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SOUND INSULATION BY ABSORBER BARRIERS

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1. INTRODUCTION

There exist situations where a certain sound insulation needed can be only hardly realised by walls, either as a consequence of a high amount of fittings (such as in the voids of false ceilings or of computer floors) or because large masses are not permitted (such as in explosion security openings). Then the absorber barrier can be applied; it is a staple of boards or quilts of fibre absorber which fit elastically to the flanking walls (see Fig.1).

The sound insulation of such absorber barriers has been studied already experimentally [1-3] and theoretically [5-9]. The theories applied up to now either make use of heavily simplified absorber models with a rigid fibre structure, or the theories make use of parameters (structure factor, mechanical stiffness of the fibres alone, coupling coefficients) which are undetermined [6-8]. Numerical estimates from these theories and the fact that absorber barriers are subdivided in a number of layers make it plausible that for practical use the existence of elastic waves in the absorber materials can mostly be neglected.

2. AIM OF THE INVESTIGATIONS

The sound transmission through a plug of thickness d of absorber material which is homogeneous and isotropic with a non-vibrating fibre matrix can easily be derived theoretically, as the absorber material is completely described by its characteristic constants, i.e. the propagation constant Γ_a and the wave impedance Z_a. They are related to the specific flow resistance Ξ of the material via the non-dimensional parameter $E = \rho_0 f/\Xi$ according to the relations of Fig.2 [1o,11]. It was to be tested whether it is possible to bring numerical and experimental results to an agreement with each other by a generalisation of Ξ to Ξ_{eff} with

$$1/\Xi_{eff} = 1/\Xi + 1/j\omega\rho_a$$

which just describes a shunting of the specific flow resistance Ξ by the mass reactance $j\omega\rho_a$, with ρ_a being the absorber's bulk density.

3. CALCULATIONS

The transmission coefficient τ of the sound power (transmission loss $R = -10 \lg \tau$) of a plane sound wave incident under an angle θ (see Fig.1) is

$$\tau = \left| \frac{4z \cdot e^{-y}}{(1+z)^2 - (1-z)^2 \cdot e^{-2y}} \right|^2$$

with the notations $y = k_o d \cdot \Gamma_{an} \cos \theta_1$; $z = Z_{an} \cdot \cos\theta/\cos\theta_1$

$$\Gamma_{an} = \Gamma_a/k_o \quad ; \quad Z_{an} = Z_a/Z_o$$

where θ_1 is the (complex) refracted angle within the material for which the boundary conditions yield

$$\Gamma_{an} \cdot \cos\theta_1 = [1 + \Gamma_{an}{}^2 - \cos^2\theta]^{1/2}.$$

The transmission factor t of the sound pressure amplitude (with $\tau = |t|^2$) can be decomposed into $t = t_o \cdot t_1 \cdot e^{-y}$ with the transmission factors t_o of the front side and t_1 of the rear side of the barrier and with the interior attenuation factor e^{-y}. It is:

$$t_o = 2z/[z \cdot (1 + r_1 \cdot e^{-\frac{y}{2}}) + (1 - r_1 \cdot e^{-\frac{y}{2}})] \; ; \; t_1 =: 1 + r_1 = 2/(1 + z) \; .$$

4. MEASUREMENTS

The insertion loss R of absorber barriers was measured which were made out of different fibrous materials (glass fibres, mineral fibres) with bulk densities ρ_a from 13 to 7o kg/m³ and corresponding specific flow resistances Ξ from about 5 to 38 KNs/m⁴ for flow normal to the boards and from 2.4 to 18 KNs/m⁴ for flow parallel to the boards. The barrier thickness d ranged from o.o8 to 1.26 m. The material boards were arranged either perpendicular to the sound direction (the boards stacked in series behind each other) or parallel (the boards stapled upon each other). After prealable tests with the absorber barrier placed into the test opening of sound insulation test rooms for doors have shown good agreement with measurements in which the absorber barrier was placed into a test duct for silencers (length 11m; cross-section o.5 x 1.o m²), the major part of the test program was performed in the silencer test duct.

5. RESULTS

Without consideration of the finite mass reactance of the absorber plug (fibre matrix rigid), the sound transmission loss for a (2-dimensional) diffuse sound incidence becomes a two-dimensional problem of the non-dimensional variables $f \cdot d/c_o$ and $\Xi \cdot d/Z_o$ (c_o = sound velocity; $Z_o = \rho_o c_o$). Contour lines of constant R_{diff} are shown in Fig.3. A comparison with a similar contour map for normal incidence (θ = o) reveals only minor differences. The strongest influence evidently has the thickness d (diagonal shift in the contour map). At low frequencies and flow resistances a flat minimum of the frequency response curves of R (horizontal trajectory through the map) can be seen. However, the vibration of the absorber plug as a whole is expected to produce the strongest differences just in that region. The influence of the angle of incidence can be seen from Fig.4 with contour lines of R (θ) for a small value of the resistance parameter $S = \Xi \cdot d/Z_o = 1$ and from Fig. 5 for a medium flow resistance S = 1o. It can be seen that the angular dependence is relatively weak, with the exception of angles near to grazing incidence (θ = 9o degrees).
If the finite mass reactance of the absorber plug is introduced via Ξ_{eff} as mentioned above and if the empirical relation

$$\Xi_{[kNs/m^4]} = 0{,}022 \cdot \rho_a{}^{1,6}{}_{[kg/m^3]}$$

for mineral fibre materials between Ξ and ρ_a is used, then the contour map of R in Fig.6 is obtained for a thickness d = o.5 m and $\theta = \sigma$. Significant differences as compared to Fig.3 appear at low frequencies.

A direct comparison between measured and computed (including mass reactance) frequency response curves of R for oblique incidence (θ = 45 degrees) with an absorber barrier made out of glasfibre boards in series (ρ_a = 12.8 kg/m³; Ξ = 4.9 KNs/m^4) is shown in Fig. 7. (The decrease of the measured points in the upper right corner is caused by the flanking transmission of the test duct). It can be realised that the computational method which makes use of Ξ_{eff} gives satisfying agreement with experimental results. This also holds for absorber barriers made out of absorber boards or quilts laying in parallel to the sound propagation under condition that the bulk density is not too high (ρ_a < 4o kg/m³). Otherwise the high stiffness of the boards parallel to their surface would give rise to elastic waves which would deteriorate the sound insulation.

6. LITERATURE

[1] Westphal, W.
wksb (1964) 18-22

[2] Mechel, F.P.
wksb (1977), 25-29

[3] Mechel, F.P. ; Royar, J.
Fortschr. d. Akustik - DAGA'76,
VDI-Verl. (1976) 205-208

[4] Kraak, W.
Hf.Tech.Elak. 71(1962) 86-98

[5] Schneider, P.
BAM-Ber. 29, 1974

[6] Stöbe, H.D.
Hf.Tech.Elak. 80(1971) 97-104

[7] Stöbe, H.D.
Z.el.Inform.Energ.techn. 2(1972) 27-32

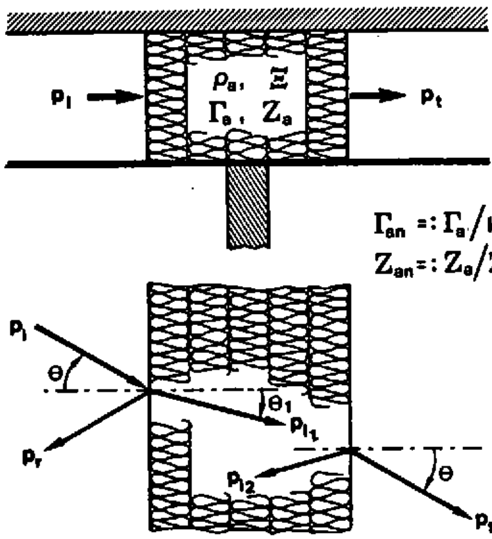
[8] Rosin, G.S.
Sov.Phys.Acoust. 19(1973) 60-64

[9] Fasold, W.; Kraak, W.; Schirmer, W. (Edit.)
Taschenbuch Akustik, VEB-Verl. Berlin (1984) 125ff

[10] Mechel, F.P.
Akustische Kennwerte von Faserabsorbern
IBP-Bericht BS 85/83 (1983)

[11] Mechel, F.P.
Fortschr. d. Akustik - DAGA'84, DPG-GmbH (1984) 31-52

ABSORBER-SCHOTT



Refraction:

$$\frac{\sin\Theta_1}{\sin\Theta} = \frac{jk_o}{\Gamma_a} \quad ; \quad \Gamma_{an}\cdot\cos\Theta_1 = \sqrt{1+\Gamma_{an}^2 - \cos^2\Theta}$$

Transmission:

$$\tau = \left| \frac{4z\cdot e^{-y}}{(1+z)^2 - (1-z)^2\cdot e^{-2y}} \right|$$

$$y =: k_o d\cdot\Gamma_{an}\cdot\cos\Theta_1 \; ; \; z = \frac{\Gamma_{an}\cdot Z_{an}}{\Gamma_{an}\cdot\cos\Theta_1}\cdot\cos\Theta$$

Fig. 1

ABSORBER - CHARACTERISTICS

$$E =: \rho_o f / \Xi$$

$E \leq E_x$:

$$\Gamma_{an} = \sqrt{-1 + j\frac{\gamma}{2\pi E}} \quad ; \quad Z_{an} = \frac{1}{j\gamma h}\Gamma_{an}$$

γ = 1.40 Adiabat.Expon.; h = Porosity

$E > E_x$:

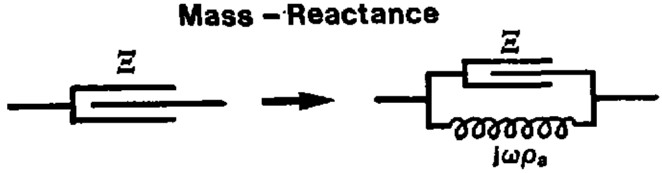
$$\Gamma_{an} = 0.208/E^{0.619} + j(1 + 0.109/E^{0.673})$$

$$Z_{an} = (1 + 0.0608/E^{0.717}) - j\,0.132/E^{0.660}$$

Γ'_{an}: $E_x = 0.04$; Γ''_{an}: $E_x = 0.008$

Z'_{an}: $E_x = 0.006$; Z''_{an}: $E_x = 0.02$

Mass - Reactance



$$\frac{1}{\Xi} = \frac{1}{\Xi} + \frac{1}{j\omega\rho_s}$$

Fig. 2

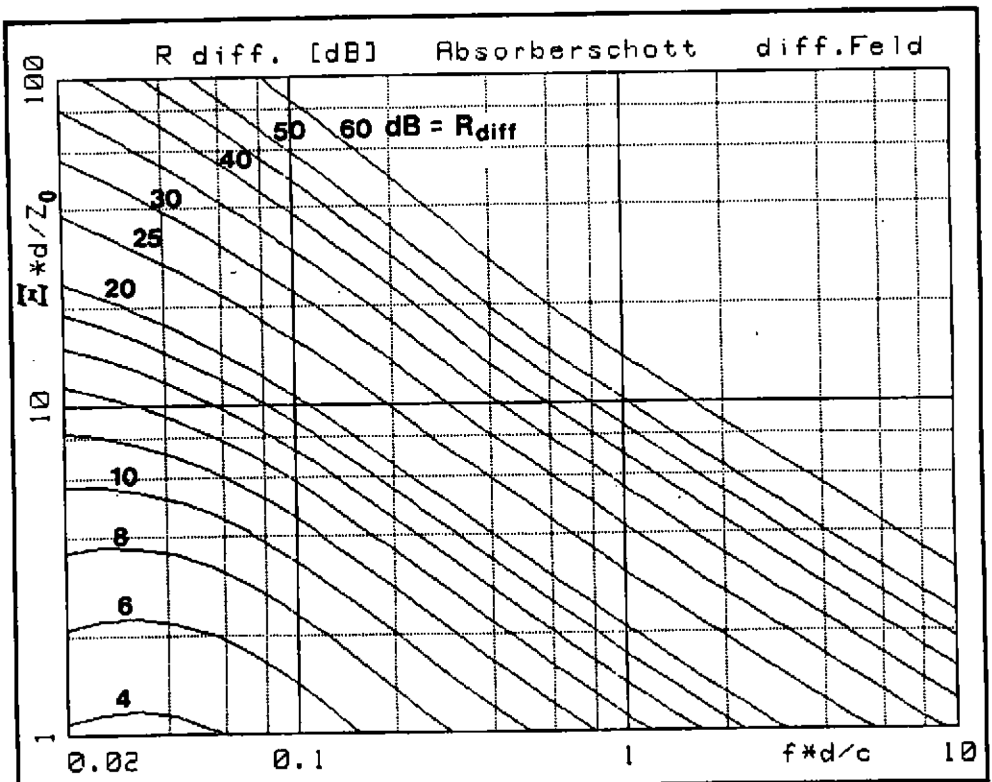

R diff. [dB] Absorberschott diff.Feld
60 dB = R_{diff}
50
40
30
25
20
10
8
6
4
Ξ *d/Z0
100
10
1
0.02
0.1
1
f*d/c
10

Fig. 3

SOUND INSULATION BY ABSORBER BARRIERS

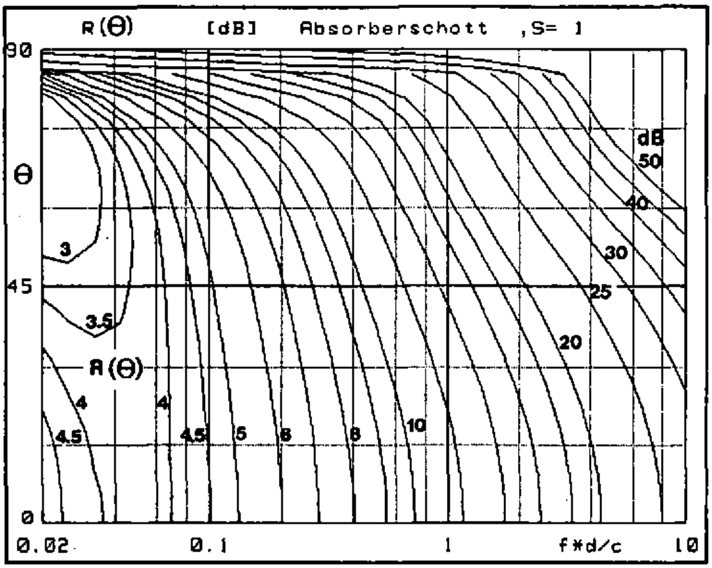


Fig. 4

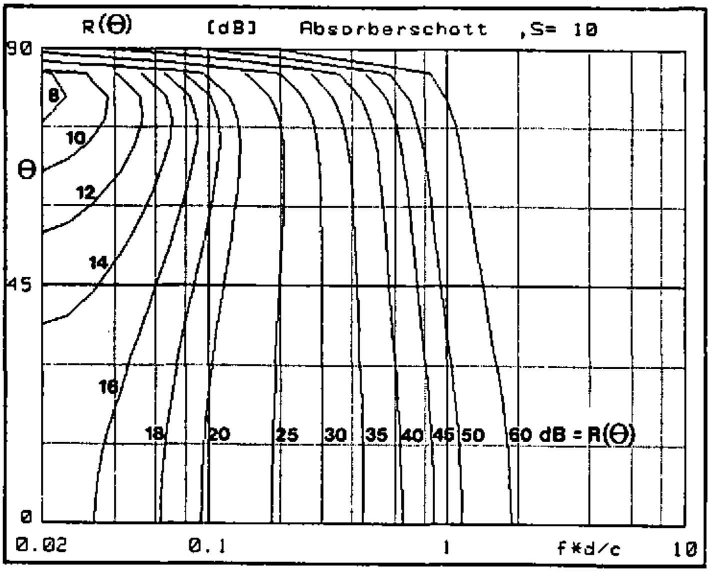


Fig. 5

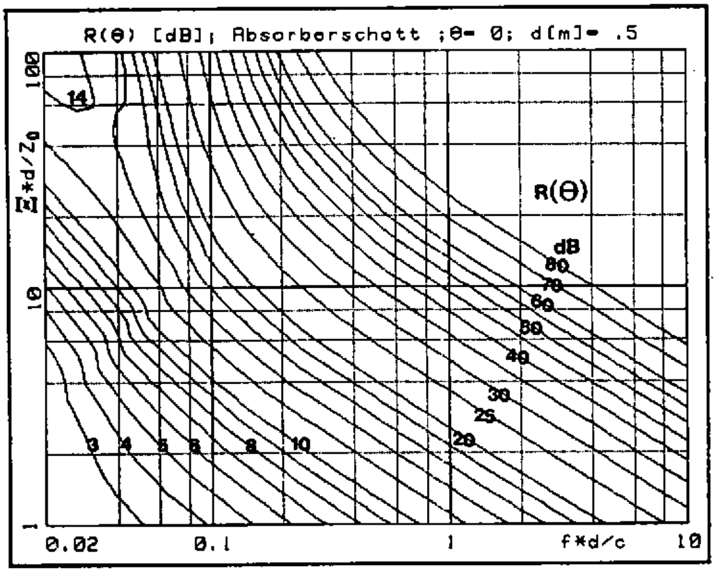

R(Θ) [dB]; Absorberschott ;Θ= 0; d[m]= .5
Ξ*d/Z0
R(Θ)
dB
80
70
60
50
40
30
25
20
3
4
5
6
8
10
14
100
10
1
0.02
0.1
1
f*d/c
10

Fig. 6

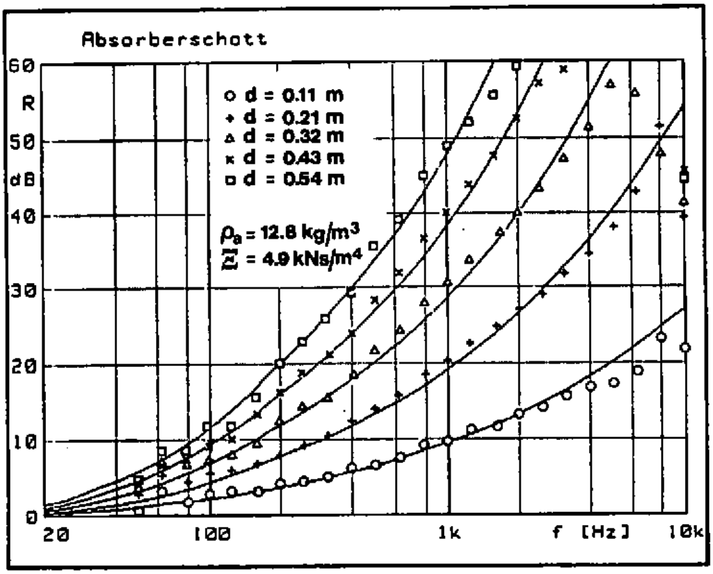

Absorberschott
o d = 0.11 m
+ d = 0.21 m
Δ d = 0.32 m
x d = 0.43 m
□ d = 0.54 m
ρa = 12.8 kg/m³
Ξ = 4.9 kNs/m⁴
60
R
50
dB
40
30
20
10
0
20
100
1k
f [Hz]
10k

Fig. 7