PERIODICA POLYTECHNICA SER. MECH. ENG. VOL. 36, NO. 1, PP. 17–31 (1992)

REDUCING THE DISCHARGE SIDE NOISE EMISSION IN RADIAL FLOW FANS WITH ABSORBENT COVER INSIDE THE SPIRAL CASING

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Received: October 28, 1992.

Abstract

The discharge noise emission in radial flow fans may be reduced by providing a sound absorbing inner cover for the spiral casing. The literature is fairly brief on the aspects of constructing such fans with covered walls. This paper summarises the results of tests, which give an insight into the effect of the variables and show solutions to certain constructional problems. On the basis of the tests it is possible to propose an absorbing structure with a good noise reduction effect, which is also economical to make. The experimental machines used made it possible to apply many variations of quality and quantity in the absorbing material and of the position and shape of same in the absorbing structure.

Keywords: radial flow fan, noise control.

1. Introduction

The literature on the noise emission of radial flow fans is not very extensive. The existing papers agree that if certain aspects are taken into consideration when designing the spiral casing, the tonal component of fan noise that operates in the usual peripheral velocity range will not be excessive [1]. Reducing broad band noise component, however, is not so simple, though the beneficial effect of smooth suction is mentioned in the literature, and it is advisable to provide a low rate of turbulence at the inflow and have a gap with an appropriate geometry. Vorticity distribution along the blades does not seem to affect the broad band spectrum under certain circumstances [2], but the problem has not been completely investigated. Some minimal noise can be expected from even the quietest fan, due to principles of operation (rotational noise, wake noise).

All these problems combined raised the necessity to provide an absorbent cover inside the spiral casing of the fan. Even the most detailed papers of the few discussing the subject list the results of the experiment on only one machine, without changing the parameters of the absorbent structure [3, 4], however, the idea was born in 1965 and manifested in a patent [5]. It seemed therefore reasonable to set up a research project, helping to a better understanding of the problem. As a first step in a series of tests aerodynamic and acoustic characteristics have been established of a reference fan without any sound absorbing structure. Later the sheet metal of the volute has been changed for various sound absorbing structures while making sure that the free flow cross-section remained unchanged, thus keeping the flow parameters φ, ψ and η of the fan as constant as possible. Concerning the absorbing structure of the experimental machine, the following five parameters have been varied:

1. The specific acoustic impedance of the sound absorbing material placed into the sound absorbing structure.
2. The position of the sound absorbing structure on the spiral casing surface.
3. The quantity of the sound absorbing material in the sound absorbing structure.
4. The depth of the space in the sound absorbing structure.
5. The shape of the sound absorbing structure.

Having selected from a series of tests by varying one parameter only the most silent one, it was this one which was undergoing the change of the next parameter. If, for one parameter, there were two equally good values or one was more favorable from another aspect, two or even three versions were tested. At the end we got the most silent version of a radial flow fan with covered walls, on the basis by the variation of the above parameters.

2. Description of the Experimental Machine and the Method of Measurement

A spiral casing was used which could easily be altered. *Fig. 1* shows a sketch of its main dimensions.

After reference tests, the single sheet metal of the spiral casing was changed for a sound absorbing structure. For reasons of stiffness, we used a perforated metal sheet to become the inner side of the sound absorbing structure. The perforated spiral casing had the same shape as the original, the free flow cross-sections remained unchanged. In every case the sheet providing air-tightness was placed last on the outside. In constructing the sound absorbing structure, we also considered the findings of papers [6, 7 and 8]. The sketches of the sound absorbing structures are always shown at the description of the relevant assemblies.

The impeller had 14 backward curved blades and was fixed directly onto the shaft of a D.C. balance motor. The measured points were taken at a rotational speed of about $n = 1700$/min. An inlet test cone (ISO 5221) was used to determine the flow rate. The throttling device was at the end of the

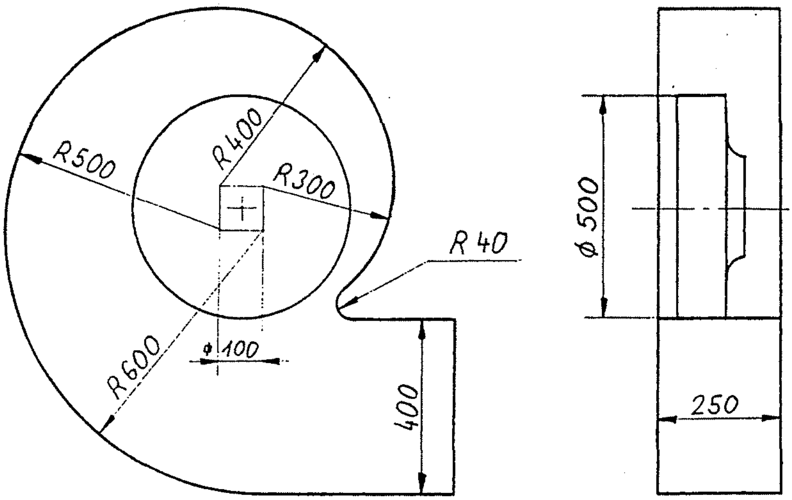

Fig. 1.

discharge duct. Measuring the torque was made possible by balancing the D.C. motor. We adhered to the regulations of the MSZ 11110 Hungarian Standard. Beside determining the characteristic curves, we also recorded the $\varphi = \varphi(p_d - p_s)$. $(p_d - p_s)$ is the difference between the static pressures on the discharge and suction sides, and φ is the flow rate coefficient.

In this way even after connecting an anechoic termination at the end of the discharge duct φ could be easily determined with the help of reading $(p_d - p_s)$ only.

When measuring the discharge side noise emission 3 microphone positions were used 4 m away from the sound source, at half-diameter of a circle, at an angle of 120° from one another, perpendicular to the tube axis, which conforms to the regulations of ISO/DIN 5136.2. *Fig. 2* shows a sketch of the experimental set-up and the microphone positions. A wind cone Type UA 0386 was mounted on a B&K microphone Type 4143. The signals from the microphone were led to a third-octave band analyser B&K Type 2112 through a preamplifier Type 2616 and to a OMC105 H type FFT analyser.

The sound power level was calculated from the averaged sound pressure levels measured in the 3 positions in the following way:

$$L_w = L + \lg\left[\frac{A}{A_0}\right],$$

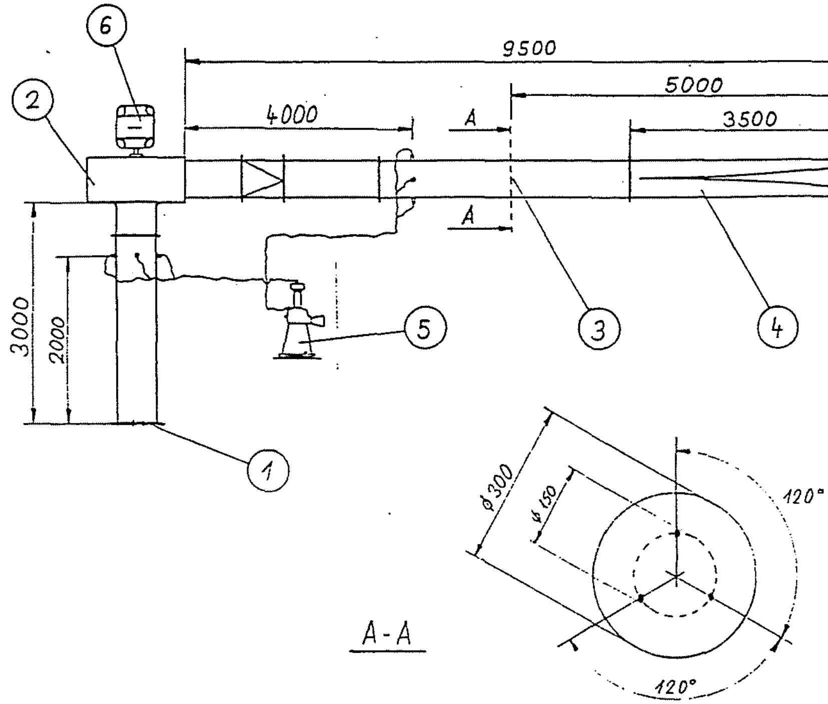

Fig. 2. 1. throttling
2. fan
3. microphone positions
4. anechoic termination
5. Betz-type micromanometer
6. D.C. balancing motor

where: L_w sound power level [dB]
L averaged sound pressure level [dB]
A cross-section area of the test duct [m^2]
A_0 1 [m^2]

3. Test Results

3.1 Starting with the reference fan the flow and acoustic characteristics are shown in *Fig. 3*.

φ, the flow rate coefficient and ψ_t, the total head coefficient are:

$$\varphi = \frac{Q}{\frac{D_2^2\pi}{4}U_2}, \qquad \psi_t = \frac{\Delta p_t}{\frac{\rho}{2}U_2^2},$$

where Q flow rate [m^3/sec]
Δp_t total head [Pa]
ρ density [kg/m^3]
D_2 outer diameter of the impeller [m]
U_2 circumferential velocity at the outer diameter [m/s]

Later we measured the flow parameters after major structural changes in the spiral casing only. The variations all along were within the tolerable accuracy range, with one exception.

3.2 As the next step the sheet metal of the volute was replaced by a perforated one and a 140 mm thick open cell polyurethane foam was placed behind it. The construction now is the same as in *Fig. 4* except the fabric cover 2.

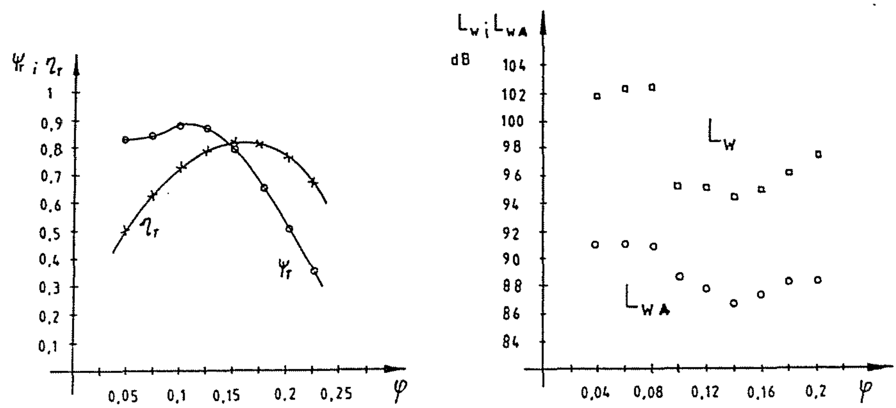

Fig. 3.

The experiments showed that the discharged total sound power level of the fan with the sound absorbent decreased by $\Delta L_w = 1.5$ dB, while the A weighted sound power level decreased by nearly 7 dB(A). However, the efficiency of the machine decreased by about 7 – 8%, and the $\psi = \psi(\varphi)$ characteristic also shifted undesirably to the left. It was due to the increased friction loss created by the segmented surface of the perforated sheet (*Fig. 5.*)

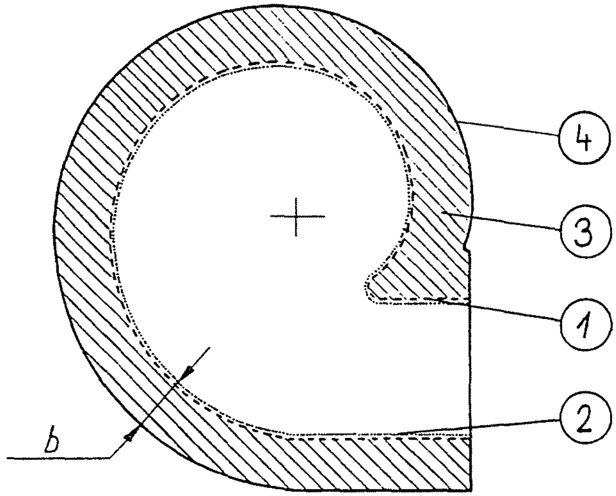

Fig. 4.
1. perforated sheet metal (1.5 mm thick), the hole diameters are 6 mm, the open area ratio is $\sigma=0.475$
2. fabric cover
3. sound absorbing material, open cell polyurethane foam with a bulk density of $\rho_t=31\,\mathrm{kg/m^3}$, with the specific acoustic impedance ratio of $\Xi=16.5$
4. outer cover sheet

Where: σ the open area ratio by:

$$\sigma=\frac{A_0}{A_t}$$

A_0 the open part of the perforated surface

A_t the total sheet surface

Ξ the specific acoustic impedance ratio:

$$\Xi=\frac{\left(\frac{\Delta p}{v}\right)_{\mathrm{st}}}{(\rho a)_{\mathrm{air}}}$$

$\left(\frac{\Delta p}{v}\right)_{\mathrm{st}}$ is the measured specific flow resistance

$(\rho a)_{\mathrm{air}}$ is the characteristic impedance of the air

To improve the acoustic parameters other than at the expense of the flow characteristic, we covered the perforated sheet on the flow side with an acoustically transparent fabric. The specific acoustic impedance ratio was $\Xi=0.7$, and its mass/surface unit ratio was $\rho_s=0.192\,\mathrm{kg/m^2}$. The fabric was glued to the sheet and we made sure that no glue penetrated the holes. In every following test, we used the perforated sheet of the above

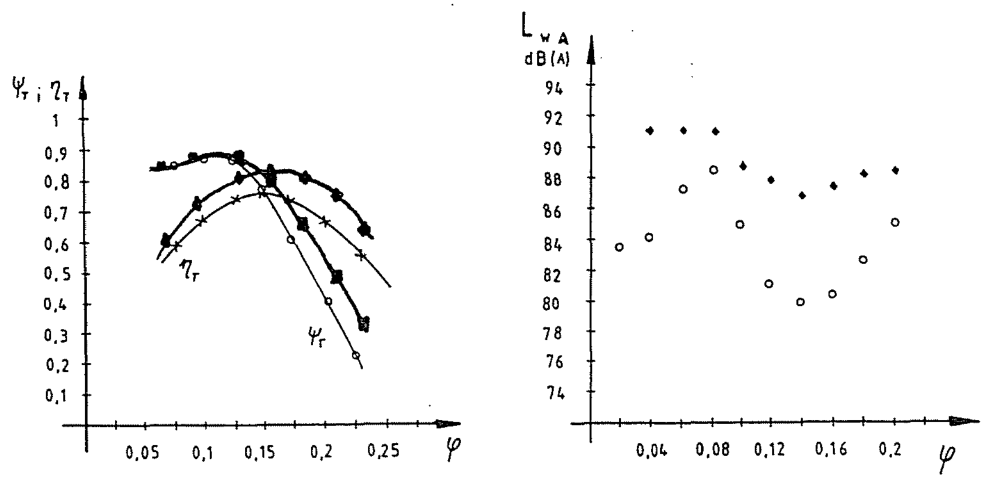

Fig. 5. ◇, □ hard wall spiral casing
○, × covered spiral casing

mentioned size with the fabric cover. Now the flow characteristics reached the values of the reference fan within the range of the measuring accuracy.

3.3 In the five steps that followed, we changed the specific acoustic impedance of the sound absorbing material shown in *Fig. 4*. The results are shown in *Fig. 6*.

The figure indicates that the most favorable fan was the one equipped with Therwoolin sound absorbent of 96 kg/m^3 or 40 kg/m^3 bulk density. For economy reason, we used the 40 kg/m^3 bulk density absorbent.

3.4 The next step was to place the sound absorbent structure onto different parts of the spiral casing surface. Only the shroud of the spiral casing (*Fig. 7/a*) or only the front panel (*Fig. 7/b*) or the shroud and the front panel together (*Fig. 7/c*) were covered by the sound absorbing structure.

The acoustic characteristics measured are shown in *Fig. 8*

In spite of the fact that the shroud surface area of the spiral casing and of the front panel were nearly equal, covering the shroud resulted in a much better attenuation. Covering both the shroud and the front panel at the same time did not produce any significant decrease in the noise power level. It is therefore that only the shroud of the fan spiral casing was covered in the following.

3.5 In the next three tests, the quantity of sound absorbing material in the sound absorbing structure was varied. A fan without any sound absorbing material (*Fig. 9/a*), an other one completely full of it (*Fig. 9/b*)

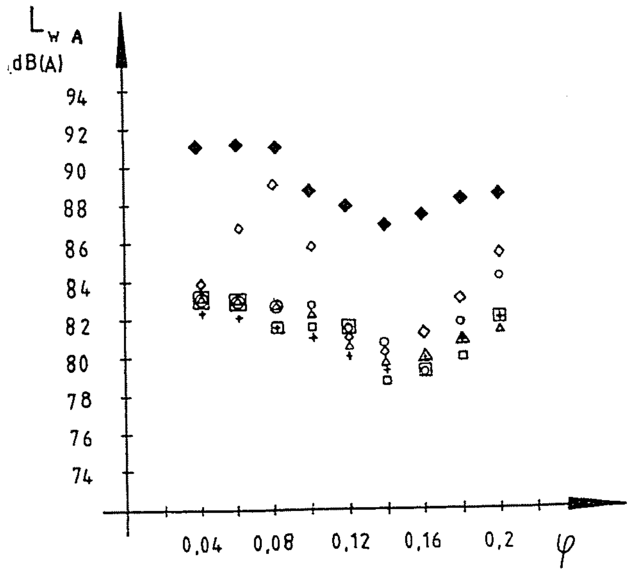

Fig. 6. ◇ hard wall spiral casing

spiral casing, covered with different sound absorbing materials:

○ $\rho_b = 10\,\mathrm{kg/m^3}$ Therwoolin $\Xi = 0.8$

+ $\rho_b = 40\,\mathrm{kg/m^3}$ Therwoolin $\Xi = 4.6$

□ $\rho_b = 96\,\mathrm{kg/m^3}$ Therwoolin $\Xi = 17.3$

◇ $\rho_b = 31\,\mathrm{kg/m^3}$ Polyurethane foam $\Xi = 16.5$

△ $\rho_b = 43.5\,\mathrm{kg/m^3}$ Köszig $\Xi = 1.3$

and with a 20 mm thick absorbent layer affixed on the perforated sheet was tested. In this last case a 120 mm space remained between the absorbing material and the outer shroud (*Fig. 9/c*).

Results are shown in *Fig. 10.* As expected, the fan was nearly equally good as regards sound power level reduction when completely full and with a 20 mm absorbent layer. It is interesting to note that even the construction without the sound absorbing material showed a nearly 7 dB(A) sound power level reduction.

3.6 The effect of reducing the space behind the perforated sheet was recorded in the next series of tests. In every case the outer sheet shroud was placed closer, while the sheet covered with a fabric, which limited the free flow space, kept its position (*Fig. 9/a,* dimension *b*). Results are indicated in *Fig. 11.*

3.7 In the next four tests again the space was changed, but this time a 20 mm thick Therwoolin sound absorbing material was sewed to the perforated sheet. Results are shown in *Fig. 12.*

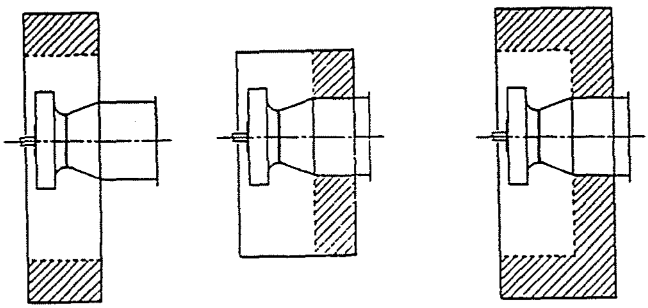

Fig. 7.

It can be seen from the results of the two tests that a significant sound absorbing effect was created even when reducing the thickness of the absorbing structure considerably.

3.8 In the last three tests the shroud of the spiral casing was replaced by an absorbing structure with variable radial dimensions. By placing the spiral casing in an inclusive parallelepiped whose side was 40 mm outside the tangent surfaces of the spiral casing. Three versions were tested. When completely full of sound absorbing material (*Fig. 13/a*), when completely empty (*Fig. 13/b*) and with the 20 mm sound absorbing material sewn to the perforated sheet (*Fig. 13/c*).

Fig. 14 shows that there was no significant difference between the three cases and this series proved to be the best of all tested. It should be noted that comparing the sound power level spectra, an average attenuation in a broader range was achieved with the absorbent where radial dimensions changed, while a greater attenuation in a narrower range was achieved with the properly designed evenly thick absorbent (*Fig. 15*).

In air conditioning it seems that the noise spectrum reflected by a favorable subjective judgement may be approached by a –5 dB/oct line [9]. The deviation of the noise power level spectrum from the –5 dB/octave line was the biggest in the 2000 Hz region with the hard walled spiral casing. Also a considerable improvement has been achieved by using absorption in

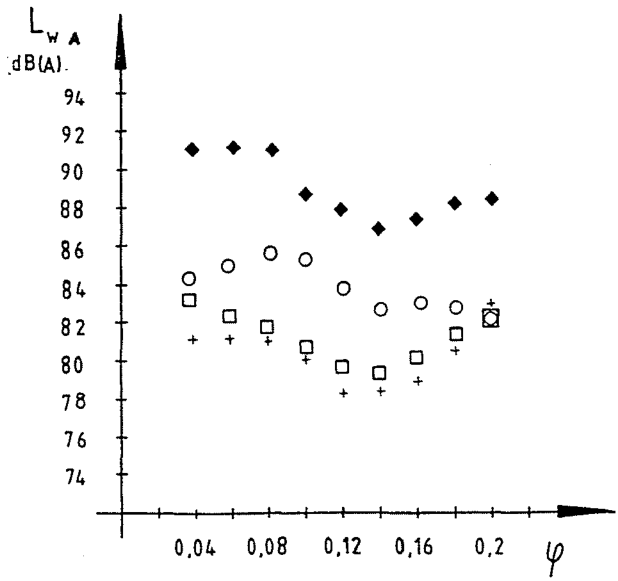

Fig. 8. ◇ hard wall spiral casing
□ spiral casing as *Fig. 7/a*
○ spiral casing as *Fig. 7/b*
\+ spiral casing as *Fig. 7/c*

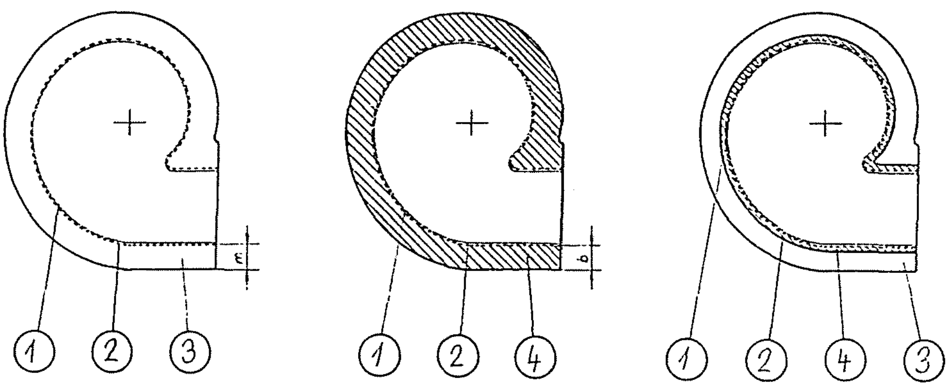

Fig. 9. 1. fabric cover 2. perforated sheet metal 3. cavity 4. sound absorbing material

the casing, a further damping effort concerning the $f_m > 10^3$ Hz would be favorable if the -5 dB/oct. line should be approached.

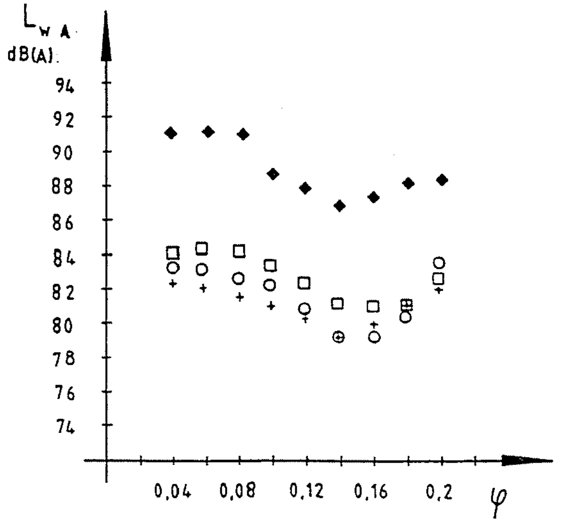

Fig. 10. ◇ hard wall spiral casing
□ spiral casing as *Fig. 9/a*
+ spiral casing as *Fig. 9/b*
○ spiral casing as *Fig. 9/c*

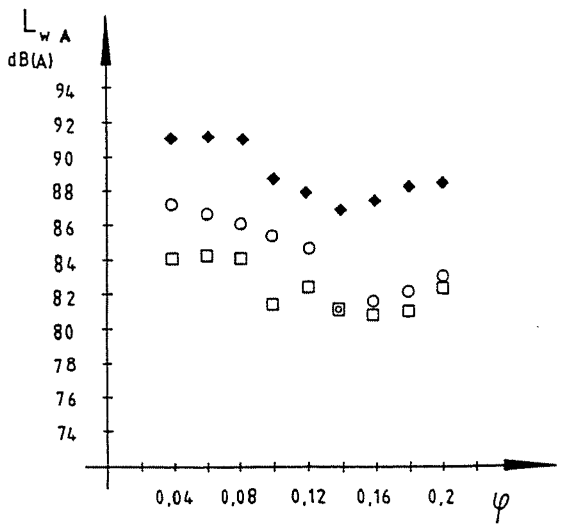

Fig. 11. ◇ hard wall spiral casing
○ spiral casing as *Fig. 9/a*, $b = 20$ mm space
□ spiral casing as *Fig. 9/a*, $b = 140$ mm space

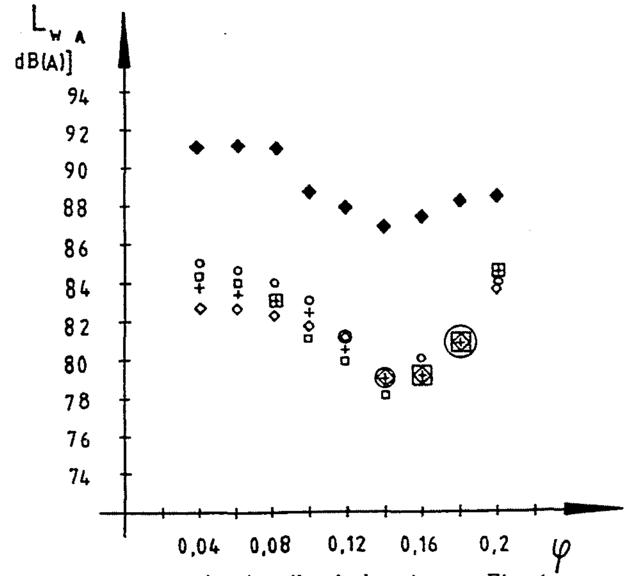

Fig. 12. ◇ hard wall spiral casing as *Fig. 1*
○ spiral casing as *Fig. 9/c*, $m = 0$ mm empty space
□ spiral casing as *Fig. 9/c*, $m = 20$ mm empty space
\+ spiral casing as *Fig. 9/c*, $m = 40$ mm empty space
◇ spiral casing as *Fig. 9/c*, $m = 120$ mm empty space

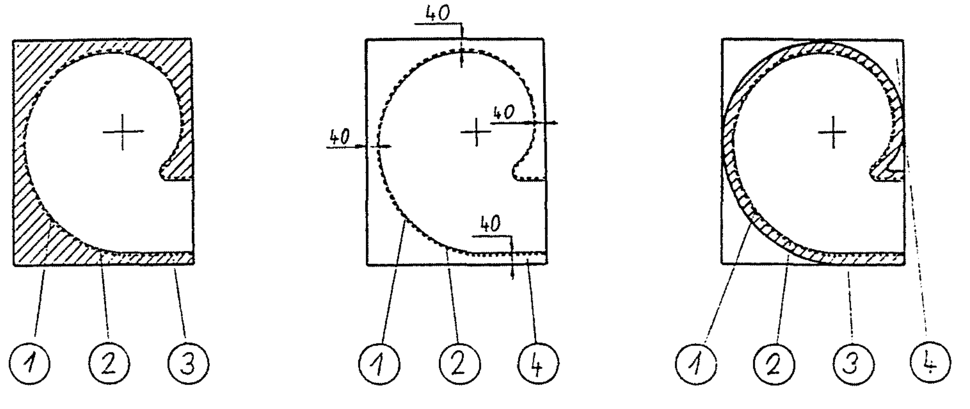

Fig. 13. 1. fabric cover
2. perforated sheet
3. sound absorbing material
4. space with variable depth

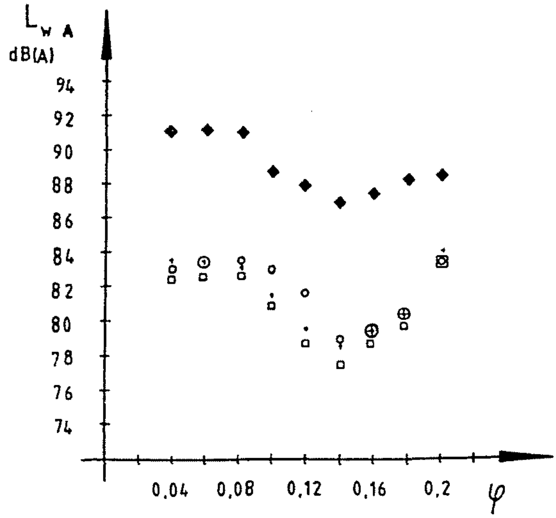

Fig. 14. ◇ hard wall spiral casing
□ spiral casing as *Fig. 13/a*
○ spiral casing as *Fig. 13/b*
\+ spiral casing as *Fig. 13/c*

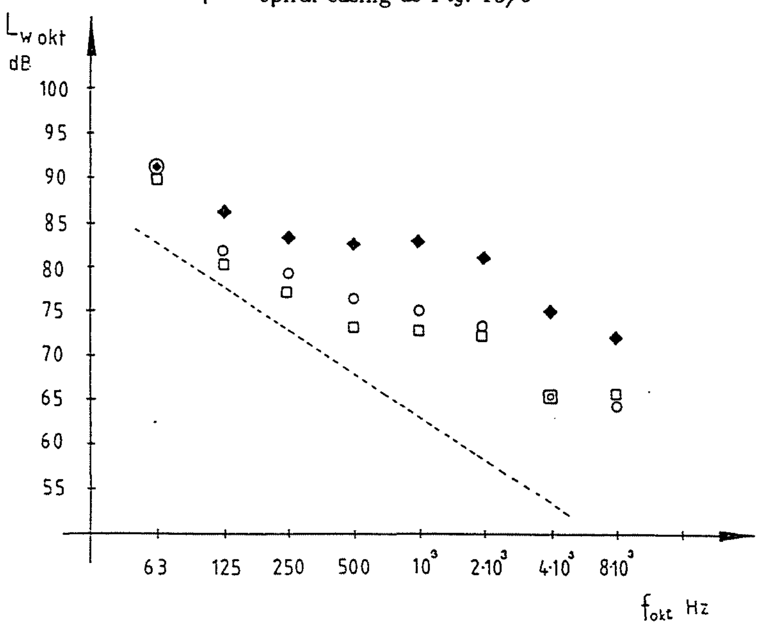

Fig. 15. ◇ hard walled spiral casing
○ spiral casing as *Fig. 4*
□ spiral casing as *Fig. 13/a*

4. Summary

A series of tests shows that the discharge side noise emission of a radial flow fan can be reduced effectively with a sound absorbent covering of the inside of the spiral casing. Based on test results described in the previous chapter the following conclusions may be drawn:

The total sound power level decreased a mere 1.5 – 2 dB in the various cases when the A weighted sound power level drop was about 7 dB(A).

The first series of tests (*3.3*) shows that the noise reduction depends little on the acoustic hardness and bulk density of the materials used. So mounting an unnecessary extra mass can be avoided by using sound absorbents with lower bulk density. The second series (*3.4*) proved that the absorbent is best placed on the shroud of the spiral casing. The combined covering of the shroud and the front panel gave no advantage that justifies the increased technical complexity.

The third series (*3.5*) proved that it is not necessary to fill completely the structure with absorbing material, but the 20 mm layer sewn to the perforated sheet is enough. An interesting result is that if no absorbing material was used whatsoever, but a layer of fabric covered the perforated sheet the noise reducing effect of the structure decreased only by 1.5 dB(A).

Fans without absorbing material, only having a casing structure as *Fig. 9/a* are less expensive to manufacture and represent a 'clean' technology.

The result of the fourth (*3.6*) and the fifth (*3.7*) test series suggest that the size of the space effects noise considerably at low φ values which is connected with the rich low frequency content of the noise in this duty point region.

In *3.8* then non constant radial space size was dominant and proved to be the best.

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