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Influence of Constructional Parameters Of Small Reciprocating Compressors on Sound Power Emissions

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Noise in environment is an increasing problem in industrial countries and is often strongly restricted by governmental laws. Therefore reducing sound levels must be one target in every design process. This paper describes in detail the influence of three constructional parameters on the total sound power emitted by a small reciprocating compressor ($V_{\mathrm{H}} = 159\,\mathrm{m}^3$) with reed valves. Measurements (sound power levels, sound pressure levels, FFT-analysis, time-signal analysis) on a test bed situated in an anechoic room by varying the operating conditions – for example – rotational speed, maximum pressure, and suction terms, were done.
These measurements are: Sound power measurements with regard to valve parameters (thickness of the reed, cylinder clearance and material of the valve plate) were achieved. Theoretically researches with the aid of computational methods (FEM-Analysis with Ansys, Calculations with MatLab) are done which show very good accordance compared to the sound level measurements. These results should lead to a better understanding of the causal connection between constructional (valve) parameters/operating conditions and the resulting sound power emissions.
Finally the differentiation of airborne and structure born noise may be helpful to minimize the noise emissions of compressors already within the design process.

1 Introduction

Sound power emissions of small reciprocating compressors [1, 2] is a research project between Hörbiger Ventilwerke Schongau and the Vienna University of Technology that should improve the knowledge in noise prevention of air-compressors. Within the duration of many years new findings in the field of constructional methods to obtain lower sound levels are achieved. More informations can be found in [5, 6] and [7].

In this article new results of test bed measurements using other materials of the valve plate are presented. Additionally this results are summarized with the results of varying cylinder clearance and thickness of the reeds and discussed in detail.

2 Measurement terms

For all tests a small reciprocating compressor (cubic capacity $V_{\mathrm{H}} = 159\,\mathrm{m}^3$, diameter of the cylinder $D = 75\,\mathrm{mm}$) – as available in busses or trucks for pneumatic brakes – is used. The compressor is situated in an anechoic testing room, described in detail in [4, 5]. Sound level measurements were done with direct intensity measurement method [3] (sound power L_{P}), using a Larson&Davis 2-channel Real-Time-Analyzer LD 2900.

2.1 Measurement conditions

There are two measurement cases which were investigated.

- Compressor with free inlet port (no suction tube)
- Compressor with suction tube installed

The test bed measurements are done at pressures 0 bar, 2,5 bar and 5 bar and at rotational speeds within the range of 1000–2000 min^{-1}.

2.2 Varied valve parameters

The three following constructional valve parameters are varied and the sound power levels are therefore measured in every measurement point (rotational speed n and pressure p within the range described in 2.1).

Figure 1: Conventional valve plate made of steel

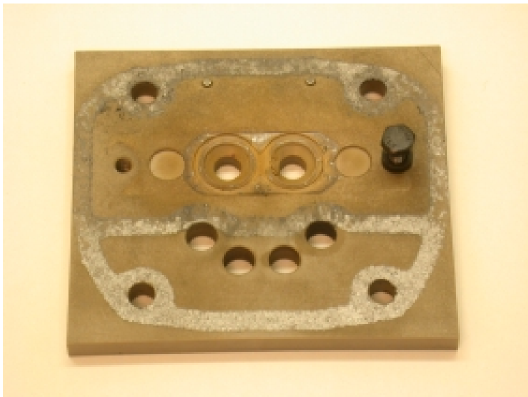

Figure 2: Valve plate made of **G**lass fibre **R**einforced **P**lastic (GRP)

1. Thickness of the reeds
2. Cylinder clearance
3. Material of the valve plate

Point 1 and 2 are described in previous articles [6, 7], the whole results are summarized and discussed in sections 3 and 4. In Point 3, glass fibre reinforced plastic (GRP) valve plates instead of steel plates are used in the tests. GRP is a new material which was not used in compressor valve technology before. Because of its good vibration damping some tests were done to lower the dominant (first) natural frequency. See the two pictures of a steel and a GRP valve plate in Figures 1 and 2.

3 Results

3.1 Variation of reed thickness

Reducing the thickness of the reeds – 0,3 mm thick reeds of steel material are normally used, we tested additionally reeds with 0,2 and 0,5 mm – leads to significant lower sound power levels L_P. Reeds with a thickness of more than 0,3 mm cause more sound power. This applies to both cases (with and without suction tube). The strongest effect of lower sound levels appears at ambient pressure (0 bar). An example plot at 5 bar shows Figure 3, for detailed results please use Table 1.

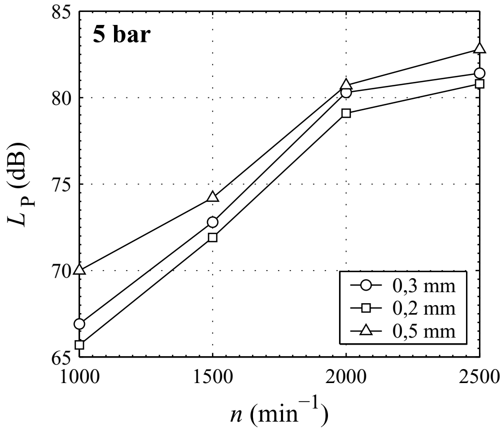

Figure 3: Compressor with mounted suction tube: Sound power levels L_P in dependence of rotational speed n and thickness d of the reed, at 5 bar

3.2 Variation of cylinder clearance

All test results show clearly that increasing cylinder clearance brings up lower sound power levels. Table 1 shows that more cylinder clearance, starting at normal clearance **N** and increasing to **S1**, **S2** and finally to the maximum clearance **S3** leads to considerably lower sound power levels. Complete measurement results are presented in Table 1, an example plot at 5 bar is presented in Figure 4.

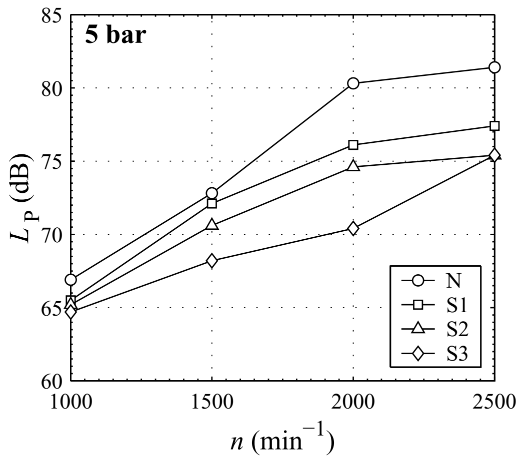

Figure 4: Compressor with mounted suction tube: Sound power levels L_P in dependence of rotational speed n and cylinder clearance, at 5 bar

3.3 Variation of valve plate material

One result of the research work on sound levels of reciprocating compressors was that test bed measurements show a markable peak in the sound power spectrum at about 2000–2300 Hz. This peak is the first natural frequency of the valve retainer. So the idea was to try materials with higher damping ability. As a consequence tests with valve plates out of glass fibre reinforced plastic (GRP) instead of conventionally used steel plates were done. These test bed measurements demonstrate lower sound power levels for both measurement cases (with or without suction tube) and strongest effects using dB(A) frequency weighting. See Figure 5.

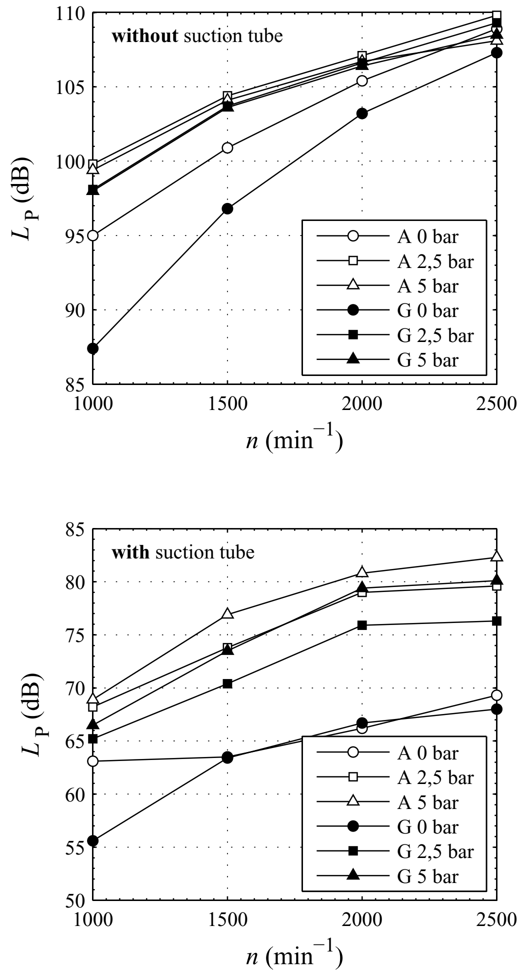

Figure 5: Compressor without/with mounted suction tube: Sound power levels L_P in dependence of rotational speed n, material of the valve plate (A ... steel, G ... GRP) and pressure p

The reason can be easily explained by using the sound power frequency spectrum. This is calculated by a self developed MatLab-programm out of the measuring results (Figure 6). Valve A (steel plate) shows a clear frequency peak at 2200 Hz, valve G (GRP plate) do not. So the difference between the sound power levels of the different valves (ΔL_P) leads to markable lower sound levels using the new GRP material.

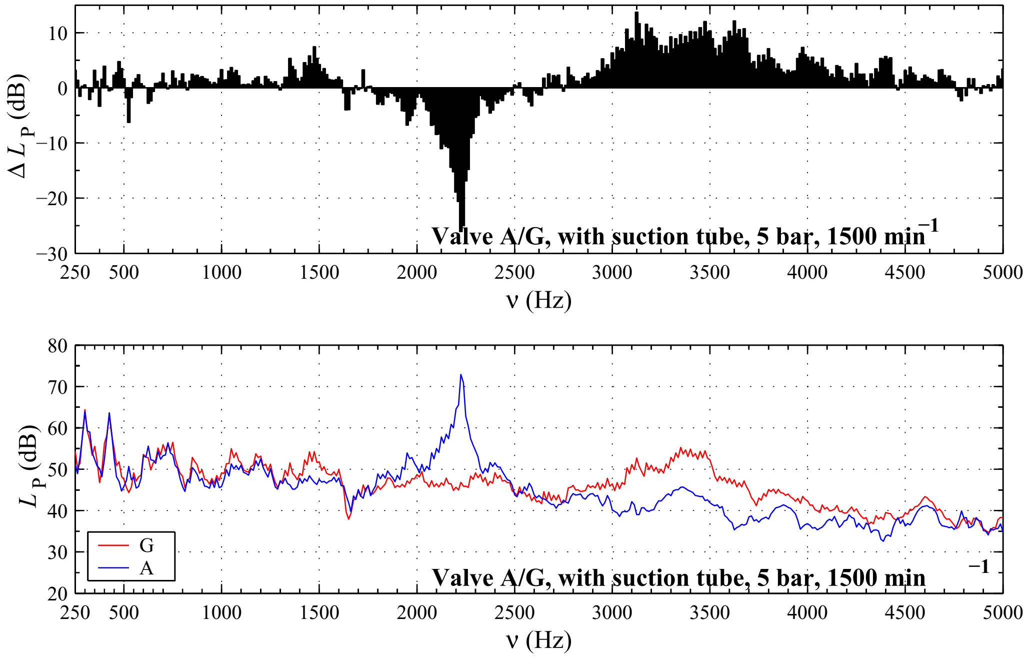

Figure 6: Comparison of sound power spectra of valve A (steel plate) and valve G (GRP-plate)

3.4 Comparison of the results

The following Table 1 shows the whole results of the variation of the three constructional valve parameters: Thickness of the reeds, cylinder clearance and material of the valve plate.

3.4.1 Reduced thickness of the reeds

The last section of Table 1 gives information about the resulting sound power levels when the thickness of the reeds is varied. Column 2 represents the sound power levels (measured with suction tube) if reeds with 0,2 mm thickness are used instead of 0,3 mm thick reeds. Column 4 shows the same measurement case but compares reeds with 0,2 mm thickness to 0,5 mm thick reeds.

As described in 3.4.2 the results are given for ambient pressure and $\geq$ 5 bar. At 0 bar the reductions of the sound power levels are −2,6/−6,3 dB, at $\geq$ 5 bar the reductions are −0,8/−2,7 dB.

3.4.2 Increased cylinder clearance

In the second section of Table 1 the influence of the cylinder clearance is noticeable. At 0 bar the cylinder clearances **S1/S2/S3** show a reduction of sound power levels of −1,5/−0,8/−0,8 dB in comparison to the normally used cylinder clearance **N**. Higher pressures lead to higher sound power reduction. So at $\geq$ 5 bar the sound power levels are −2,6/−3,2/−5,5 dB.

3.4.3 GRP-material of the valve plate

The first section of Table 1 presents the different sound power levels when using glass fibre reinforced plastic as material of the valve plate instead of conventional steel valve plates. **A** stands for the steel plate, **G** for the GRP-plate. Pressure range is between 0 bar and 5 bar. The maximum of 5 bar is limited by the GRP material. Both cases are measured (with and without suction tube) and the results are given in dB and dB(A).

The practically relevant measurement values are the case with suction tube (as used in the busses or trucks) and with A-frequency weight (dB(A)) because the human aural sense feels similar. Therefore at 0/2,5/5 bar pressure the reduction of the measured sound power levels are −1,8/−4,5/−3,2 dB(A).

4 Conclusions

Test bed measurements were done varying three constructional valve parameters. These are the thickness of the reeds, cylinder clearance and the material of the valve plate.

All three methods show sound power reductions of about 3 dB or more. Unfortunately the methods are not free of problems. The thinnest reeds (0,2 mm), do not resist the stress at high rotational speeds and pressures and therefore this heavy load leads to cracks and erosions after very few stress cycles. See Figure 7.

Increasing the cylinder clearance significantly reduces sound levels but also effect the volumetric efficiency of the compressor negatively. Thus the cylinder clearance could not be increased too much, especially if high pressures are occurring.

The last method to decrease the sound levels considerably is to use other plate materials instead of steel – for example – glass fibre reinforced plastic (GRP). However, the lifetime of the valve plate, in particular the sealing boarders of the outlet valves, is very low. So for an industrial usage better materials must be applied. In Figure 8 the damage of the sealing boarders are demonstrated.

$\Delta\overline{L_P}^a$ (dB), **using a valve plate made of GRP**

Pressure (bar)		**A** ⇔ **G** (dB) without ST[b]	**A** ⇔ **G** (dB) with ST	**A** ⇔ **G** (dB(A)) without ST	**A** ⇔ **G** (dB(A)) with ST
0	$\Delta\overline{L_P}$	**−3,9**	**−2,1**	**−4,2**	**−1,8**
	Min.	−7,6	−7,5	−8,7	−5,9
	Max.	−1,6	+0,5	−1,5	+0,4
2,5	$\Delta\overline{L_P}$	**−0,9**	**−3,2**	**−0,9**	**−4,5**
	Min.	−1,7	−3,4	−1,9	−5,4
	Max.	−0,5	−3,0	−0,4	−3,8
5	$\Delta\overline{L_P}$	−0,4	**−2,4**	−0,4	**−3,2**
	Min.	−1,4	−3,4	−1,8	−4,4
	Max.	+0,4	−1,4	+0,7	−2,0

$\Delta\overline{L_P}$ (dB), **if cylinder clearence** S **is increased from N to S1, S2 and S3**

Pressure (bar)		**N** ⇔ **S1** − $A_S = 100\,\%$ without ST	**N** ⇔ **S1** − $A_S = 100\,\%$ with ST	**N** ⇔ **S2** − $A_S = 100\,\%$ without ST	**N** ⇔ **S2** − $A_S = 100\,\%$ with ST	**N** ⇔ **S3** − $A_S = 100\,\%$ without ST	**N** ⇔ **S3** − $A_S = 100\,\%$ with ST
0	$\Delta\overline{L_P}$	±0,0	**−1,5**	−0,5	**−0,8**	**−3,3**	**−0,8**
	Min.	−0,2	−3,6	−0,8	−3,2	−4,5	−4,2
	Max.	+0,5	+1,5	+0,5	+3,6	−2,4	+5,3
≥ 5	$\Delta\overline{L_P}$	**−4,0**	**−2,6**	**−5,0**	**−3,2**	**−6,6**	**−5,5**
	Min.	−7,9	−4,2	−8,0	−6,0	−9,2	−9,9
	Max.	−0,9	−0,4	−1,2	−0,4	−3,4	−2,2

$\Delta\overline{L_P}$ (dB), **if** d^c **is reduced to 0,2 mm**

Pressure (bar)		**0,2** ⇔ **0,3** − $A_S = 100\,\%$ without ST	**0,2** ⇔ **0,3** − $A_S = 100\,\%$ with ST	**0,2** ⇔ **0,5** − $A_S = 100\,\%$ without ST	**0,2** ⇔ **0,5** − $A_S = 100\,\%$ with ST
0	$\Delta\overline{L_P}$	**−5,4**	**−2,6**	**−4,9**	**−6,3**
	Min.	−9,5	−5,0	−8,8	−8,4
	Max.	−2,1	−1,5	−0,1	−1,7
≥ 5	$\Delta\overline{L_P}$	**−1,3**	**−0,8**	−0,7	**−2,7**
	Min.	−2,8	−1,9	−1,8	−4,3
	Max.	−0,3	+0,9	+0,2	−1,3

[a] $\Delta\overline{L_P}$ . . . average deviation of sound power levels of different valve types
[b] ST . . . suction tube
[c] d . . . thickness of the reeds

Table 1: Summary of the measurement results of sound power levels (rotational speed between 1000–2500 min^{-1}). Representative measurement values (out of the measurement inaccuracy of ±0,7 dB) are written in bold.

Figure 7: Damage (Cracks) of the two 0,2 mm thin reeds

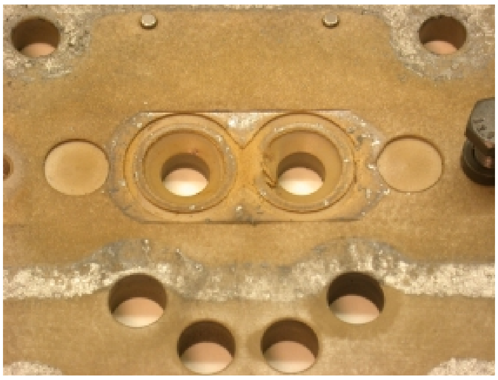

Figure 8: Damage on the sealing boarders of the GRP-valve plate

Summarizing the described methods show some new approaches for research work, but the lifetimes of the materials (valve plate and reeds) are far too short. So other materials should be tested and rated for useability. Increasing cylindrical clearance reduces the volumetric efficiency of the compressor obvious and is therefore only of marginal interest.

Test bed measurements with other materials will go on and new results will be presented at one of the following Acoustics conferences.

Acknowledgments

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