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ON SOUND POWER DETERMINATION IN FLOW DUCTS

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The following problems associated with sound power determination in circular ducts carrying mean flow are discussed: axial standing waves due to sound reflections from the duct end; acoustic loading of the source; turbulent flow pressures superimposed on the sound field; discrimination between sound pressures and turbulent flow pressures; radial measurement position in the duct in view of higher order mode sound propagation and directional characteristic of the microphone probe used; modal distribution of sound power. Early and recent work on the above topics is reviewed. Brief descriptions of the standardized in-duct method ISO 5136:1990 (*International Organization for Standardization, Geneva, Switzerland, International Standard*) [1] and the revised measurement procedure ISO/DIS 5136:1999 (*International Organization for Standardization, Geneva Switzerland, Layout for a Draft International Standard*) [2] are given.

1. INTRODUCTION

The determination of sound power in a flow duct is of practical interest because many air moving devices are connected to a duct, the most common examples being fans. In the late 1960s, an international working group was set-up (ISO/TC 43/SC 1/WG 3, "Noise from heating, ventilating and air-conditioning equipment") to establish an international measurement standard on "Sound measurement procedure for air moving devices connected to either a discharge duct or an inlet duct". Two Draft International Standards ISO/DIS 5136:1977 [3] and ISO/DIS 5136.2:1985 [4] were prepared before the first international standard ISO 5136:1990 [1] was published. In 1993, ISO 5136 was adopted as a European Standard, EN 25136 [5].

Practical experience with this standardized in-duct method indicated that technical problems existed with the standardized measurement procedure and with some of the frequency corrections used. In 1996, ISO/TC 43/SC 1 formally decided to set-up a new working group ISO/TC 43/SC 1/WG 47 to revise ISO 5136:1990 [1], and 3 years later the layout for ISO/DIS 5136 [2] was finalized.

In Section 2 of this paper, the general problems of sound power determination in flow ducts are reviewed. The standardized measurement procedure of ISO 5136:1990 [1] is described in Section 3. Practical experience gathered with this standard is summarized in Section 4. In Sections 5 and 6, theoretical and experimental studies are presented which were carried out by DLR and the Technical University in Berlin to resolve some of the problems revealed in previous experiments involving ISO 5136:1990 [1]. In Section 7, finally, the revised standard ISO/DIS 5136 [2] is described.

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2. DESCRIPTION OF THE GENERAL PROBLEM

The following general problems exist when one attempts to determine the sound power of an air moving device radiated into a duct connected to it.

- The sound power generated by an acoustic source depends on its acoustic load, i.e., the acoustic impedance presented to its inlet and outlet side.
- The sound power propagating from the source down the duct is reflected at duct discontinuities and/or the duct end.
- Above a certain frequency, the sound pressure in the duct is not uniform over the duct cross-section but depends on the transverse position.
- The microphone in the flow duct is exposed not only to the sound pressures to be measured but also to the unsteady pressures associated with the turbulent flow; hence a special windscreen is required.
- In a practical measurement situation, it is difficult to decide whether or not the microphone signal is contaminated by the turbulent flow pressures, even when a windscreen is used.

2.1. AXIAL STANDING WAVES AND ACOUSTIC LOADING

Sound reflections occurring at the end of the test duct result in axial standing waves, and as a result, the sound pressure to be measured is dependent on the axial location. This problem can be solved by attaching an anechoic termination to the test duct. Many designs have been tested by various authors; most of them are depicted in ISO 5136 [1]. One example which was originally developed by Wollherr [6], and further developed by the first of the present authors, is shown in Figure 1. The use of an ideal anechoic termination provides a well-defined acoustic loading which is equal to the characteristic wave impedance ρc (ρ the density, c the speed of sound) and independent of frequency. In practice,

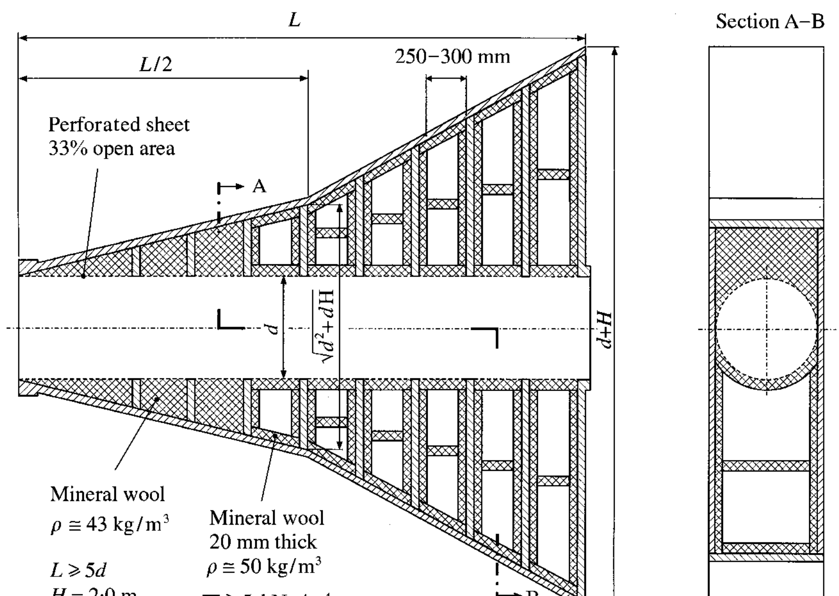

Figure 1. Schematic of an anechoic pipe termination.

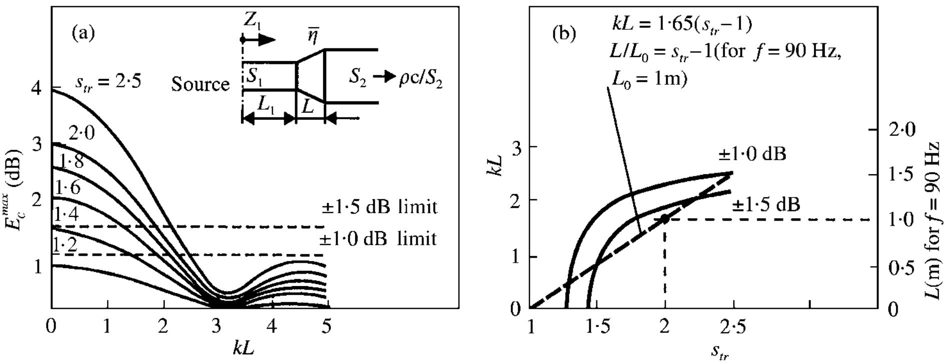

Figure 2. (a) Maximum error of the sound power transmitted into a duct with a transition piece relative to the sound power transmitted into a straight duct for a source of constant acoustic volume velocity (i.e., volume velocity independent of acoustic load); (b) necessary transition length for two error limits (after Bolleter *et al.* [7]).

of course, residual reflections from the duct termination exist. For a well-designed anechoic termination, the pressure reflection coefficient decays monotonically with frequency in the plane wave frequency range. The reflection coefficients of the higher acoustic modes are highest close to their cut-on frequencies with again a monotonic decrease with frequency; compare the results in the paper by Bolleter *et al.* [7].

Very often it is necessary to use transition ducts to connect the device to be tested to the measurement duct. It is well known that changes in cross-section involve sound reflections as well as changes in acoustic loading. This effect, among others, was studied by Bolleter *et al.* [7]. Figure 2(a) shows the maximum error in transmitted sound power caused by a transition piece of length L and area ratio $s_{tr} = S_{large}/S_{small}$ (k = wave number). The error in transmitted power is the difference between the sound power radiated by the source under test into the test duct via the transition piece and the sound power that would be radiated into a duct without a change in cross-section. In Figure 2(b), the $\pm$ 1·0 and $\pm$ 1·5 dB error limits are plotted in the kL–s_{tr} plane. This graph can be used to determine the dimensionless transition piece length kL necessary to meet either one of the two error limits as a function of the area ratio s_{tr}. The above considerations formed the basis of the minimum length requirement for transition pieces prescribed in ISO 5136 which, for the sake of simplicity, were specified as $L/L_0 \geqslant s_{tr} - 1$ ($L_0 = 1$ m). This limit is equivalent to the condition $kL \geqslant 1{\cdot}65$ ($s_{tr} - 1$) at 90 Hz, which is the straight line shown in Figure 2(b). When the maximum allowable area ratio $s_{tr} = 2$ is used, the resultant errors in sound power are 1·8 dB at 50 Hz, 1·7 dB at 63 Hz, 1·4 dB at 80 Hz, and 1·1 dB at 100 Hz.

The results shown in Figure 2 were arrived at by assuming an acoustic source of constant acoustic volume velocity, i.e., a source which produces the same acoustic volume velocity independent of its acoustic load. However, Bolleter *et al.* [7] claimed that the assumption of a constant acoustic pressure source leads to the same maximum errors.

2.2. HIGHER-ORDER ACOUSTIC MODE SOUND PROPAGATION IN THE TEST DUCT

Only when the sound wavelength is large compared with the cross dimension of a duct is the sound pressure uniform over the cross-section. If sound reflections from duct discontinuities are absent, acoustic pressure and acoustic particle velocity v are related by the plane wave relation $p/v = \rho c$, and determination of the sound power transmitted in a duct of cross-sectional area S without flow is given by $P = \overline{pv}S$, where the overbar indicates time averaging. In logarithmic form, the so-called plane wave formula reads

$$L_w = L_p + 10\lg(S/S_0) - 10\lg[\rho c/(\rho c)_0], \tag{1}$$

where $S_0 = 1\ \mathrm{m}^2$ and $(\rho c)_0 = 400\ \mathrm{N\,s/m^3}$ are the usual reference values.

In the frequency region where higher order acoustic modes can propagate, the sound pressure amplitude varies over the duct cross-section, and determination of the duct sound power requires integration of the sound intensity over the cross-section. For the no-flow case one has

$$P = \int_S \overline{pv}\,\mathrm{d}S. \tag{2}$$

The sound intensity measurements are time consuming because they require a large number of measurement positions in the duct and, hence, the technique is not very practical. Even worse, in the presence of superimposed mean flow the sound intensity technique is difficult, if not impossible, to apply due to the sound propagation characteristics in ducts. In the modern sound intensity measurement technique, the acoustic particle velocity is determined via the pressure gradient, more precisely via the pressure difference at two measurement points slightly separated in space. It was shown by Munro and Ingard [8] that “the acoustic intensity cannot be uniquely reconstructed from a measurement of the pressure and pressure gradient at a single point, when there is mean flow”, the reason being that there is no unique relationship between pressure gradient and acoustic particle velocity when the directions of flow and sound propagation are different.

A practical approach to sound power determination in a circular pipe was taken by Barret and Osborne [9]. They determined theoretically a radial microphone position in the pipe where the sound pressure is of such magnitude that application of the plane wave equation (1) gives the correct sound power even though higher order modes are present. Under the assumption that the first 10 propagational modes within each given frequency band carry the same acoustic power (mode model of “equal modal sound power”, EMSP), this “optimum radial position” is half-way between the pipe axis and the wall ($2r/d = 0{\cdot}5$), provided a microphone with uniform directivity is used.

Bolleter *et al.* [7] found theoretically that for a directional sensor like a microphone equipped with a turbulence screen (see the following section) the “optimum radial position” is closer to the pipe wall, where the mode amplitudes are generally larger, to compensate for the microphone directivity. For a slit-tube microphone with 400 mm effective length they recommended the following positions: $2r/d = 0{\cdot}8$ for pipe diameters from $d = 0{\cdot}15$ to $< 0{\cdot}50$ m, and $2r/d = 0{\cdot}65$ for $d = 0{\cdot}50$ to $2{\cdot}0$ m. Also, circumferential averaging at the specified radial position was found necessary to obtain a sound pressure amplitude representative for the sound power transmitted in the duct.

2.3. SUPPRESSION OF TURBULENT FLOW PRESSURES

The microphone placed in the duct is subject not only to the acoustic pressures but also to the pressure fluctuations associated with the turbulent duct flow. To suppress the effect of these turbulent pressures on the microphone, the use of a windscreen is necessary. Friedrich [10] introduced the concept of a long cylindrical windscreen for use in ducts, and Neise [11] later on suggested a slit-tube design, derived a one-dimensional theory for this design and experimentally verified the theoretical results. A sketch of such a windscreen is shown in Figure 3.

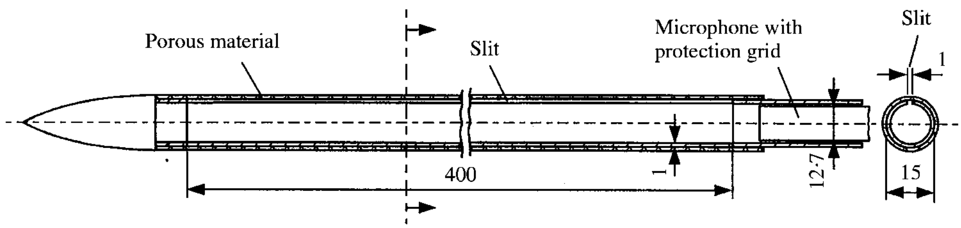

Figure 3. Schematic of a slit-tube windscreen for 0·5-inch microphones (from Neise and Stahl [12]).

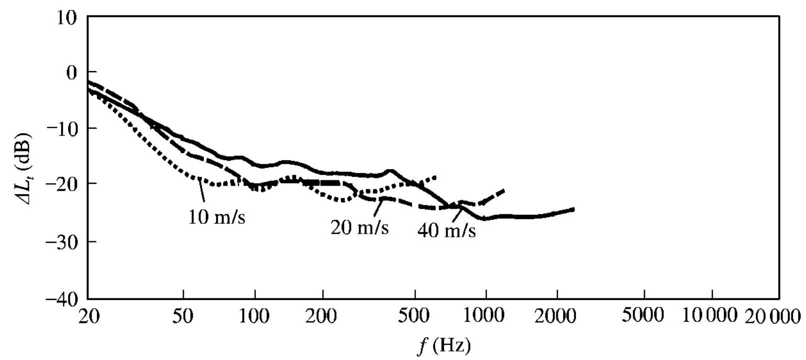

Figure 4. Difference between the flow noise levels as measured with a microphone with slit tube and by a microphone with nose cone (from Neise and Stahl [12]).

The basic principle of this flow noise suppressor can be explained as follows. A fluctuating pressure field outside the tube excites pressure disturbances along the inner tube wall. Inside the tube, the pressures are propagated as sound waves in both directions, i.e., from the point of excitation to the microphone as well as to the tip. For ease of explanation, it is assumed here that all sound waves are totally absorbed at the tip. The pressure at the microphone is then determined by the sum of the waves propagating from the various points of excitation along the slit length to the microphone. When adding them up one has to consider their phase shifts due to the different travelling times from the points of excitation to the microphone. If the propagation velocity of the external pressures is different from the speed of sound, the pressures at the microphone have different phase relationships, and hence the measured pressure is diminished. This effect becomes larger as the ratio of external wavelength to slit-tube length becomes smaller. Thus, the microphone will measure the correct amplitude of a plane sound wave travelling in the direction of the tube axis, while the turbulent pressure fluctuations which are convected at a velocity of the order of the flow velocity, $U_c < c$, are sensed to a lesser degree. Therefore, the ratio of acoustic pressures to flow noise pressures is increased. In Figure 4, the flow noise level of a microphone equipped with a slit-tube is compared with that of a microphone fitted with a nose cone. The experimental results were obtained in a "quiet" duct flow; see the papers by Neise [11] and Neise and Stahl [12]. The findings of these two studies were supported later by experiments performed by Shepherd and La Fontaine [13] and a theoretical analysis by Munjal and Eriksson [14].

Note that since the reduced microphone response is the result of a difference in the phase velocities inside and outside, sound waves propagating with or against the mean flow are also sensed to a lesser degree. The axial phase velocity of a sound wave that impinges on the slit-tube microphone at an angle is also different from the speed of sound, and this explains why this probe has a directional characteristic. This feature is important in view of the propagation characteristics of the higher order acoustic duct modes.

2.4. DISCRIMINATION BETWEEN SOUND PRESSURES AND TURBULENT FLOW PRESSURES

In the practical situation of sound measurements in flow ducts one has to ensure that there is a sufficient signal-to-noise ratio between the sound signal and the flow noise due to the turbulent flow at the microphone, even when a windscreen is used. Neise and Stahl [12] presented two procedures to determine the flow noise spectra.

The first method is based on experimental data for the flow noise spectra of 0·5-inch condenser microphones with nose cones in circular pipes as functions of the pipe diameter, the flow velocity, and the turbulence level. This latter information, however, is very seldom available, and therefore this method seems of limited practical value.

The second method is based on the assumption that the sound signal to be measured and the flow noise are mutually uncorrelated and on the experimental observation that there is a difference ΔL_t between the flow noise spectra of a slit-tube microphone and a nose cone microphone, e.g., as shown in Figure 4. Two measurements are necessary: one with a nose cone microphone and one with a slit-tube microphone.

For a given frequency band, the fluctuating pressure in a flow duct is the sum of the sound pressure in the duct, p, and the flow noise pressures, p_t. Upon the assumption of sound and flow noise being uncorrelated, one can write for the nose cone (NC) and the slit-tube (ST)

$$p_{NC}^2 = p^2 + p_{tNC}^2 \quad \text{and} \quad p_{ST}^2 = p^2 + p_{tST}^2. \tag{3}$$

The difference between the flow noise levels as measured with the nose cone, L_{tNC} and with the slit-tube, L_{tST} can be expressed as

$$\Delta L_t = L_{tST} - L_{tNC} = 20\log(p_{tST}/p_{tNC}). \tag{4}$$

Equations (3) and (4) can be used to derive relations for the true sound pressure level as well as for the flow noise level at the slit-tube microphone. It was shown by Neise and Stahl [12] that the requirement that the flow noise level be at least $\Delta L_{min} = 6$ dB lower than the sound pressure level reading of the slit-tube microphone is equivalent to the condition that the difference between the pressure level readings when using the nose cone and when using the slit tube must not exceed a maximum value ΔL_{max} which is a function of the flow noise suppression capability ΔL_t of the slit tube. The steps necessary for the second procedure are described in the original paper by Neise and Stahl [12] and also in ISO 5136 [1].

In the latter, a third method is described which involves a second sound measurement with a silencer mounted between the source under test and the microphone location. The silencer must have the same cross-sectional area and the same length as the replaced part of the test duct. The silencer shall have an insertion loss of at least 10 dB for each frequency band of interest. The requirement for the minimum signal-to-noise ratio of sound to turbulence noise of 6 dB is fulfilled if the average sound pressure level obtained with the silencer in place is at least 5 dB lower than without the silencer.

3. DESCRIPTION OF THE IN-DUCT METHOD ISO 5136:1990 [1]

The sound power radiated by a source into a duct depends on the type of duct, characterized by its acoustic impedance. Strictly, this statement applies to each mode of

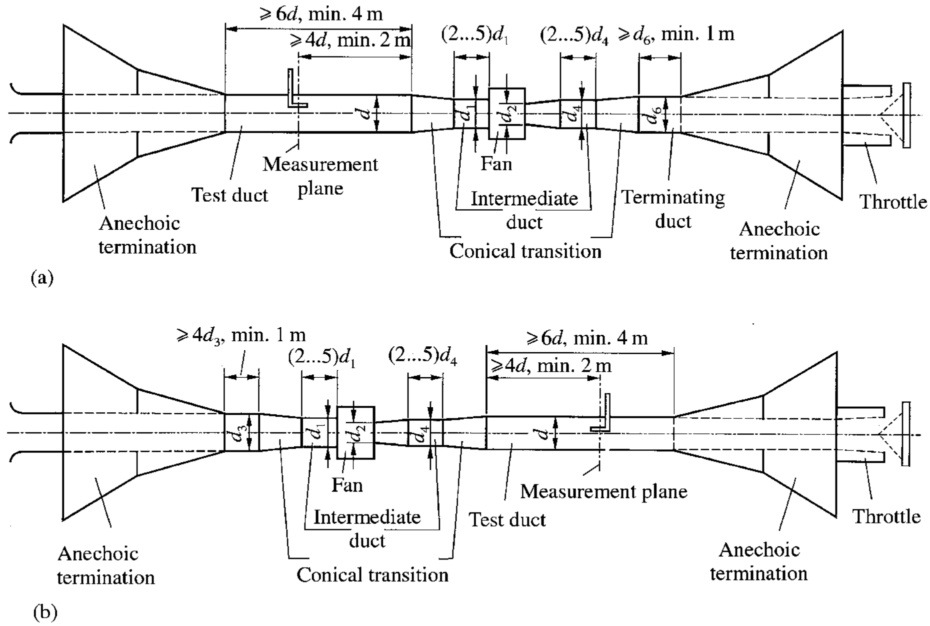

Figure 5. Test arrangement for the in-duct method ISO 5136:1990 [1]: (a) Sound measurement on the fan inlet side (b) Sound measurement on the fan outlet side.

sound propagation in the duct, but it is commonly related only to the fundamental mode, i.e., the plane wave mode, where the sound wavelength is large compared with the duct's lateral dimensions and the acoustic loading effects are most important. Therefore, the duct has to be specified for a measurement method. In ISO 5136 [1], the test duct is of circular cross-section and terminated nearly anechoically. The sound power determined under these special conditions is a representative value for actual applications, because the anechoic termination provides an acoustical impedance which is nearly independent of frequency and lies about midway between the negative and positive impedances encountered in practice. The duct terminations are specified in ISO 5136 [1] in terms of the maximum reflection coefficient as a function of frequency.

A schematic layout of the test arrangement for sound measurements on the fan inlet side and the fan outlet side according to ISO 5136 [1] is presented in Figure 5.

The in-duct method ISO 5136 [1] is specified for engineering grade accuracy measurements. The range of test duct diameters covered is from 0·15 to 2·0 m, the range of flow velocities 0 to 30 m/s, the maximum swirl angle allowed in the test duct is 15°, and the one-third octave band centre frequency range is from 50 to 10 000 Hz.

Each test duct can be used for a limited, specified range of fan sizes by employing conical duct transitions which have to meet certain requirements. Intermediate ducts have to be mounted at the fan inlet and outlet to ensure undisturbed flow conditions. If the non-measured side of the fan is normally ducted in the practical application, a terminating duct with an anechoic termination has to be mounted on this side. If the non-measured side of the fan is normally unducted, no terminating duct is required.

To suppress the effect of the turbulent pressures on the microphone, the use of a cylindrical windscreen ("turbulence screen", "sampling tube", "slit-tube", c.f., section 2.3) is prescribed. The turbulence screen according to ISO 5136 [1] shall suppress the turbulent pressure fluctuations by at least 10 dB in the frequency range of interest, and its directivity characteristic has to be within specified limits. Two of the procedures described in section 2.4 are incorporated in ISO 5136 [1] to determine whether or not there is a sufficient signal-to-noise ratio of sound to turbulence.

The microphone with the windscreen is mounted at a specified radial position such that the measured sound pressure is acceptably well related to the sound power by the plane wave formula, even in the frequency range of higher order duct modes, c.f., section 2.2. The radial measurement position r is a function of the test duct diameter d, i.e., $r = 0{\cdot}8\, d/2$ for $0{\cdot}15\ \mathrm{m} \leqslant d < 0{\cdot}5\ \mathrm{m}$ and $r = 0{\cdot}65\, d/2$ for $0{\cdot}5\ \mathrm{m} \leqslant d \leqslant 2{\cdot}0\ \mathrm{m}$. A circumferential average has to be obtained by measuring at least three evenly spaced azimuthal positions or by a continuous circumferential traverse.

The sound power level in the test duct of cross-section S is determined by the circumferentially averaged sound pressure level $\overline{L_p}$ obtained by using the relation

$$L_w = \overline{L_p} + 10\,\mathrm{lg}(S/S_0) - 10\,\mathrm{lg}[\rho c/(\rho c)_0] + C_1 + C_2 + C_3 + C_4, \tag{5}$$

where C_1 is the freefield microphone response correction, C_2 is the frequency response correction of the turbulence screen, C_3 is the flow velocity correction which accounts for the change in the frequency response of the turbulence screen as a result of the superimposed flow, see section 2.3, and C_4 is the so-called modal frequency correction which accounts for the fact that the sound pressure measured by a microphone with turbulence screen at the specified radial position does not give exactly the correct sound power in the duct when applying the plane wave formula. C_4 is dependent on the directivity characteristic of the microphone with the turbulence screen because of the propagation angle of the higher order duct modes.

The head-on frequency response C_2 has to be calibrated under acoustic freefield conditions. Data for C_3 as a function of frequency and flow velocity, and for C_4 as a function of frequency and duct diameter are tabulated in ISO 5136 [1]. The C_3-data are based on theoretical investigations by Neise [11], and the data for the modal correction C_4 on theoretical and experimental studies by Bolleter and Chanaud [15], Bolleter and Crocker [16] and Bolleter *et al.* [17] which were extrapolated to higher frequencies and duct diameters. The C_4-data given in ISO 5136 [1] were computed by Bolleter [17] based on the following assumptions: (1) reflections at the duct end are disregarded due to the anechoic termination used; (2) the microphone in the duct is placed sufficiently far from the source so that non-propagational duct modes can be neglected; (3) the total sound power which is transmitted through the duct at any one frequency is distributed uniformly over all propagational modes (mode model of "Equal Modal Sound Power" EMSP); (4) the various duct modes excited by the source are mutually uncorrelated; (5) the effect of the mean flow on the sound field in the duct is neglected, i.e., $M = 0$; the only flow effect considered is that on the plane wave sensitivity of the slit tube (C_3-correction); (6) the directivity characteristic of the microphone fitted with the turbulence screen is described by the following empirical formula: $D = 1/(1 + K_0 kL\,\Omega^3)$, where k is the wave number, L the length of the slit tube, Ω the incidence angle of the sound waves relative to the slit tube and K_0 is the directivity coefficient which has to lie within the following limits: $K_0 = 0{\cdot}05$–$0{\cdot}2$ for $f < 3500$ Hz and $K_0 = 0{\cdot}05$–$0{\cdot}3$ for $f \geqslant 4000$ Hz.

4. PRACTICAL EXPERIENCE WITH ISO 5136:1990 [1]

4.1. EUROPEAN ROUND-ROBIN TEST

In 1984/1985 a European round-robin test was carried out to determine the level of uncertainty with which fan sound power levels can be determined following the procedure

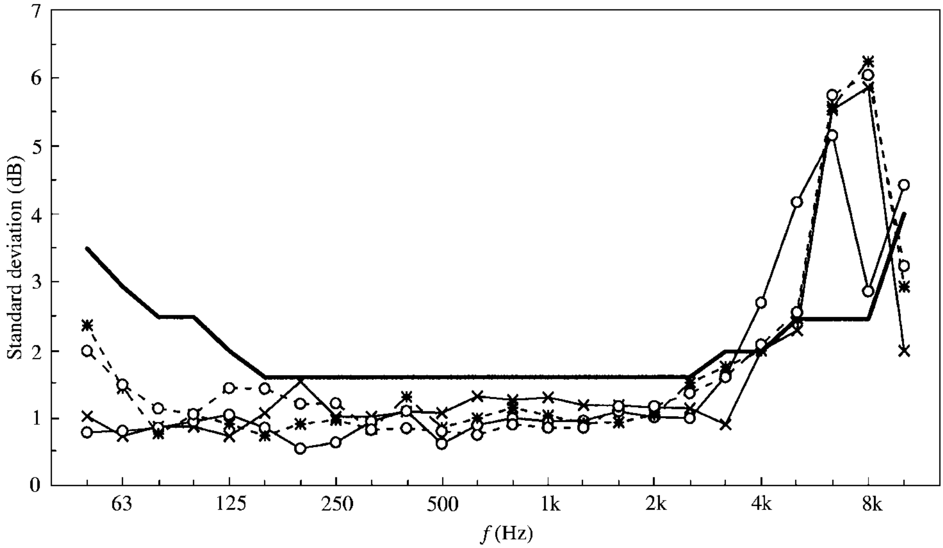

Figure 6. Overall standard deviations for centrifugal fan sound power levels found in a European round-robin test (after Bolton [18]): —×—, reference rig inlet duct; —○—, reference rig outlet duct; - - × - -, laboratory rigs inlet duct; - - ○ - -, laboratory rigs outlet duct; ———, limit ISO 5136.

described in ISO/DIS 5136.2 [4]; see the compound report by Bolton [18]. Six laboratories took part in the test. Experiments were done with an axial fan and a centrifugal fan. Each laboratory performed the measurements first with a reference test rig, which was shipped together with the fans, and secondly with a rig of its own design. The standard deviations of the sound power levels determined for the centrifugal fan for all participating laboratories were found to be within the limits specified in ISO/DIS 5136.2 [4], see Figure 6,† and similar values were found for the inlet-duct deviations on the axial fan. For the outlet side of the axial fan, the uncertainties were greater than the specified values as a result of the amount of swirl in the discharge duct which was larger than allowed in the standard. The results of the round-robin test were used to establish the standard deviation data given in ISO 5136 [1].

Swirl flow often occurs in the outlet ducts of axial fans, in particular when they have no guide vanes, and in these cases increased flow noise levels may arise at the microphone with turbulence screen. It was shown by Farzami and Guedel [19] that this problem can be overcome by placing a flow straightener between the fan outlet and the measurement plane for the acoustic measurements.

4.2. EXPERIMENTAL COMPARISON OF FAN NOISE MEASUREMENT STANDARDS

Holste and Neise [20] performed an experimental comparison of the following standardized sound power measurement procedures as applied to fans: the reverberation-room method ISO 3741 [21] and ISO 3742 [22], the freefield method over

†The larger standard deviations at frequencies beginning at 4000 Hz were found to be caused by the fact that, depending on the frequency analysers used by the various laboratories, the high-frequency portions of the spectra were buried in the electronic noise floor of the analyzing equipment.

a reflecting plane ISO 3744 [23], and the in-duct method ISO 5136 [1]. In spite of the fact that the in-duct sound power and the freefield sound power are principally different quantities, there is considerable practical interest in comparing the results of the in-duct method with those of the freefield method and the reverberation room methods and *vice versa*.

Six fans with impeller diameters between 450 and 510 mm were used for the experiments. The sound power measurements were done at the respective optimum fan operation conditions. The experimental data obtained by Holste and Neise [20] showed very good agreement between the results of the freefield method and reverberation-room method. Both methods seem equally well suited for determination of the sound power radiated from an unducted fan inlet or outlet, however; the reverberation-room method is applicable only to smaller fans because the size of the test object is limited to 1% of the room volume.

Differences were found by Holste and Neise [20] between the results of the freefield and the reverberation-room method on the one hand, and the results of the in-duct method on the other. In Figure 7, the level differences between freefield sound power spectra and in-duct sound power spectra are plotted versus the non-dimensional frequency kR_{eq} where k is the wave number and $R_{eq} = (A/\pi)^{1/2}$ is the equivalent radius of the fan inlet or outlet area A. In the frequency region where only plane sound waves can propagate in the test ducts, the in-duct method yields higher levels than the freefield method, which is due to the reflection of sound waves at the fan inlet or outlet, when the duct is removed. The measured differences between in-duct and freefield sound power levels can be described approximately by the reflection characteristics of a flanged duct end. Another reason for the differences between in-duct and free-space levels is the change in acoustic loading with and without a duct connected.

In the frequency range of higher order mode sound propagation, the in-duct sound power levels are lower than the freefield levels. The level difference is frequency dependent, with average values of about 3 dB on the inlet side and about 5 dB on the outlet side. Similar results were reported by Bolton [18, 26]. In this frequency regime, where the wavelength is small compared to the cross dimensions of the duct, effects of sound reflection and acoustic loading are unlikely to play a significant role, and one would expect all test methods to deliver the same result. Since the freefield tests and the reverberation room tests yielded consistent results, it appeared that the reason for this discrepancy lies in the in-duct method. As a possible cause, Holste and Neise [20] cited the two frequency correction terms C_3 and C_4 in ISO 5136 [1]. The data for the flow velocity correction C_3 are the result of theoretical investigations by Neise [12] and were not verified experimentally at that time. However, later experiments by Neise [27] showed that there is good agreement between measured and calculated C_3-data.

As mentioned before, the modal correction C_4 accounts for the directivity of the microphone with the turbulence screen with respect to the propagation angle of the higher order duct modes. The assumptions made for calculating the C_4-data given in ISO 5136 [1] are described in section 3.

Michalke [28] and Davy [29] showed that higher values for the modal correction C_4 are in fact obtained when the theoretical slit-tube model described by Neise [11] was used to calculate the microphone probe directivity, rather than the empirical relation used by Bolleter [17], see section 3. Theoretical studies further revealed that the modal correction is a function of the mean flow velocity [28], as well as of the flow direction relative to that of sound propagation. For that reason, an experimental and theoretical study was started at DLR and The Technical University in Berlin to obtain more accurate data for the modal correction to be implemented in the standard.

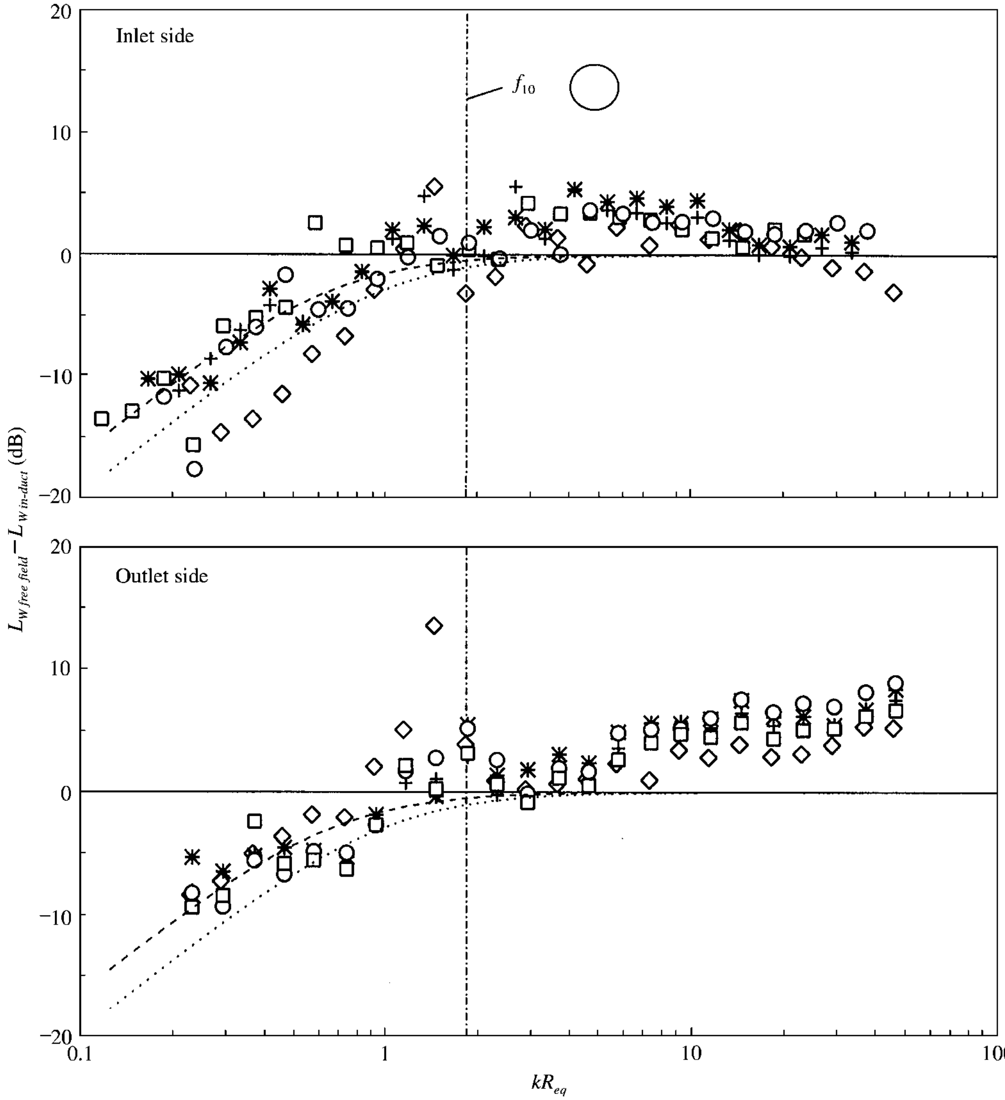

Figure 7. Comparison of fan sound power levels determined by using the in-duct method ISO 5136:1990 [1] and the freefield method ISO 3744:1981 [23] (after Holste and Neise [20]): - - - -, duct end in wall (after Mechel *et al.* [24]; · · · · · ·, unflanged duct end (after Levine and Schwinger [25]). ◇ Axial fan; + Centr. fan b.c. blades; ✱ Centr. fan b.c. blades; ○ Centr. fan radial blades; □ Centr. fan f.c. blades.

5. DETERMINATION OF MODAL CORRECTIONS

5.1. EXPERIMENTAL DETERMINATION OF MODAL CORRECTIONS

To determine modal correction data for a specified measurement path in a circular duct, one needs to know, or to be able to measure, the actual ("true") sound power transmitted in the duct. There is no experimental technique readily available for this task. It was mentioned before that the modern sound intensity technique, where the acoustic particle velocity is determined via the pressure difference at two slightly displaced points, cannot be applied in the higher order mode frequency regime in the presence of flow because there is no unique relationship between pressure gradient and axial acoustic particle velocity when the directions of flow and sound propagation are different [8, 30]. Other reliable methods for measuring the acoustic particle velocity in a turbulent flow environment are not known. Because of this difficulty, many investigators have tried to find other ways to determine the sound power in a duct. A thorough review of these papers was given by Arnold [31]. Here only a few are outlined.

Bolleter and Crocker [16] devised a correlation method in which the sound power was determined via azimuthal averages of the cross-spectra of sound pressure measured at various axial, radial and azimuthal positions. The method requires certain assumptions to be made about the degree of correlation between the acoustic duct modes. In this way they were able to resolve the first nine higher order acoustic duct modes.

Michalke [32, 33] proposed a method for sound power determination which involves the cross-spectra W_{12} of the sound pressure at various locations in the circular duct. If the microphones are spaced apart sufficiently, the turbulent pressure fluctuations at the two measurement positions are uncorrelated and do not influence the cross-spectra. Thus, a flow noise reduction device is not needed. The method is valid for tonal as well as broadband noise. The sound field is decomposed into duct modes and the sound power is determined for each individual mode. Summation of all modal terms yields the total transmitted sound power. Michalke [33] suggested locating the microphones at the same radial position ($r_1 = r_2$), the same or opposite circumferential position ($\Delta\varphi = 0$ or $180°$), and at different axial positions. Further, Michalke suggested taking the area-average of the cross-spectrum W_{12}. In this way, all intermodal terms which are affected by the degree of correlation between two modes *mn* and *uv* disappear and the number of unknown terms is reduced. This also means that no assumptions concerning the correlation between different modes have to be made like in other modal decomposition techniques, e.g., the one used by Bolleter and Crocker [16], who still kept some intermodal terms. Michalke's method was described in more detail by Arnold [31] who showed, theoretically and experimentally, that Michalke's method yields reliable data only for the non-dimensional frequency range up to $kR = 5{\cdot}3$. Arnold [31] refined Michalke's method in that he measured the area-averaged cross-spectra for a number of N_φ equidistant angular microphone positions spaced $\Delta\varphi_k = 360°/N_\varphi$., rather than placing the two microphones at either 0 or $180°$ angular displacement. With this modification, Arnold was able to apply a more stable algorithm and to extend the useful frequency range of the modal analysis technique to about $kR = 30$ and to resolve the modal sound power of up to 100 individual acoustic modes. Numerical simulations showed that the total sound power in one-third octave bands can be calculated with a maximum error of ± 1 dB. For more details, see the papers by Arnold [31, 34]. Some of his experimental data will be shown in section 6.

Arnold's [31] experimental results also enabled him to decide that the mode model of "equal modal energy density" (EME) best describes the modal distribution of the sound energy generated by ventilating fans, which is in accordance with the findings of Frommhold and Mechel [35] (see also Neise *et al.* [36]).

None of the experimental methods studied in the past can be used to determine the "true" sound power in the entire frequency range of practical interest, i.e., the standardized in-duct method covers a non-dimensional frequency range up to $kR = 183$ (10 kHz in a 2 m diameter pipe). Hence, one needs to find other ways to determine modal correction data, and in the following a theoretical approach is described.

5.2. THEORETICAL DETERMINATION OF MODAL CORRECTIONS

The calculation of C_4 modal correction data performed by Bolleter [17] was outlined in section 3. Arnold *et al.* [37] re-calculated the modal correction data based on Michalke's

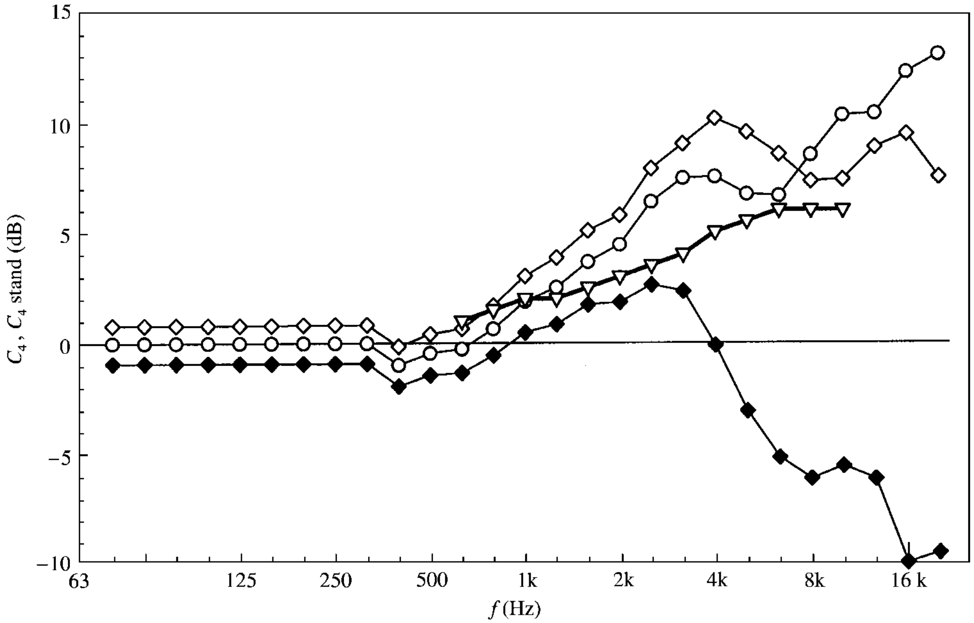

Figure 8. Modal correction C_4 calculated for a standard turbulence screen (slit tube for 13 mm microphone; slit 1 mm wide and 400 mm long; acoustic flow resistance of slit covering ρc; duct radius $R = 250$ mm; radial position $r/R = 0{\cdot}65$; mode model of equal modal sound power EMSP; modes correlated; after Arnold *et al.* [38]): —◇—, C_4, $M = 0{\cdot}1$; —○—, C_4, $M = 0$; —◆—, C_4, $M = -0{\cdot}1$; —▽—, $C_{4\,stand}$, ISO 5136 [1]; $M > 0$, outlet side; $M < 0$, inlet side.

[32, 33] theory for a microphone equipped with a "standard" turbulence screen (slit tube) placed at the radial position specified in ISO 5136 [1] under the following assumptions.

(1) Reflections at the duct end are disregarded due to the anechoic termination used.
(2) The microphone is located in the acoustic far field so that non-propagational duct modes can be neglected.
(3) Different mode models are compared, e.g., equal modal sound power, EMSP; equal modal energy density, EME.
(4) The influence of the degree of correlation between higher order duct modes is investigated.
(5) The effect of superimposed mean flow on sound propagation is taken into account, for upstream as well as downstream propagation, and the effect of the mean flow on the slit-tube response is also considered: at higher frequencies higher order modes propagate in the duct. Their angle of incidence to the turbulence screen depends on the modal order (m, n), the Helmholtz number kR (k the wave number, R the duct radius) and on the flow Mach number M.
(6) The directivity characteristic of the turbulence screen is described by the theoretical model of the slit tube put forward by Neise [12].

In Figure 8, results of this calculation for a standard length slit-tube and the mode model of equal mode sound power (ESMP) are compared with the data published in ISO 5136 [1]. Clearly, the modal correction is a function of the superimposed flow velocity, in both magnitude and direction. On the outlet side ($M > 0$), the new modal correction data are somewhat larger than the ISO data. On the inlet side ($M < 0$) the modal correction can assume negative values, depending on frequency and flow velocity.

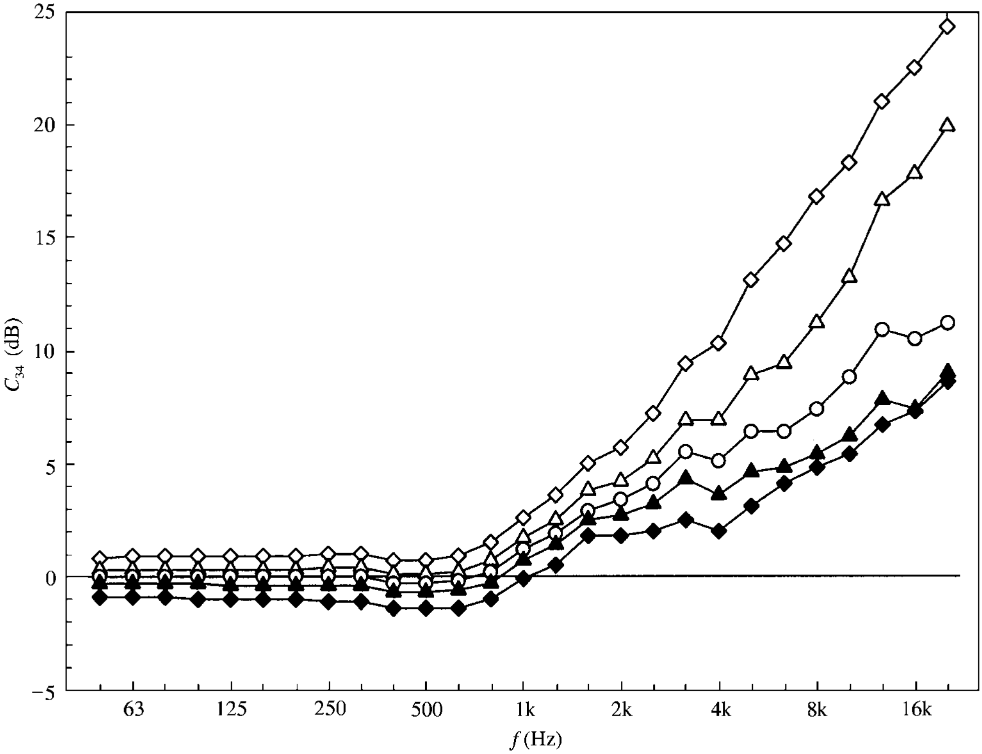

Figure 9. Combined flow velocity and modal correction C_{34} calculated for a standard turbulence screen and different flow velocities (slit tube for 13 mm microphone; slit 1 mm wide and 400 mm long; acoustic flow resistance of slit covering ρc; duct radius $R = 250$ mm; radial position $r/R = 0{\cdot}65$; mode model of equal modal energy density EME; modes correlated; after Arnold [31, 34]: —◇—, $M = 0{\cdot}1$; —△—, $M = 0{\cdot}04$, —○—, $M = 0$; —▲—, $M = -0{\cdot}04$; —◆—, $M = -0{\cdot}1$; $M > 0$, outlet side; $M < 0$, inlet side.

In Bolleter's [17] approach which was adapted for ISO 5136 [1], the influence of superimposed flow on the propagation of sound was disregarded, and thus the modal correction C_4 was considered independent of flow velocity. Only the effect of the superimposed flow on the acoustic sensitivity of the turbulence was accounted for in terms of the flow velocity correction C_3. The above results show that this concept, i.e., using two corrections to account for the effects of flow and modal sound propagation separately, has to be abandoned. Arnold *et al.* [37] proposed that instead of using the two frequency corrections C_3 and C_4, which are both influenced by the response of the turbulence screen and are both functions of the mean flow velocity, a combined flow velocity and modal correction C_{34} should be introduced which accounts for the effect of mean flow on the acoustic response function of the turbulence screen as well as on the propagation of higher order sound waves in the test duct

$$C_{34} = C_3 + C_4. \tag{6}$$

In Figure 9, computed results are shown for the combined correction for a standard length slit tube in a pipe of 0·5 m diameter. The mode model EME is used here. The frequency dependence of the C_{34}-data is much smoother than that of the former C_3- and C_4-data; c.f. Figures 8 and 9. C_{34} increases with the flow Mach number M. The correction for the outlet duct is higher than for the inlet duct.

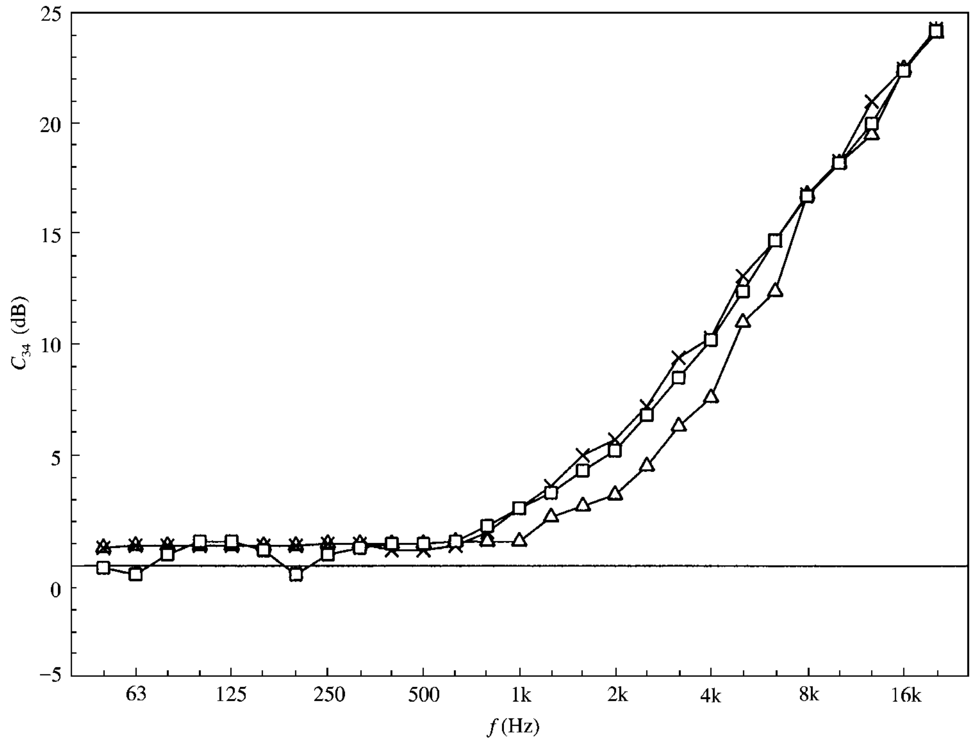

Figure 10. Combined flow velocity and modal correction C_{34} calculated for a standard turbulence screen and different duct sizes (slit tube for 13 mm microphone; slit 1 mm wide and 400 mm long; acoustic flow resistance of slit covering ρc; flow Mach number $M = 0{\cdot}1$ (outlet duct); mode model of equal modal energy density EME; modes correlated; after Arnold [31, 34]: $M = 0{\cdot}1$; —△—, $R = 87{\cdot}5$ mm, $r/R = 0{\cdot}8$; —×—, $R = 250$ mm, $r/R = 0{\cdot}65$; —□—, 3000 mm, $r/R = 0{\cdot}65$.

C_{34}-corrections for different duct sizes are shown in Figure 10 for the largest flow Mach number considered, $M = 0{\cdot}1$. The radial microphone position r depends on the duct radius R. At high frequencies C_{34} hardly varies with R. Hence, the maximum magnitude of the combined flow velocity and modal correction C_{34} is determined by the flow Mach number M rather than the duct size given by the duct radius R.

6. COMAPRISON OF SOUND POWER SPECTRA OBTAINED BY MODAL ANALYSIS TECHNIQUE AND IN-DUCT METHOD

Arnold [31, 34] applied his improved modal analysis technique involving the area-averaged cross-spectra (see section 5.1) to a centrifugal and an axial fan and compared these "true" sound power spectra with the results of the in-duct method using the previous C_3- and C_4-corrections as well as the new C_{34}-corrections. As an example, the results for the axial fan are depicted in Figure 11. The fan has 12 blades and eight outlet guide vanes and runs at a speed of $n = 2960$/min. The mean flow velocity in the test ducts is $U = 16{\cdot}8$ m/s and the corresponding Mach number $M = 0{\cdot}048$.

The diagrams on the left-hand side show the sound power level L_W, and on the right-hand side the level differences between the in-duct method and the cross-correlation method are plotted in an enlarged scale. In the outlet duct, application of the C_3- and C_4-corrections gives very low levels at high frequencies, whereas in the inlet duct the levels are too high.

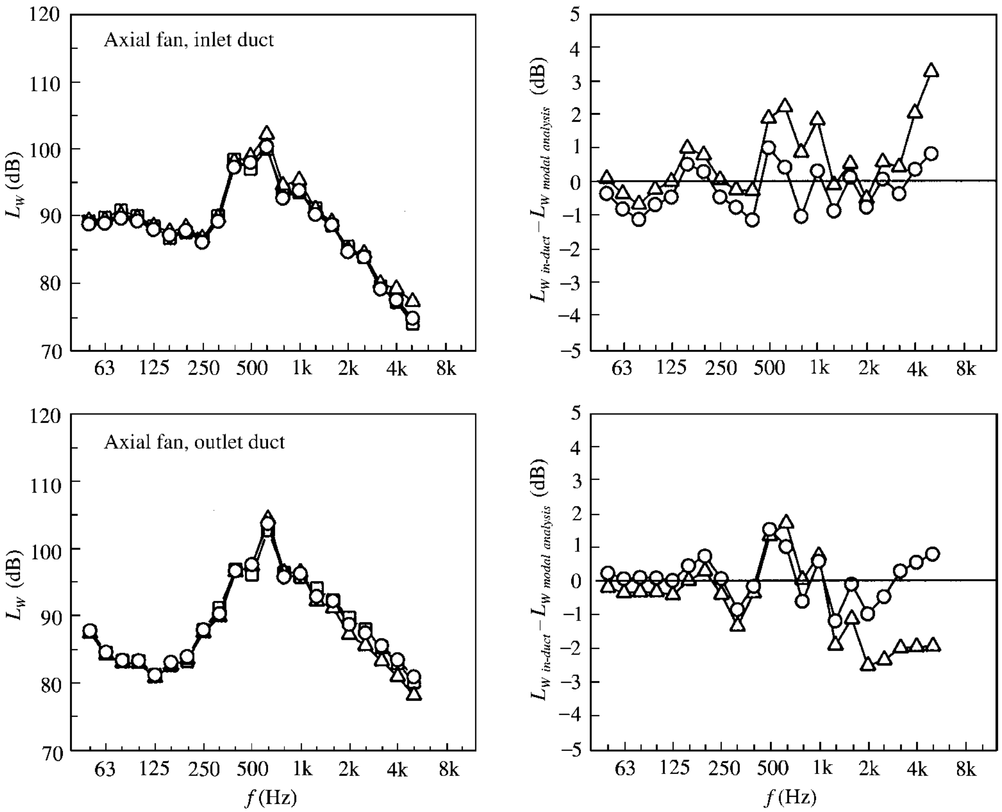

Figure 11. Comparison of sound power spectra of an axial-flow fan obtained by applying the modal analysis technique and the in-duct method with corrections C_3 and C_4 (ISO 5136:1990) and C_{34} (ISO/DIS 5136:1999); after Arnold [31, 34]: Left diagrams: —□—, modal analysis; —△—, in-duct, old C_3-, C_4-corrections; —○—, in-duct, new C_{34}-corrections. Right diagrams: Level difference to modal analysis spectra: —△—, in-duct, old C_3-, C_4-corrections; —○—, in-duct, new C_{34}-corrections.

Application of the new C_{34}-corrections yields very good agreement between the results of the in-duct method and the modal analysis technique. In this case, the C_{34}-corrections were calculated for a standard slit tube with ρc flow resistance of the slit covering, using the mode model EME. Higher order modes are assumed to be uncorrelated. This has been shown to be the most appropriate estimate for the calculation of C_{34} when the exact degree of correlation between the various duct modes is not known

In Figure 12, freefield method and in-duct method with old and new corrections are compared. The measurements were made by Holste and Neise [20]. For the freefield tests, the measured (non-ducted) side of the fan was connected to a cone in order to avoid sudden changes of the acoustical impedance and to make the two methods comparable. At high frequencies the in-duct sound power spectra obtained by using the new C_{34}-corrections agree much better with the freefield spectra than when using the previous C_3- and C_4-corrections.

Based on the insights gained by the various studies described above, the following recommendations were given to ISO/TC 43/SC 1 for revision of ISO 5136 [1]:

- Replace the existing tables for the flow velocity correction C_3 and modal correction C_4 by tables for the combined frequency correction C_{34}.

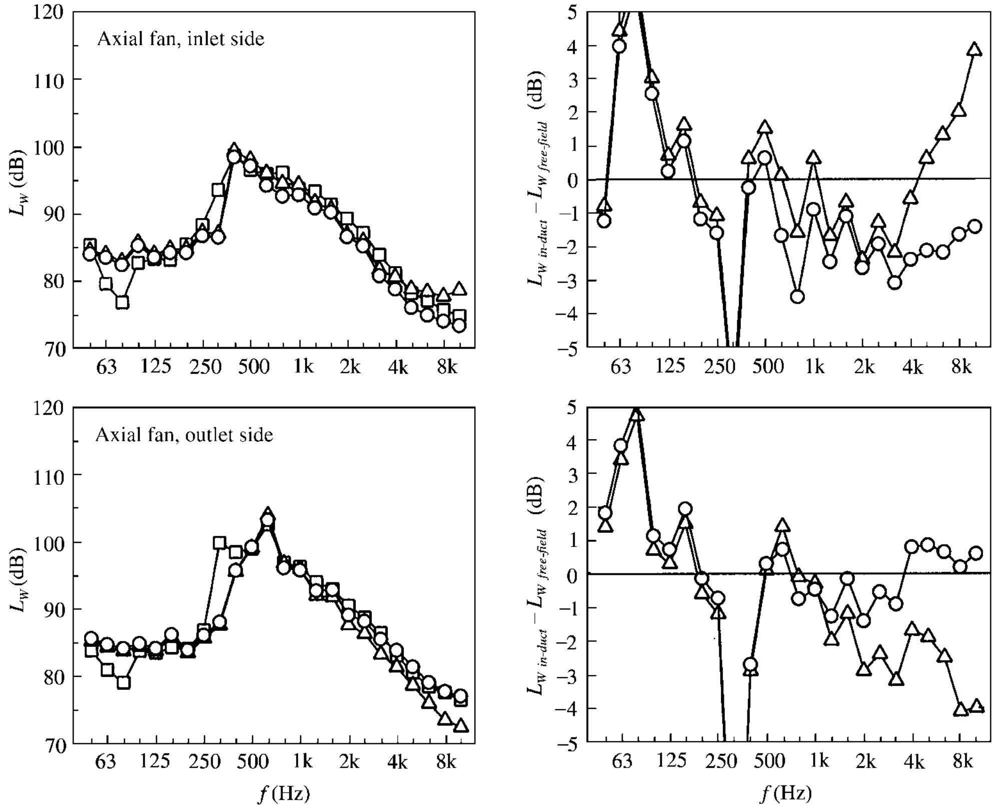

Figure 12. Comparison of sound power spectra of an axial-flow fan obtained by applying freefield method ISO 3744 [23] and in-duct method with corrections C_3 and C_4 (ISO 5136:1990) and C_{34} (ISO/DIS 5136:1999); after Arnold [31, 34]: Left diagrams: —□—, freefield method with cone; —△—, in-duct, old C_3-, C_4-corrections; —○—, in-duct, new C_{34}-corrections. Right diagrams: Level difference to free-field spectra: —△—, in-duct, old C_3-, C_4-corrections; —○—, in-duct, new C_{34}-corrections.

- Maintain the present measurement position and procedure in the test duct, so that existing test data can be used to re-calculate sound power spectra using the new combined flow velocity and modal correction C_{34}.
- Allow for a flow straightener in the outlet duct between the source under test (fan) and the microphone to eliminate the negative effect of swirl flow on the microphone probe.

7. REVISED IN-DUCT METHOD ISO/DIS 5136:1999 [2]

A schematic layout of the test set-ups for the revised in-duct method is depicted in Figure 13. The test duct diameter range is from 0·15 to 2·0 m. With the transition pieces allowed, the range of fan inlet equivalent diameters which can be tested is from 0·104 to 2·0 m, the range of fan outlet equivalent diameters is from 0·104 to 2·390 m.

A test method for small test ducts in the range 0·070 m $\leqslant d <$ 0·15 m is described in an appendix of ISO/DIS 5136:1999 [2]; this allows testing of fans with inlet and outlet equivalent diameters down to 0·0485 m. In another appendix a method for large test ducts in the range 2 m $< d \leqslant$ 7·1 m is described which allows testing of fans with inlet equivalent diameters of up to 7·1 m and outlet equivalent diameters of up to 8·5 m.

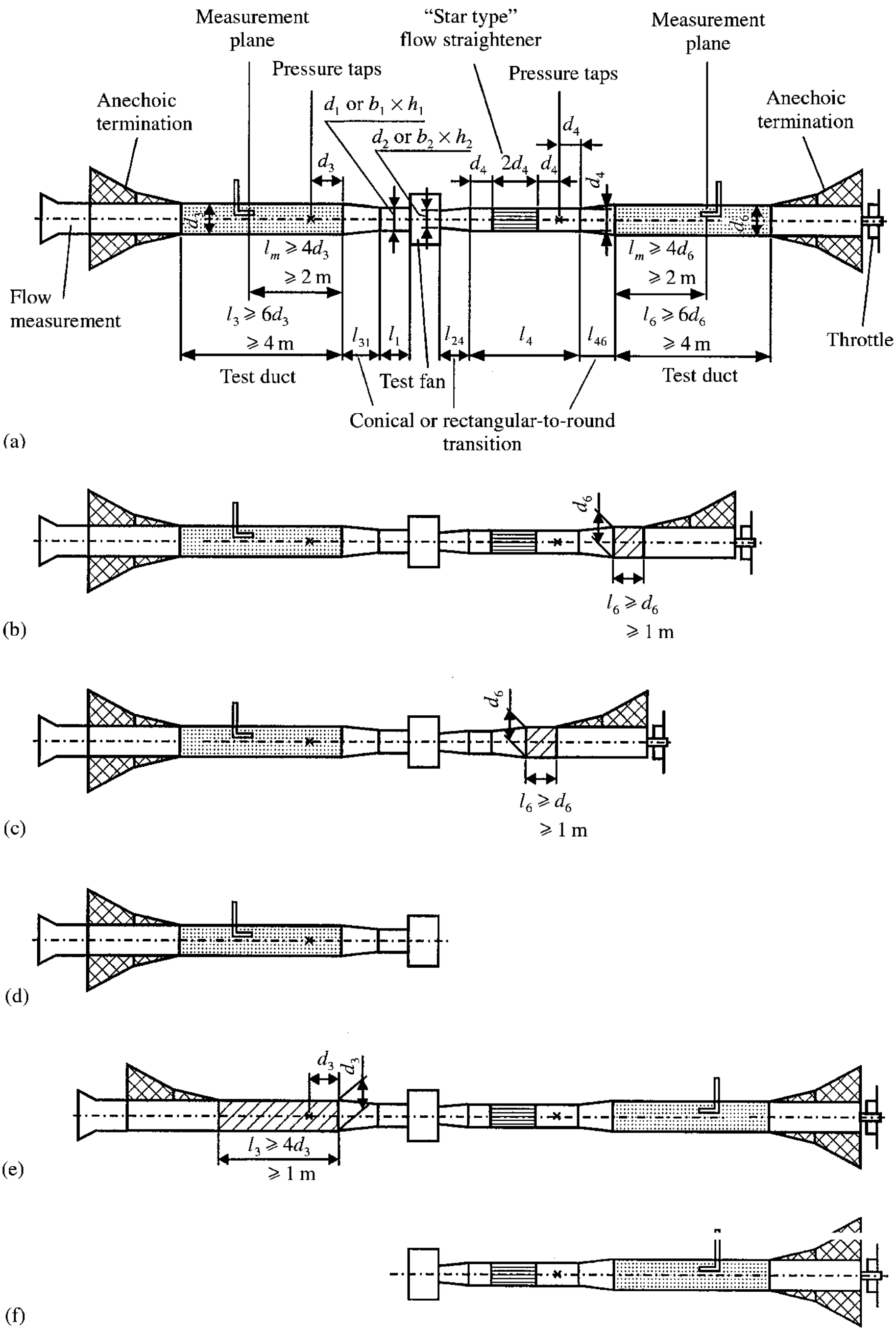

Figure 13. Test set-ups for the in-duct method ISO/DIS 5136:1999 [2]: (a) Simultaneous measurement of inlet and outlet in-duct noise; installation categopry D; simultaneous mesurement of aerodynamic performance possible. (b) Measurement of inlet in-duct noise only; installation category D; simultatenous measurement of aerodynamic performance possible. (c) Measurement of inlet in-duct noise only; installation category D. (d) Measurement of inlet in-duct noise; installation category C. (e) Measurement of outlet in-duct noise only; installation category D; simultaneous measurement of aerodynamic performance possible. (f) Measurement of outlet in-duct noise only; installation category B.

Table 1

Maximum pressure reflection coefficients of test ducts and terminating ducts according to ISO/DIS 5136 [2]

f (Hz)	Test duct	Terminating duct
50	0·40	0·80
63	0·35	0·70
80	0·30	0·60
100	0·25	0·50
125	0·15	0·30
160	0·15	0·30
> 160	0·15	0·20

To specify a standardized acoustical load impedance for ducted installations, all ducts connected to the test fan have to be terminated anechoically. A duct in which the sound pressure is to be measured for determination of the in-duct sound power is called "test duct". Ducts which are used only to provide the standardized acoustic loading, i.e., in which no sound measurements are to be made, are called "terminating ducts". The maximum permissible pressure reflection coefficients for test ducts and terminating ducts as specified in ISO/DIS 5136:1999 [2] are given in Table 1. The specifications for test duct terminations are as in the previous standard while those for terminations used for terminating ducts are relaxed.

The range of permissible flow velocities in the test duct is now from 0 to 40 m/s. Three different types of windshield are allowed: foam ball (up to 15 m/s), nose cone (up to 20 m/s), and turbulence screen (up to 40 m/s). The last named is the preferred probe, and the uncertainties specified in the standard apply only when the turbulence screen is used. No uncertainty information is given for the cases when foam ball or nose cone are used.

Compared with the previous version of the method, the ducting arrangements on the inlet and outlet side were modified to enable simultaneous aerodynamic performance testing according to ISO 5801 [38]. A "star-type" flow straightener is mounted upstream of the outlet test duct. The straightener is necessary for aerodynamic fan performance testing, and it eliminates the negative effect of swirl flow on the microphone windshield. Two effects are to be considered when performing the sound measurement in the outlet duct: (1) the swirling flow entering the flow straightener may generate excess noise at the measurement station which may or may not be of higher level than the sound pressure level produced by the air moving device under test. (2) Without a flow straightener in place, the swirling flow around the measurement microphone may generate excess flow noise which may or may not be of higher level than the sound pressure level produced by the air moving device under test. Both effects tend to increase the measured pressure level over the sound pressure level produced by the air moving device under test. Which of the two effects is stronger depends on the amount of swirl, the flow velocity in the duct, the microphone shield used and the acoustic strength of the source under test. Therefore, the outlet duct noise shall be measured with and without the flow straightener in position. Of the sound pressure level readings taken, the lowest shall be considered to represent the true sound pressure in the test duct, for each one-third octave band of interest.

The in-duct sound power level is now governed by the relation

$$L_w = \overline{L_p} + 10\lg(S/S_0) - 10\lg[\rho c/(\rho c)_0] + C_1 + C_2 + C_{34}, \tag{7}$$

where C_{34} is the new combined flow velocity and modal correction. Data for C_{34} are given in ISO/DIS 5136:1999 [2] as a function of frequency for the test duct diameter and flow velocity range covered; for information, C_{34}-data are given also for flow velocities up to 60 m/s and for the extended duct diameter ranges described above. The radial measurement position of the microphone with turbulence screen is the same as in ISO 5136 [1].

8. CONCLUSIONS

With the revision of the in-duct method ISO/DIS 5136:1999 [2], technical problems that became evident in previous experiments have been removed. The flow velocity correction C_3 and the modal correction C_4 used in ISO 5136 [1] are replaced by a new combined flow velocity and modal correction C_{34}. Recent tests with the new correction proved that very good agreement is now obtained between in-duct sound power levels and freefield sound power levels in the frequency region of higher order sound propagation. Comparison with a new in-duct modal analysis technique also confirmed the validity of the C_{34}-corrections.

A star-type flow straightener is used upstream of the discharge test duct to remove swirling flow. Also the test set-ups were modified to enable simultaneous acoustic and aerodynamic performance testing. The in-duct test procedure requires that the sound pressure be measured in an anechoically terminated test duct. Consequently, the method is largely insensitive to environmental conditions and background noise problems. Only three microphone positions in the duct, or one circumferential microphone traverse, are required. The method has been tested specifically for fans in a European round-robin test [18] and was found to be practical and to give reproducible results. The in-duct method is of "engineering grade measurement accuracy" when the turbulence screen (slit tube) is used as a microphone wind shield.

Two other basic acoustic test methods with the same class of accuracy, which are based on sound pressure measurement, are available for testing fan noise: the reverberation room methods ISO 3743-1/2 [39, 40] and the freefield method over a reflecting plane ISO 3744 [41]. Note, however, that these methods are for the determination of sound power radiated into free space while the in-duct method is for the determination of sound power radiated into a duct. The two reverberation room methods are applicable to small-to-medium size fans only because of the size restriction in the standards.

The freefield method ISO 3744 [41] covers the one-third octave band frequency range from 50 to 10 000 Hz and is applicable to all fan sizes and to all types of noise. For engineering grade accuracy measurements, the environmental correction has to be $K_2 \leqslant 2$ dB over the entire frequency range of interest. In a European round-robin test, it was found not possible to satisfy this criterion for all frequencies (see references [42, 43]). In the latter study, values of as much as $K_2 = 4$ dB were observed at certain frequencies, despite the fact that the environmental test conditions were favourable, considering practical situations. Another problem was the generation of background noise by the fan drive system and by the throttle needed to control the fan operating condition, both of which are difficult to separate from the fan noise to be measured.

A detailed comparison of freefield method, reverberation room method, and in-duct method as applied to fans was given by Neise [44]. In Table 2, the estimated values of the standard deviations of reproducibility of sound power levels of the in-duct method are compared to those of the reverberation-room methods and the freefield method. ISO 3743-1/2 [39, 40] are applicable only for octave bands between 125 and 8000 Hz, while ISO 3744 [41] and ISO 5136 [1, 2] are for one-third octave bands in the range 50–10 000 Hz. Up

TABLE 2

Estimated standard deviations of reproducibility of sound power levels according to various ISO-*standards*

f (Hz)	ISO 3743-1	ISO 3743-2	ISO 3744	ISO 5136
50			5·0	3·5
63	—	—	5·0	3·0
80			5·0	2·5
100			3·0	2·5
125	3·0	5·0	3·0	2·0
163			3·0	2·0
200			2·0	2·0
250	2·0	3·0	2·0	2·0
315			2·0	2·0
400			1·5	2·0
500	1·5	2·0	1·5	2·0
630			1·5	2·0
.	.	.	.	.
.	.	.	.	.
3150			1·5	2·0
4000	1·5	2·0	1·5	2·0
5000			1·5	2·5
6300			2·5	3·0
8000	2·5	3·0	2·5	3·5
10 000			2·5	4·0

to 315 Hz, the in-duct method yields the best accuracy and is only slightly worse than the freefield method up to 4000 Hz.

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