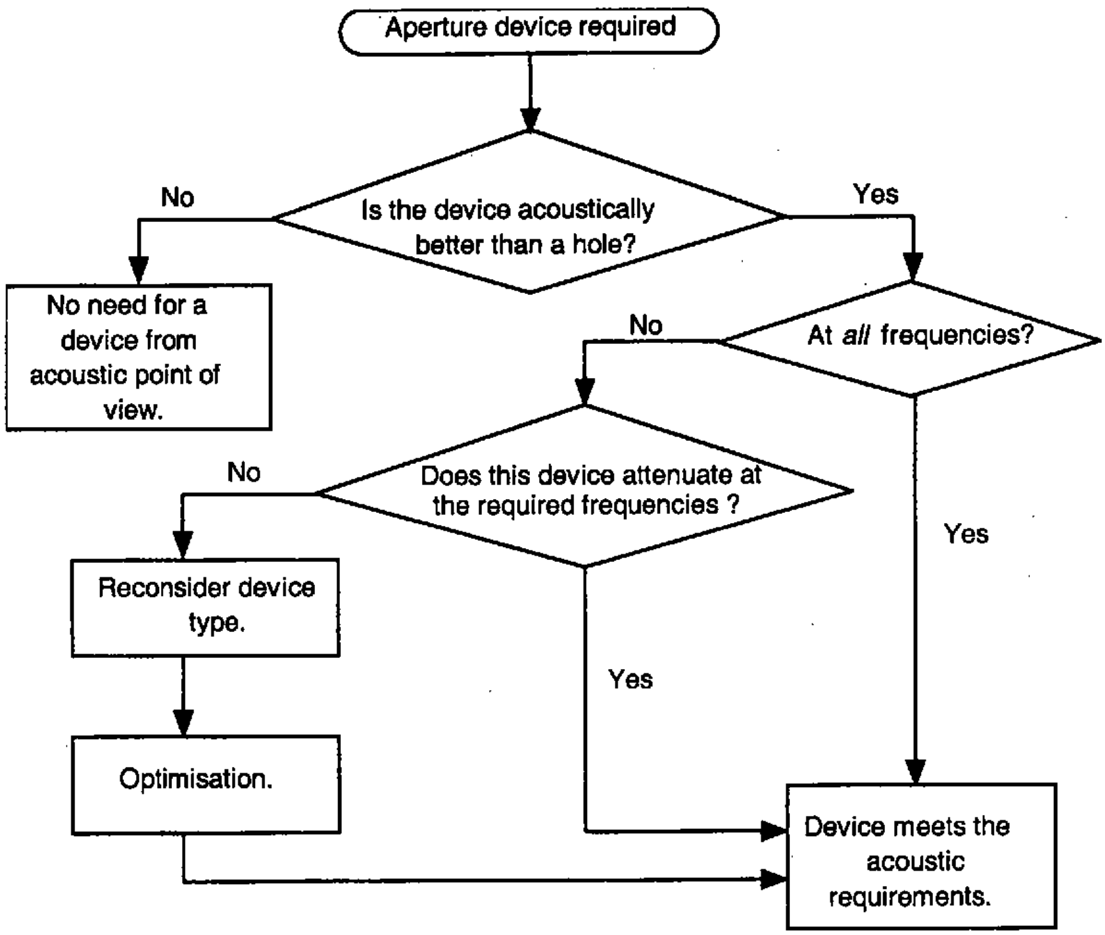


Figure 1. Idealised approach for engineering practice.

Accordingly, it can be argued that the myriad of configurations and the organisation of existing data makes it very time consuming and difficult to achieve good solutions in engineering practice and the generic formulation mentioned above stands out as even more worthwhile. Upon applying a 'systems engineering approach' it is desirable to be able to assess the effects of a device decoupled from both any exterior or duct field. Since rigorous descriptions of those fields are available, the designer is then free to advice and select any device to match a particular situation.

With the terminology borrowed from transmission line theory an impedance formulation can been adopted and the aim of the present contribution is to shed some light on the implications of as well as conditions for the application of such an approach. In practice, with a cross-sectionally, arbitrarily shaped duct in which the device is situated, it is not sufficient to just consider the plane wave transmission but higher order modes and scattering must also be taken into account. To retain an impedance formulation, an expansion of the field in the modal components becomes necessary and thence to identify the individual modal contributions to the total field.

3. BASELINES FROM THEORY

To arrive at a practical approach for the assessment of devices or design of duct/device/duct configurations with high acoustic performance, it is useful to establish some of the theoretical baselines. Consider therefore at first two semi-infinite spaces separated by a partitioning wall having a small hole as depicted in Figure 2.

In Figure 2 is illustrated the well-thumbed low frequency or more correctly, the small Helmholtz number approximation as deduced from a matched asymptotic expansion, see e.g. [2]. It is assumed that a plane wave is incident upon the orifice in the large wall with a wavelength which is substantially greater than the dimensions of the orifice. At the incident side of the wall, at some distance, the field is thus composed of the incident and the (specularly) reflected waves plus a spherical wave associated with scattering due to the change in impedance at the orifice. The resulting field at the transmitted side, again at some distance from the wall, is described by the (spherically) diffracted wave. In the vicinity of the orifice the strictly, somewhat more involved field is 'lumped' as an equivalent volume flow. It is seen that this modelling leads to equal transmitted and net reflected contributions.

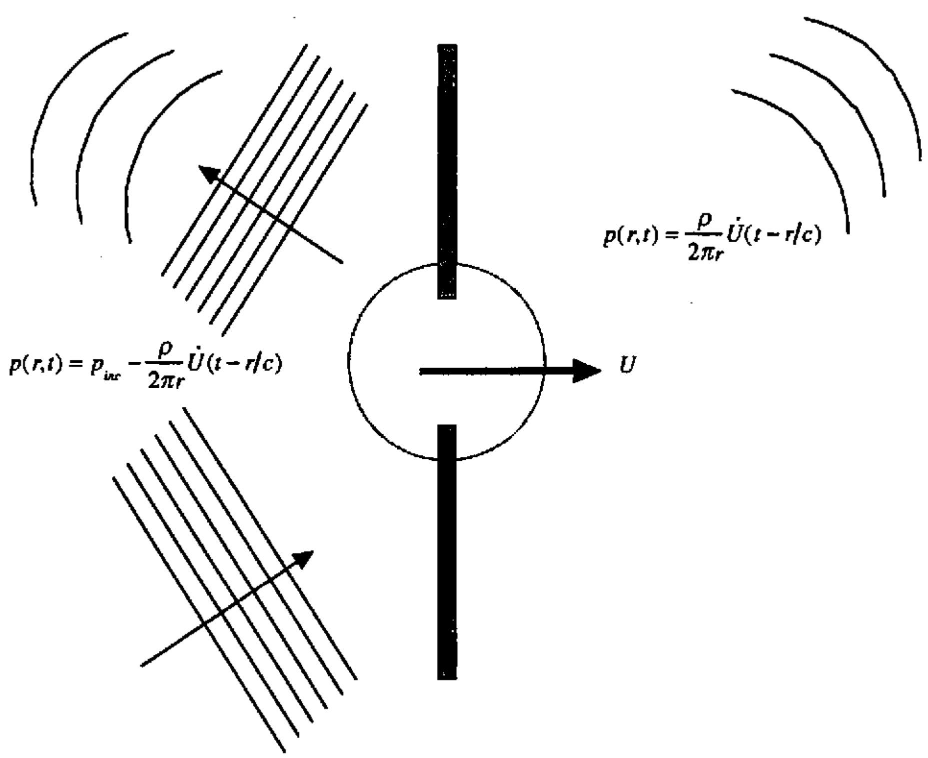
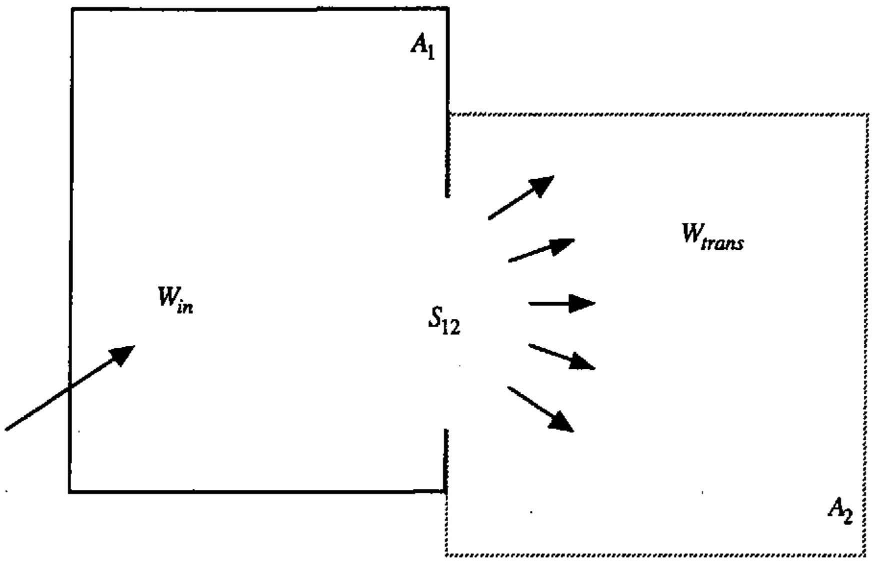


Figure 2. Small Helmholtz number approximation.

Another baseline can be obtained for diffuse or single sided diffuse fields as illustrated in Figure 3.



$$\frac{W_{trans}}{W_{in}} = \frac{1}{1 + A_1\left(\frac{1}{S_{12}} + \frac{1}{A_2}\right)}$$

Figure 3. Diffuse to diffuse field or diffuse to free field transmission.

Here, a suitable starting point is the case of two reverberant rooms coupled via a large opening, see e.g. [3]. Power is fed into the source room and part of it is absorbed in the room and part is transmitted to the receiving room and dissipated there. Similar to the case of sound transmission through a wall the opening between the rooms is assigned a transmission efficiency (transmission coefficient based on power). For the opening a unity transmission efficiency is assumed and hence only its area appears in the ratio of transmitted to incident power. If now the receiving space is not reverberant but, say, a free outdoor sound field, the absorption area A_2 becomes infinite and the power ratio is further reduced to involve only the absorption area of the source room to the open area.

For many of the applications mentioned in the introduction, however, the typical dimensions of the aperture is of the order of the wavelength of the sound and neither of the two asymptotic cases are applicable. Moreover, both the asymptotic descriptions effectively circumvent the features of any device. Another limiting factor for industrial and engineering applications is the possible flow through the aperture and device. To allow for an assessment of at least the first order effects accordingly, any attempt to unify the treatment and develop a useful approach for engineering practice must be based on a more detailed description.

Revisiting the plane wave and small Helmholtz number scenario, transmission line theory may be invoked and the device assigned a four-pole description, see Figure 4. This means that the device and duct is described by an impedance matrix, $\mathbf{Z}_{DD}$. For plane waves in the duct, sub-matrices for the parts before and after the device can be directly established such that the effect of the device itself can be identified. Conversely, with the effect of the device known, the usually sought transfer impedance of the complete configuration can be computed,

$$Z_{DD,12} = \frac{Z_{A,12} Z_{D,12} Z_{B,12}}{\left(Z_{A,22} + Z_{D,22}\right)\left(Z_{B,11} + Z_{D,11}\right) - \left(Z_{D,21}\right)^2} . \tag{1}$$

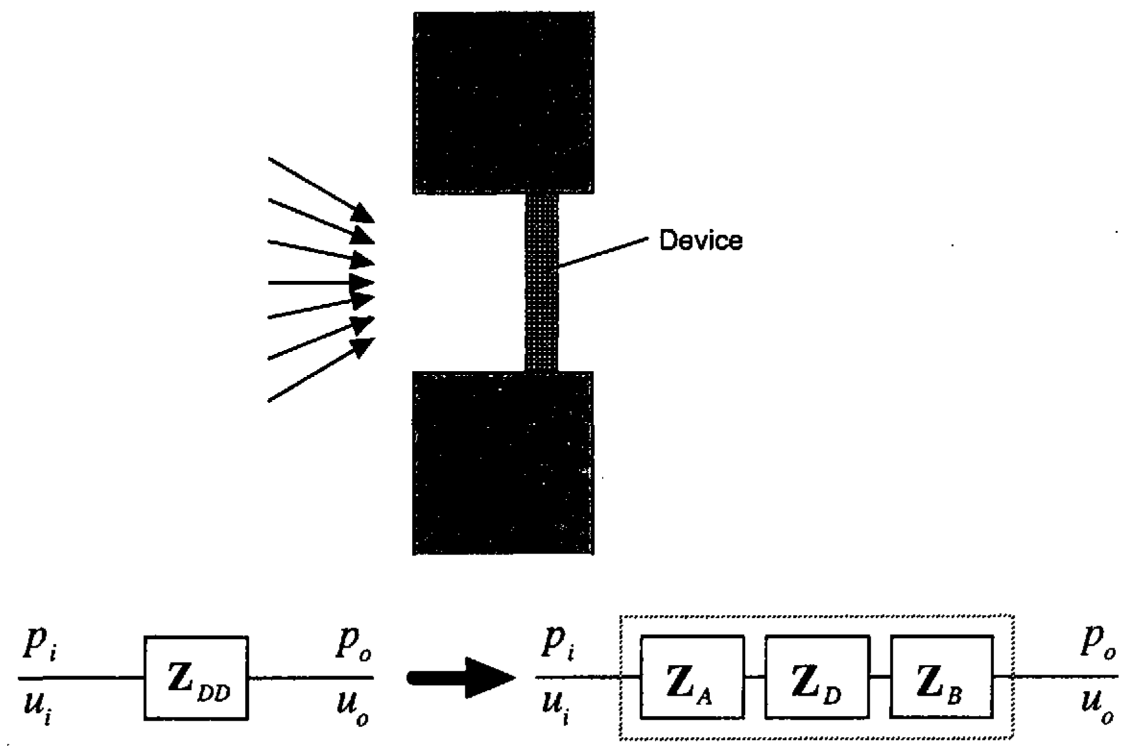


Figure 4. Schematic diagram of the impedance formulation.

As such, the expression for the transfer impedance can also be made to account for the impedances of source and receiver spaces.

For the practical case where the dimensions of the aperture grow large in comparison with the wavelength in the upper frequency range, the impedance approach outlined above is no longer directly applicable. This is due to the presence of higher order modes in the duct and some extension is necessary to accommodate for their contributions. Such an extension is enabled through an expansion of the duct field in terms of its cross-sectional modes and by virtue of their orthogonality, component-wise modal impedances can be developed. Hence, a sum of expressions similar to that in Eq. (1) results to establish the sound transmission. The apparent simplicity of a one-to-one correspondence of the modal components, however, is hampered by the scattering and thence potential cross-coupling between various modes. Although the formulation can be developed rigorously, the preservation of the apparent simplicity for engineering practice is highly desirable and it is thus justified to investigate the implications of cross-coupling.

To facilitate the mathematical treatment the analysis can be confined to a circular geometry without sacrificing salient physics. Thus, a circular orifice of radius R in a rigid wall can be considered. Following [4], the expression for the wavenumber domain spectrum of the scattered velocity potential is obtained as

$$\tilde{\phi}^{s_m}(\zeta) = \frac{-i}{\kappa}\sum_{n=1}^{\infty} k_z \left(b_n^m \cos k_z d - c_n^m \sin k_z d\right) I^m\left(k_n,\zeta\right), \tag{2}$$

in which the cross-coupling of modal components is manifested by the integral

$$I^m\left(k_n,\zeta\right) = \begin{cases} \dfrac{R}{k_n^2-\zeta^2}\left[k_n J_{m+1}\left(k_n R\right) J_m(\zeta R) - \zeta J_m\left(k_n R\right) J_{m+1}(\zeta R)\right]; & k_n \neq \zeta \\ \dfrac{R^2}{2}\left[J_m^2\left(k_n R\right) - J_{m-1}\left(k_n R\right) J_{m+1}\left(k_n R\right)\right]; & k_n = \zeta \end{cases} \tag{3}$$

where k_n is the modal wave number and ζ the recoil wavenumber. Accordingly, it is of interest to consider this term in more detail. In Eq. (2) c_n^m and b_n^m are the amplitudes of the waves in the orifice and the parenthesis may therefore be considered the source for the scattered field. Since this factor depends on both the circumferential order m and radial order n, each mode will have different excitation strength in the general case.

4. SOME NUMERICAL RESULTS

In Figure 5, $I^m(k_n,\zeta)$ is plotted against ζ/k_n for the (m,n) modes (0,2) and (0,3). The scattered field will only propagate when ζ/k_n is greater than unity. It is not possible to show the plane wave mode (0,1) in this normalised form as k_n for the plane wave mode is zero.

Although, for clarity, only two $(0,n)$ modes are shown, all higher $(0,n)$ modes have a maximum at unity ζ/k_n but reduce in magnitude as n increases. Figure 5 also shows $I^m(k_n,\zeta)$ for the modes 1,1, 1,2 and 1,3. It is observed that all the modes have a maximum at ζ/k_n=1, apart from the 1,1 mode. This pattern repeats for higher orders (m,n) where, for modes with n =1, the contribution is a maximum slightly above unity while for modes with n>1, the maxima occur at unity.

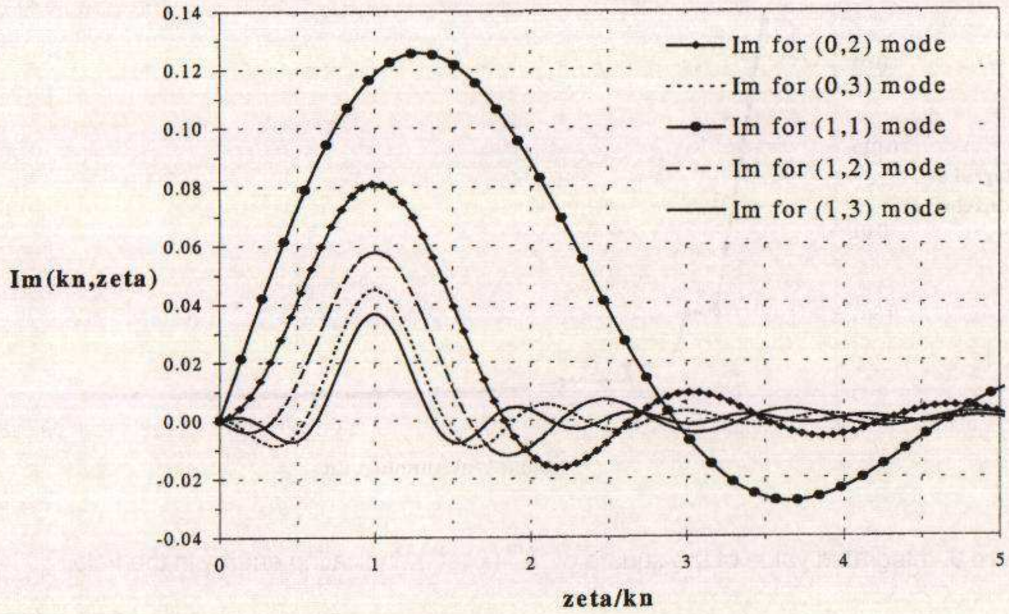


Figure 5. $I^m(k_n,\zeta)$ for various modes.

In order to assess the contribution of each mode to the scattered field, the integral of $I^m(k_n,\zeta)$ has been evaluated. Using the integrated value of $I^m(k_n,\zeta)$ versus modal wave number k_n, on a log-log scale, two straight-line relationships are observed; one line running through the (*m*,1) data points and the other through the (*m*,*n*>1) data points. Simple curve fitting provides the following approximations for the integral of $I^m(k_n,\zeta)$.

$$\int I^m(k_n,\zeta)\,d\zeta = ak_n^b\ ; \qquad \begin{array}{lll} (m,1) & a = 0.6318 & b = -1.8819 \\ (m,n>1) & a = 1.1939 & b = -2.0683 \end{array} \qquad (4)$$

If $I^m(k_n,\zeta)$ is considered the coupling strength, then the square of $I^m(k_n,\zeta)$ will be proportional to the energy in the field. Figure 6 shows the integrated value of the square of $I^m(k_n,\zeta)$ versus the modal wave number. Again, two straight lines can be established on a logarithmic scale. These correspond to the (*m*,1) and (*m*,*n*>1) lines observed earlier. The following simple expressions for the square of $I^m(k_n,\zeta)$ has been determined,

$$\int \left(I^m(k_n,\zeta)\right)^2 d\zeta = ak_n^b; \qquad \begin{array}{lll} (m,1) & a = 0.1380 & b = -2.9796 \\ (m,n>1) & a = 2593 & b = -2.9379 \end{array} \qquad (5)$$

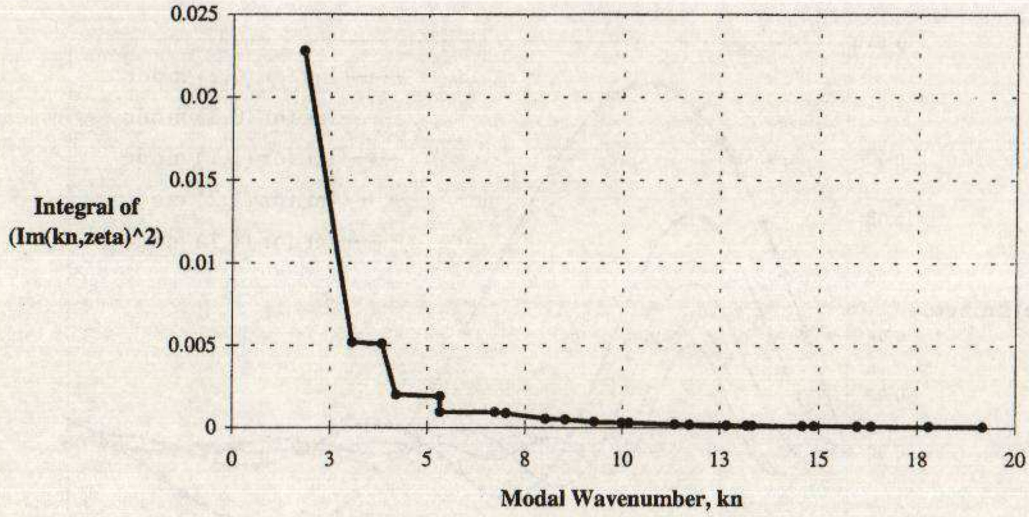


Figure 6. Integrated value of the square of $I^m(k_n,\zeta)$, indicating energy in the field.

Using Eqs. (4) and (5), it is possible to estimate the modal coupling contributions to any mode (*m*,*n*) of interest. Obviously, from Eq. (5), for any *m* set of modes, the relationship between the value of the integral of the squared quantity for the first circumferential (n =1) mode and any other *n* mode may be determined. For example, the 1,2 mode contribution is 8% of the 1,1 mode contribution and the 1,3 is 2% of the 1,1, mode.

5. CONCLUDING REMARKS

Based on the discussion of the requirements on an engineering approach for the assessment of the acoustic effect of aperture devices it is tentatively concluded that a viable basis is established through an impedance formulation. The lucidity as well as manageability of the approach sketched is allied with the influence of modal coupling on the scattered and transmitted fields. The presented numerical comparison of the modal contributions to the scattered field from an orifice in a rigid baffle indicates a rapidly decreasing strength of the modal coupling with increasing difference in modal order. From the simple approximations developed it is possible to establish the error, in energy terms, incurred by neglecting higher modes. This error can then be used to compensate predictions based on a modal impedance approach when the effects of scattering are neglected.

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