1982

Reduction of the Compressor Inlet Infrasound Emission

J. R. Piechna

REDUCTION OF THE COMPRESSOR INLET INFRASOUND EMISSION

Janusz R. Piechna
Warsaw Technical University
Poland

ABSTRACT

The infrasounds attenuation is inefective mainly because of the long sonic wave diffraction, so that the reduction of the infrasounds source is the only acceptable method. Unfortunately low frequency pressure pulsation has the great influence on the compressor operation. Each change in suction line results in change of compressor volumetric efficiency.
The paper presents examples of numerical investigation and results of some experimental verification for several inlets with mufflers of different types.
The infrasound emission reduction, its efficiency that influence the compressor operation and correlation with mufflers dimensions are in the centre of interest.

INTRODUCTION

Though the perception of sound by the human ear is limited to frequency higher than 16-20 Hz, as the man responce to infrasound exposure, the mechanical, physiological and psychological effects are observed.
The low speed air compressors suction installations very often are one of a strong sources of infrasounds. The one of such installation became a subject of presented investigation. Vibration was the problem.
The measurements carried out near the house located about 150 m from the compressor plant gave the results shown in Figure 1 and Figure 2. Figure 1 presents acoustic spectrum measured near the house and Figure 2 is the spectrum of acceleration of the window glass. Some further measurements excluded other sources of vibration and the coclusion was that the mechanical vibration is due to the acoustic waves. Noise rating curves additionaly drawn in Figure 1 show that rather high sound pressure levels are not annoying from the acoustic point of view. On the other hand, mechanical vibration in the low frequency range caused by the infrasound components produce annoyances which are individually very different.
So the problem of reduction of vibration was accompanied by the problem of the infrasound reduction.

OBJECT OF INVESTIGATION

The considered compressor plant was equiped with three two-stage double-acting reciprocating compressors VC 750. Each compressor had its own sepaprate suction line / Figure 3 /.
In Figure 4 the results of computation and the results of measurements conducted at one of the compressors inlet are presented. The small schematic drawing in Figure 4 explains used nomenclature.

METHODS OF ATTENUATION

Attenuation of infrasounds in far field is rather problematical because of very long waves diffraction. So the only practical method of sound pressure levels decreasing is the reduction of sound source.
The several different methods of infrasound emission have been cosidered. The schemes of considered types of mufflers [1,2,3,4,5] are shown in Figure 5.
The simplest method is the change of the suction pipe length /b/. It is possible to use a multiple expansion chamber /c/, or Helmholtz resonator type of muffler /d/. Also resistive types of mufflers /e,f/ have been taken into account.

REQUIREMENTS

Each method of attenuation has its own advantages and disadvantages [1,2,3] . So it is necessary to specify technical requirements for mufflers. There are many different requirements [5,6] but three of them seem to be the most important. They are: sufficiently high attenuation, little influence on compressor operation and small dimensions. And in this case cosideration was limited only to the requirements mentioned above.

MATHEMATICAL MODEL

The simple mathematical model of compressor and suction installation has been used to optimise the muffler configuration.
The overal mathematical model consists of compressor cylinders model, model of flow through the valves, suction line model with mufflers and acoustic model.

Compressor cylinders model

For modeling the double-action cylinder a modified polytropic compression equation was used[4] . Assuming that the density during suction and discharge period is constant one can get

$$\frac{dp_c}{dt} = \frac{n\,p_c}{V_c}\left(Q_{v_s} - Q_{v_d} - \frac{dV_c}{dt}\right);$$

for the outer and inner side of cylinder where

$$V_{c_{in}} = A_{in}\,r\,(1-\cos\omega t) + V_{m_{in}};$$

and

$$V_{c_o} = A_o\,r\,(1-\cos(\omega t+\gamma)) + V_{m_o};$$

For reference purposes crank angle $\varphi = \omega t$ is considered zero at bottom dead center.

Model of flow through the valve

Because of limited solution only to the low frequencies, simplified model of flow [7,8] was used

$$Q_v = \alpha F\sqrt{2\Delta p/\varrho}; \qquad \Delta p > 0;$$
$$Q_v = 0; \qquad \Delta p < 0;$$

for both suction and discharge valve.

Suction line model with mufflers

The lumped parameter model of nonstationary flow seemed to be sufficiently accurate [4,8,9,10] for low frequencies oscillations.
So the set of differencial equations

$$\frac{dQ}{dt} = \frac{1}{M_a}\Delta p - \frac{R_a}{M_a}Q; \qquad M_a = \frac{\varrho\Delta x}{A};$$
$$\frac{dp}{dt} = \frac{1}{C_a}\Delta Q; \qquad C_a = \frac{A\Delta x}{\varrho a^2};$$
$$R_a = \frac{\lambda u_o}{2d};$$

was build for suction pipe. Also the mufflers models were done basing on the lumped parameter model. And only the resistive type of muffler needs some explanation.
The flow through the oriffice was modeled in the way similar to that used for valves. It was assumed that kinetic energy of flow in the oriffice is completely dissipated inside the chamber.

Acoustic model

Far field acoustic pressure level was found by expansion of the flow rate at the inlet of suction pipe into a Fourier series and by the use of model presented in [8] .

$$p_{a_n} = \frac{\varrho_o\,\omega_n\,Q_n}{4\pi R};$$

NUMERICAL SOLUTION

The set of differential equations has been solved by the Runge-Kutta method. For programing, the symulating language MIMIC and the standard Fortran have been used. The use of Fortran significantly reduces the time of computation in comparison with MIMIC.

RESULTS

The influence of muffler on the compressor operation was considered in following way. The average flow rate for the compressor without the suction pipe has been computed and taken as a refference. So every computed flow rate has been compared with the ref-erence flow rate. A coefficient of volumetric efficiency was defined as

$$L = \frac{\bar{Q}}{\bar{Q}_{ref}};$$

Figure 4 shows results of computation for the existing suction line configuration. In this case, because of the resonance with the second harmonic, the L coefficient was 1.09 .
The results of pressure measurements done on the real installation, shown by dashed line in Figure 4, fit the computations quite well.

The first considered modification of the suction line was the change of its length. From the analysis made by another method /harmonic solution of the wave equation [11] / the length of the inlet pipe about 7 m seemed to be the optimum. In Figure 6 the results of computation are shown. The attenuation of about 9 dB and L coefficient equal 1.04 have been expected.
Figure 7 comprises the results of computation carried out for the suction line with multiple expansion chamber. Dimensions of the muffler were chosen acceptable from technical point of view. Attenuation of about 9 dB and L coefficient 1.02 was predicted.
Figure 8 shows numerical results achieved with Helmholtz resonator type of muffler.

The attenuation of the second dominanting frequency harmonic is 13.5 dB and L coefficient 1.01 . The high amplitude of pressure pulsation inside the resonator chamber is characteristic.

Figure 9 presents a series of flow rates curves at the inlet of suction line with resistive type of muffler. There are six curves /Figure 9a/ for different values of chamber volume /range V= 0-1 m^3, L_{min}=1.005, maximum attenuation 17 dB/.

The muffler construction changes /series of oriffices or series of mufflers with the same overall volume/ do not change significantly the attenuation in low frequency region /Figure 9b/.

In Figure 10 the results of commutation for moderate size resistive muffler /V= 0.25 m^3/ are shown. Attenuation was 12 dB and L coefficient 1.02 .

Figure 11 shows analysed mufflers and the compressor dimensions.

After careful consideration, it has been decided to modify the one compressor inlet by lengthening pipe to 7 m and to place a Helmholtz resonator type of muffler on a second compressor.

Unfortunetly due to some technical problems the real length of pipe after modification was 8 m. In Figure 12 the spectra of the pulsating pressure in pipe measured in the neighbourhood of compressor, and the noise spectra measured near the compressor plant /before and after modification/ are shown. The dominanting harmonic /14 Hz/ has been damped about 10 dB but the increase of another harmonics values can be observed.

Figure 13 shows the results of measurements carried out on the compressor with Helmholtz resonator muffler. The results are quite satisfying.

Figure 14 shows the acoustic pressure spectra measured inside the house before and after modifications. The vibration practically disapeared.

CONCLUSIONS

The reduction of the sound and infrasound is not directly connected with attenuation of the pressure pulsation.

The reduction of the low frequency noise is always connected with the lost of compressor volumetric efficiency /lost of dynamic supercharging/.

It is possible to get 10-15 dB reduction of infrasound emission by mufflers with technically acceptable dimensions.

The Helmholtz resonator type of muffler gives good results without the additional effects.

The resistive type of muffler seems to be very interesting solution [1,3] particularly for low and moderate frequencies.

The lumped parameter model proved to be sufficiently exact for low frequencies.

NOTATION

a - speed of sound
A - piston area
C_a - acoustic compliance
F - valve slit area
l - strap length
M_a - acoustic inertance
n - polytropic exponent
p - pressure
Q - flow rate
$\bar{Q}$ - average flow rate
Q_v - flow rate through the valve
R - distance from the compressor
R_a - acoustic reactance
r - crank throw
t - time
V_m - clearance volume

α - flow coefficient
φ - crank angle
λ - friction coefficient
ω - rotational velocity
ρ - density

SUBSCRIPTS

c - cylinder
d - discharge
in - inner part of cylinder
o - outer part of cylinder
s - suction
v - valve

REFERENCES

1. J.Brablik "Attenuation of gas pulsations using a perforated tube" Procd., 1976. Purdue Compressor Technology Conference

2. D.D.Davis,G.M.Stokes,D.Moore,G.L.Stevens "Theoretical and experimental investigation of mufflers with comments on engine exhaust muffler design", NACA Report 1192

3. J.F.T.MacLaren,A.B.Tramschek,O.F.Pastrana "A study of unsteady gas flow in perforated pipes in compressor systems",Purdue Compressor Technology Conf.,Procd.,1976.

4. J.Piechna "Współpraca sprężarki tłokowej z układem tłumiącym pulsacje ciśnienia" /Cooperation between reciprocating compressor and pressure pulsation damping system/, Ph.D. Thesis, Warsaw Technical University, 1981.

5. R.Singh,W.Soedel,"A review of compressor lines pulsation analysis and muffler design research",Purdue Compressor Technology Conference, Procd., 1974.

6. W.W.Nimitz "Reliability and performance assurance in the design of reciprocating compressor installations", Purdue Compressor Technology Conf., Procd., 1974.

7. J.Brablik,"Computer simulation of the working process in the cylinder of a reciprocating compressor with piping system",Purdue Compressor Technology Conference, Procd., 1974.

8. Mutyala,B.R.C.,"A mathematical model of Helmholtz resonator type gas oscillation discharges of two-cycle engines", Ph.D. Thesis, Purdue University,1975.

9. D.Margolis,"Modeling of two-stroke internal combustion engine dynamics using the bond graph technique", SAE Transaction 750860

10. D.F.Miller,B.W.Hatten,"Muffler analysis by digital computer", ASHRAE Transactions No 1710

11. T.Abe,T.Fujikawa,S.Ito,"A calculating method of pulsation in a piping system" Bulletin of the JSME Vol.13,No.59,1970

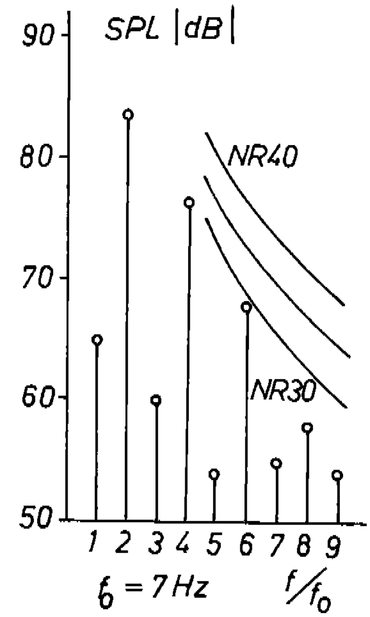

Figure 1. Sound spectrum near the house

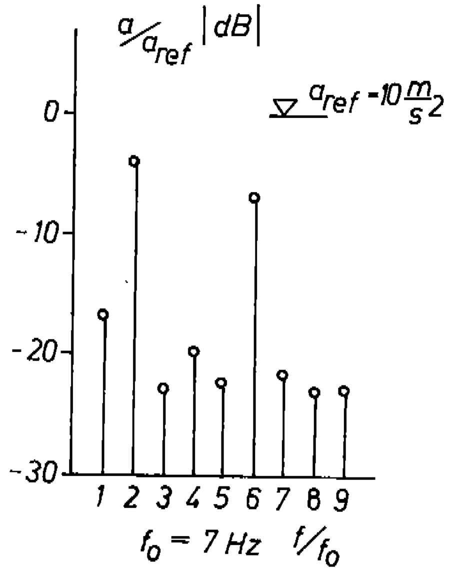

Figure 2. Spectrum of acceleration

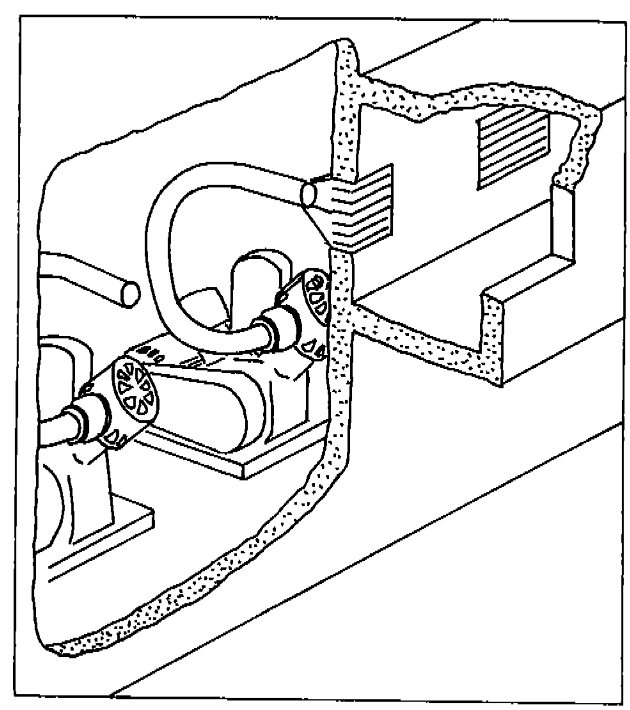

Figure 3. Compressor arrangement

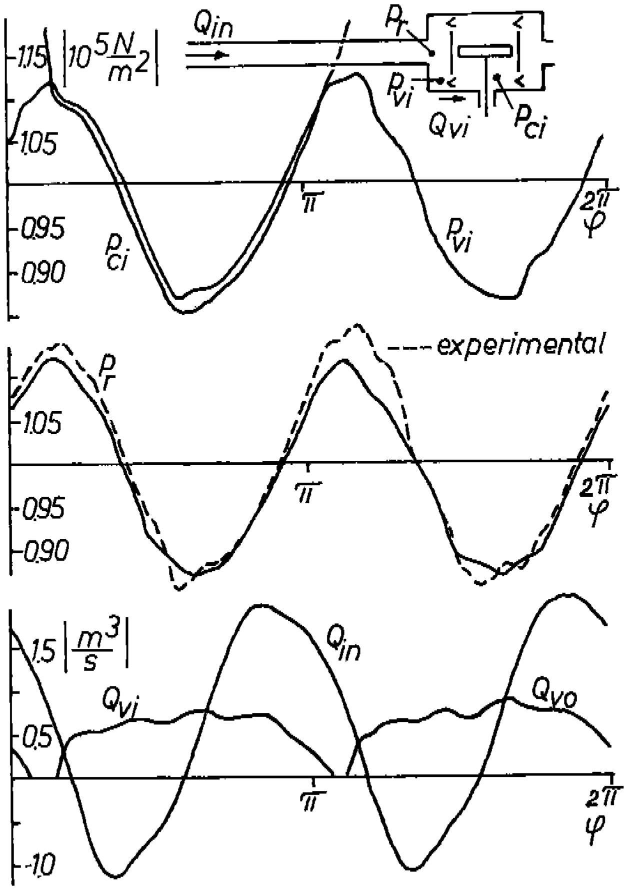

Figure 4. Standard inlet configuration

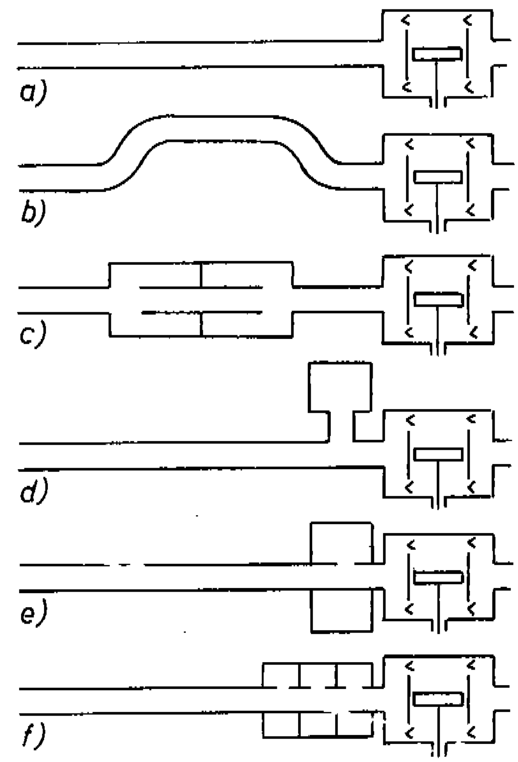

Figure 5. Compressor inlet modifications

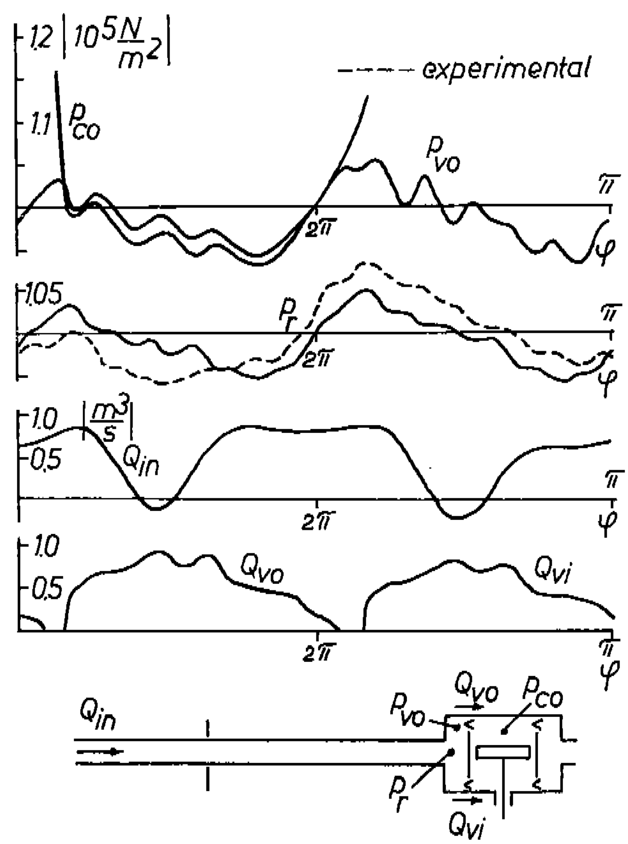

Figure 6. Elongated inlet pipe

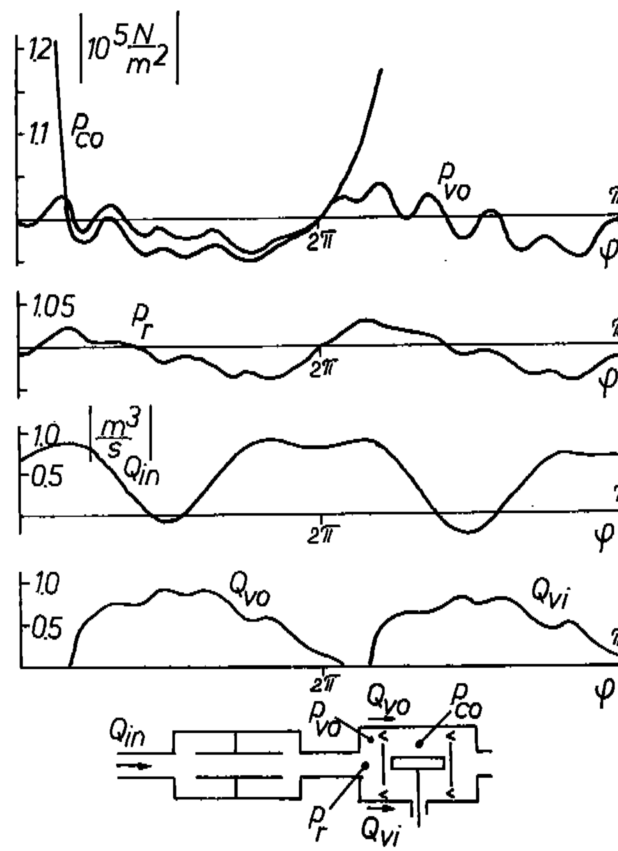

Figure 7. Inlet with multiple expansion chamber

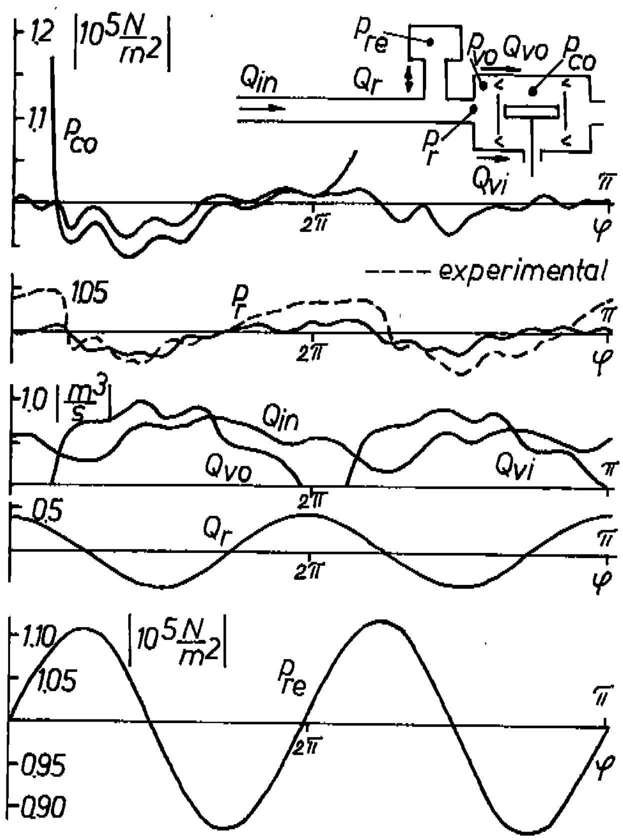

Figure 8. Inlet with Helmholtz resonator

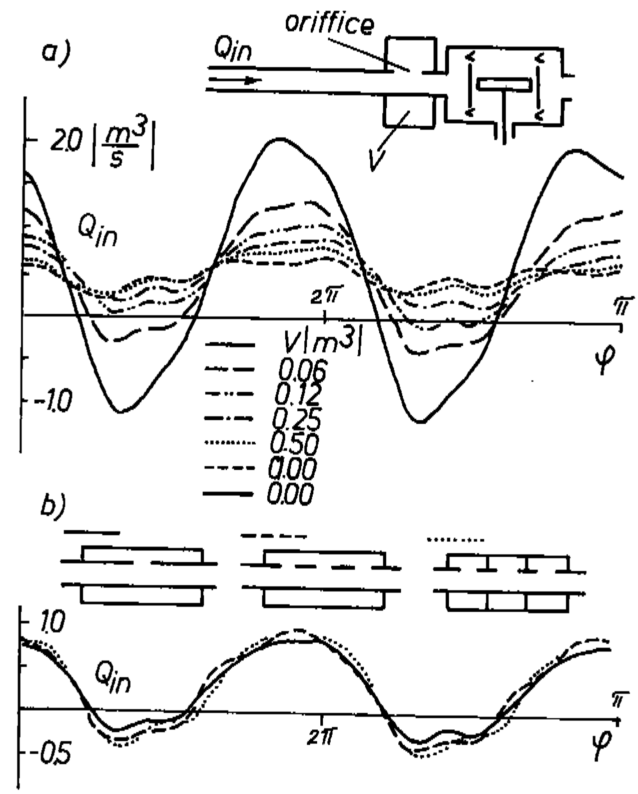

Figure 9. Influence of the resistive muffler volume and shape

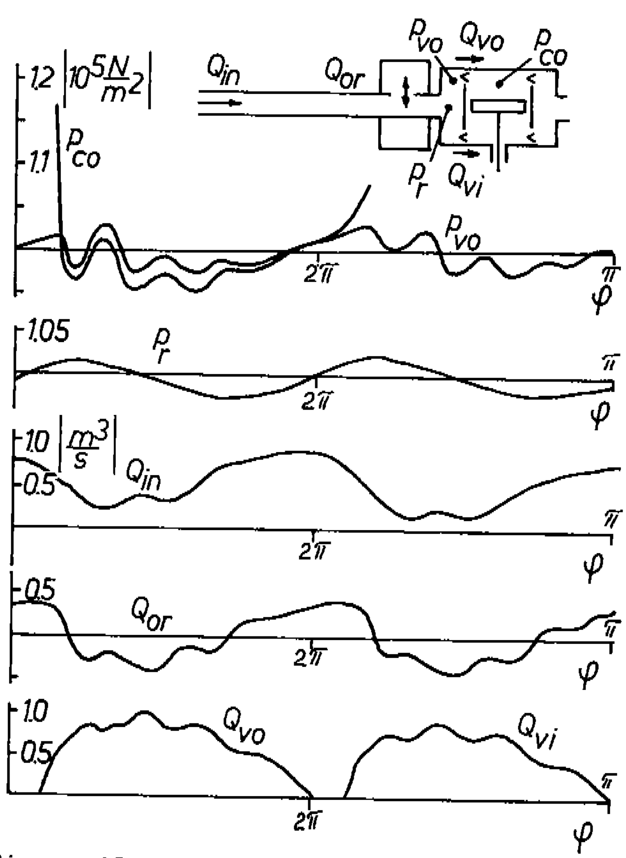

Figure 10. Inlet with resistive type of muffler

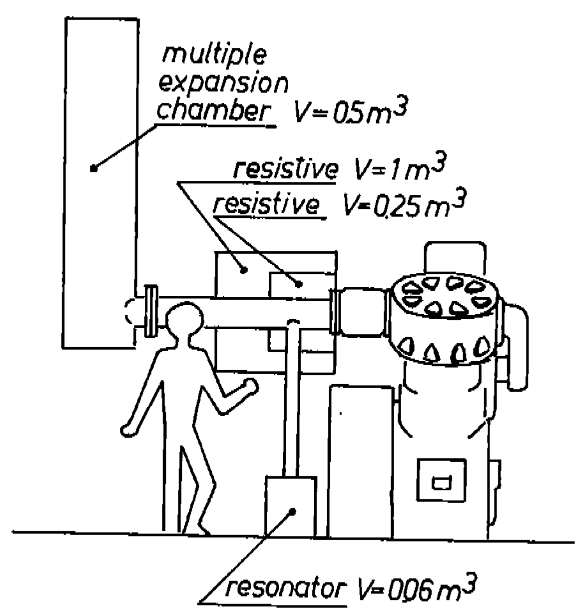

Figure 11. Comparison of dimensions

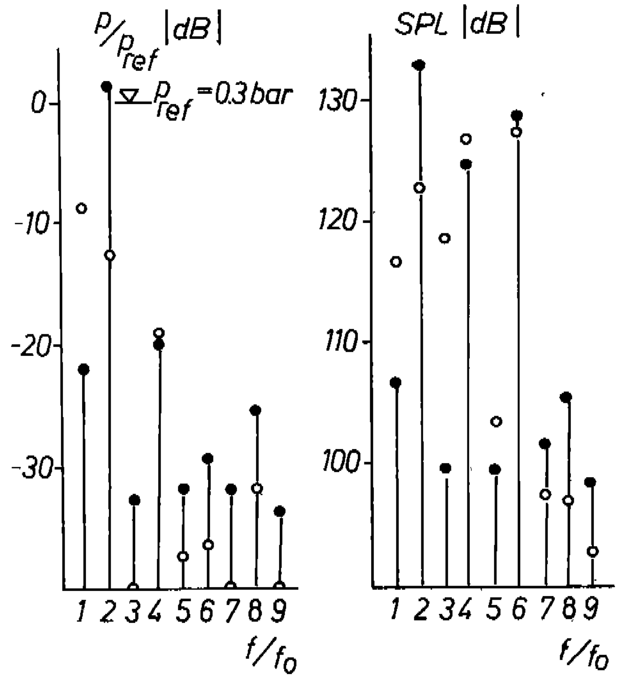

Figure 12. Elongated inlet pipe

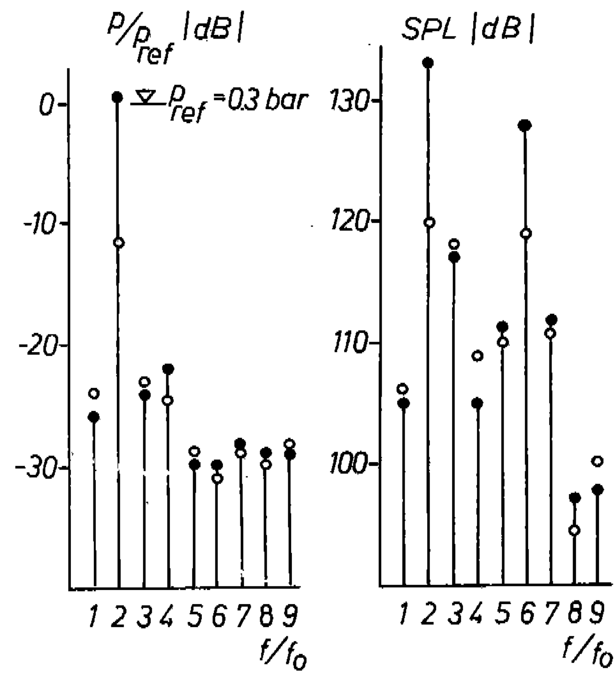

Figure 13. Inlet with Helmholtz resonator

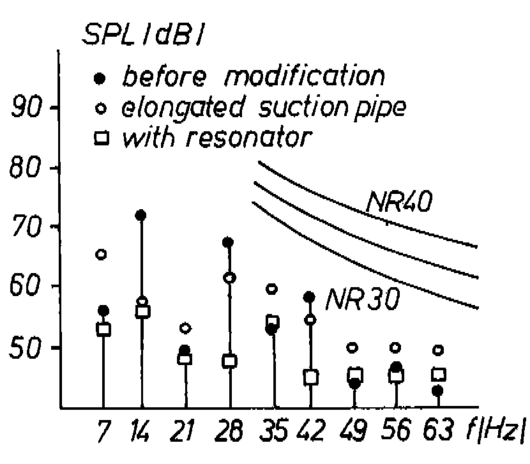

Figure 14. Sound spectra inside the house