XVII IMEKO World Congress
Metrology in the 3rd Millennium
June 22–27, 2003, Dubrovnik, Croatia

DETERMINATION OF SOUND CAUSED BY COMPRESSOR VALVES USING TIME SIGNAL ANALYSIS

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Abstract – The measurement of sound levels of noise sources located very closely together – for example the inlet and outlet valve of a piston compressor – is a difficult task. This paper describes a method to evaluate the emitted noise levels of each valve. To achieve precise results, an analysis of the sound pressure level depending on time (time-signal) was done. Signal processing and computations of the measurement results lead to clearer separation of the sound emitted by each sound source (valve).

Keywords: noise measurement, time signal, sound intensity.

1. INTRODUCTION

State of the art measurements using the sound pressure level L_p have the clear disadvantage that the radiation of acoustic sources, which are very close together, cannot be measured separately. Sound power measurement [1, 2] by using "direct" intensity method delivers significantly more information about the distribution of the sound field but requires very extensive instrument resources. Besides, increasing the resolution (means more measurement points) is limited by the time needed for preparation and measuring. Also it is not possible to separate the two sound sources completely.

Because of the above described reasons another method is used for to solve this specific acoustic problem. In this case – the sound propagation of a reciprocating compressor [3, 4] and herein the sound emission of inlet and outlet valve – we benefit from the fact that the noise signal occurs time-shifted. Thus the *time signals* of the sound pressure p referring to $p_0 = 2{\cdot}10^{-5}$ N/m^2 is used for further analysis.

2. TEST BENCH AND MEASUREMENTS

Fig. 1 describes the test bench for measuring the time dependent signals of *sound pressure*, *elevation of each valve lamella* (opening and closing time of the valve) and *accelerations* on the cylindrical head. The compressor (1) is mounted on a stiff plate (with very low sound absorption coefficient), and is driven by an electric motor (2) surrounded by a soundproof housing (3). To prevent noise interferences and sound reflections the whole test bench is situated in a sound-absorbing testing room [5]. The oil pump needed for lubricating the compressor and the pressure vessel are located outside for the same reasons.

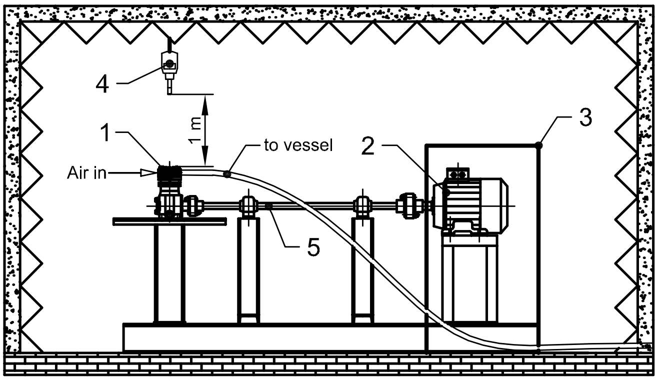

Fig. 1 Test bench for time signal measurements, situated in a sound-absorbing testing room

2.1. Measurements

The time signals of the sound pressure are measured with a precision sound-level meter (4) in a representative point in a distance of 1 meter above the cylindrical head of the compressor. Instead of the time in milliseconds we use as an equivalent the angle φ of the rotating crankshaft (x-axes in the charts). To obtain a reference for the position of the piston and to get the rotation speed of the compressor, a reflecting mark is attached to the surface of the drive shaft (5) and combined with an optical sensor. When the piston reaches its upper position φ is defined as 0 degrees, a complete revolution is divided into 360 degrees.

Accelerations are measured with an accelerometer to get an impression of the structure borne noise caused by the opening/closing impact of the valves. For test bench control (rotation speed), measurement (air-/oil-pressure, air-/oil-temperature) and the whole data processing are done using the measurement and automation software LabVIEW.

These measurements are done for two different operating conditions:

a) *with* cylindrical head and *opened* inlet port
b) *with* cylindrical head but with suction tube (*closed* inlet port)

Two optical sensors detect the opening and closing of the valves by measuring the elevation of each valve lamella [6, 7]. Such a typical chart for one complete revolution (=360°) is shown in Fig. 4.

Case a) shows primary the airborne noise [8] emitted by the air flow through the inlet valve/port and the influence of the flow rate and inlet valve parameters. Case b), which describes the more practise-oriented case, is a measure for structure borne noise [9], because the air flow is mainly absorbed by the suction tube connected to the inlet port.

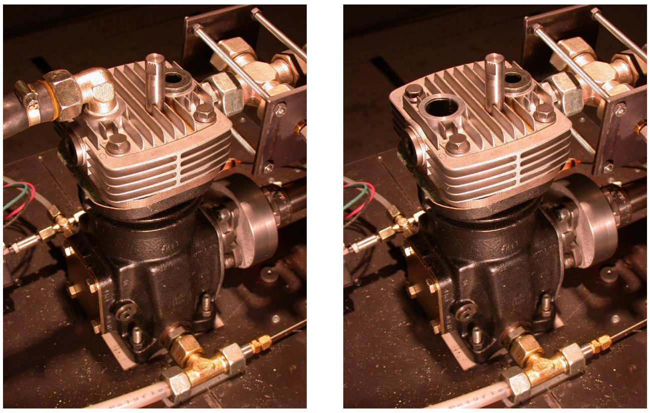

Fig. 2. Compressor with (left figure, case b) and without suction tube (right figure, case a)

Therefore the differentiation into two cases tries to demonstrate the very complex connection between airborne and structure borne noise in a more comprehensible way.

3. MEASUREMENT RESULTS AND EVALUATION

As mentioned in subsection 2.1 the measurements are done for two different operating conditions.

3.1. Operating conditions without suction tube

Fig. 3 shows an example of a measured time signal of the sound pressure p/p_0 with open inlet port. In accordance to Fig. 4 and Fig. 5 one can see, that the noise caused by the inlet valve starts exactly when the inlet valve opens ($\varphi \sim 50°$). Moreover the amplitude of the time signal decreases while the inlet valve closes.

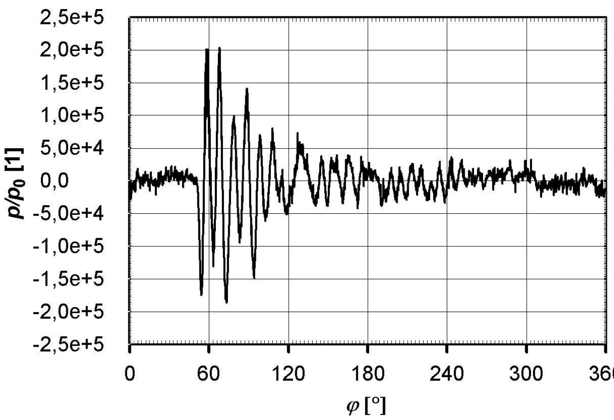

Fig. 3. *Compressor without suction tube:* Time signal p/p_0 of the emitted sound pressure, at 1000 min^{-1} and 13 bar

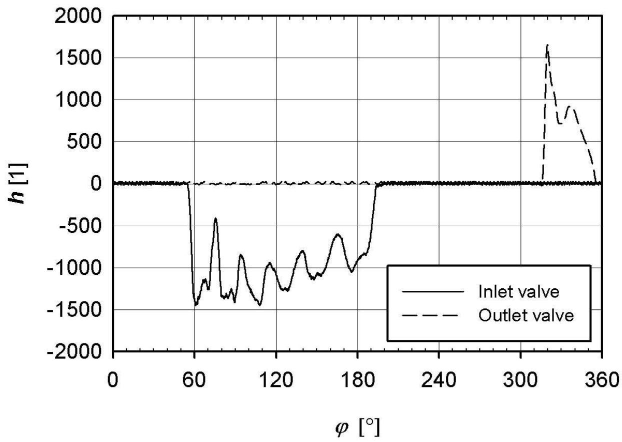

Fig. 4. Elevation *h* of the valve lamellas of inlet and outlet valve, depending on the angle of rotation φ

The noise emitted by the outlet valve cannot be seen so clearly in the chart of the time signal p/p_0. This is because of the very dominant "airborne" sound level radiated by the inlet valve (for the case with opened inlet port). Thus, the smaller noise level – caused by the outlet valve – mainly "structure borne" noise is not visible in the time signal, but can be seen very well in Fig. 5.

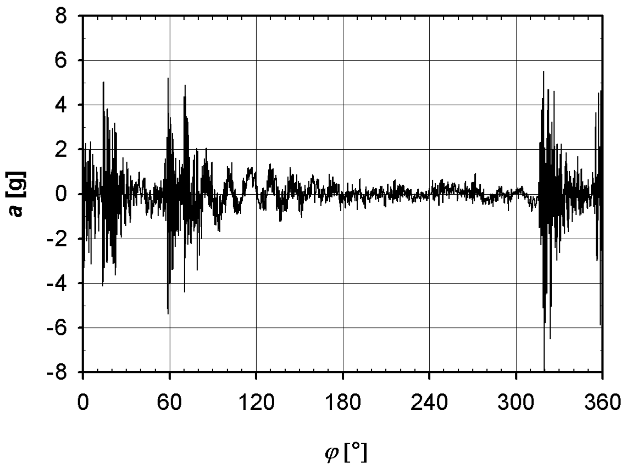

Fig. 5. *Compressor without suction tube:* Acceleration signal measured on a representative point on the cylindrical head of the compressor

This figure shows the acceleration a in dependence of the angle of rotation φ measured at a representative point (see the exact position of the accelerometer in Fig. 6) on the cylindrical head, above the outlet valve.

The opening of the inlet valve can be seen very clearly, it starts opening at 50° and the amplitude of the acceleration signal increases. From that point on the resulting oscillation decreases (however much slower compared to the increase). At approximately 190° to 200° the inlet valve closes and the amplitude of the signal remains nearly constant. There is no "closing impact" of the inlet valve lamella noticeable in the diagram.

At 320° a dominant peak in the acceleration line shows the opening of the outlet valve. The lamella strikes against the limitation bracket and generates vibrations in the whole cylindrical head of the compressor with a rapidly increasing amplitude. Very close to the upper dead-centre (near 360°) the outlet valve closes and again the lamella strikes against the valve plate. Once more vibrations in the system result and can be seen in the chart (Fig. 5). The oscillating signal lasts till ~40°.

These characteristic peaks as described in the diagram are very similar to the elevation curves of input and output valves in Fig. 4.

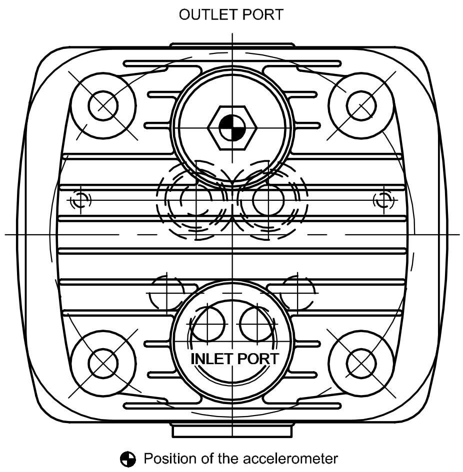

Fig. 6. Position of the accelerometer on the cylindrical head of the compressor; opened inlet port

3.2. Operating conditions with suction tube

The time signal for this case shows different results compared to the first case. Firstly the amplitude of the sound pressure (noise level) is smaller because of the drastically reduced airborne noise (suction tube!). Secondly the lower pressure signal p/p_0 shows more signal noise and interferences. The emitted noise of the inlet valve is not so dominating and therefore opening and closing of the inlet (and outlet) valve cannot be clearly identified. Fig. 7 shows only more activity within the range of the opening of the inlet valve (~60°) and the opening of the outlet valve (~320°). Thus, in this case one is not able to readout the characteristic points (described in subsection 3.1) with the same accuracy.

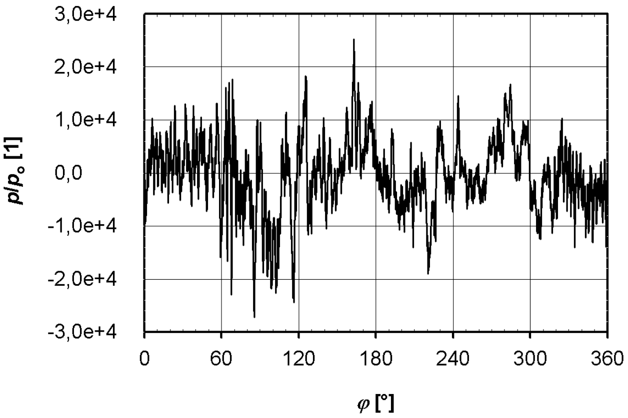

Fig. 7. *Compressor without suction tube:* Time signal p/p_0 of the emitted sound pressure, at 1000 min^{-1} and 13 bar

4. CALCULATIONS

With the results of the time signals, calculations of the noise levels were done to establish the share of each valve in the whole noise emission. To figure this out, the sound levels (time signals p/p_0 of the noise pressure) [10] are computed for different periods.

4.1. Noise level calculations out of the time signals

The calculations of the time signals are done for two major sections. First section t_{I} is the time period (or an equal angle of the crankshaft φ_{I}) from opening to closing of the inlet valve. A subtraction of this period from one complete period t_{P} (0°–360°) leads to the second section t_{R}. So out of Fig. 4 the period t_{I} ranges from 60° to 200° and t_{R} contains 0°–60° and 200°–360°.

Over these two fields the noise levels are calculated out of the time signal and this leads to two sound levels: L_{I} (φ_{I}) and L_{R} (φ_{R}). A factor L_{D} is defined as

$$L_{\mathrm{D}} = L_{\mathrm{I}} - L_{\mathrm{R}} \tag{1}$$

and represents a measure for the "sound activity" of the inlet valve compared to the sound activity of the outlet valve and noise caused by the "mechanical" load (pressure).

4.2. Operating conditions without suction tube

The next diagram (Fig. 8) shows – for example – the behaviour of the factor L_{D}, as function of pressure in the vessel p_{v} and rotation speed n. Additionally the chart shows how dominating the sound propagation of the (open inlet port!) inlet valve is.

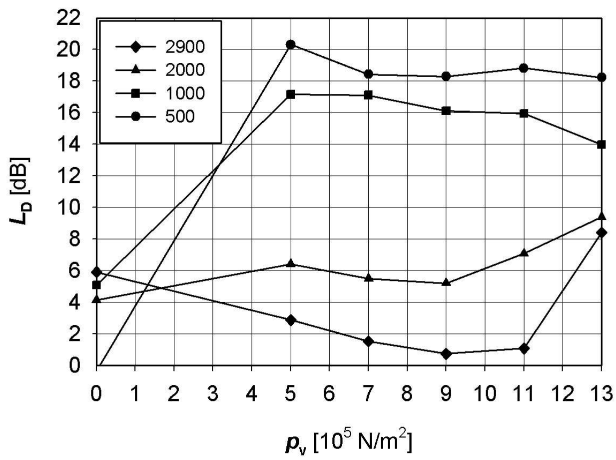

Fig. 8. *Compressor without suction tube:* Calculated level differences L_{D}, for different rotation speed (500 min^{-1} to 2900 min^{-1}) and pressures in the vessel (0 to 13 bar)

At low rotation speed (500 min^{-1} and 1000 min^{-1}) Fig. 8 shows a significant influence of airborne noise in contrast to higher speed (2000 min^{-1} and 2900 min^{-1}), where the structure borne noise gains in relevance. For pressures p_{v} above 5 bar, (up to 13 bar) the accumulated noise levels within the period of opened inlet valve (φ_{I}) differ up to 13 dB – at 500 min^{-1} and 1000 min^{-1}. In addition the chart shows clearly that the sound emitted by the inlet valve dominates for all operating conditions (p_v and n) – L_D is always greater than 0. With increasing rotation speed L_D gets lower because the noise caused by higher mechanical load increases. This results in disproportionate higher L_R (increasing noise of the outlet valve!) and thus lower L_D.

In this way one gets essential hints in which part of the construction modifications will have the most influence to reduce noise radiation.

4.3. Operating conditions with suction tube

The next diagram (Fig. 9) shows a completely different situation for this operating condition. Calculations show that the factor L_D varies only between –2 dB and 3 dB, instead of 0 dB to a maximum of 20 dB for case a). Furthermore L_D is nearly independent of rotation speed n (within the range of 500 min^{-1} to 2900 min^{-1}) and pressure p_v in the vessel (0 bar–13 bar). Therefore the mean value out of the sound level differences L_D, for all pressures and speed yields in approximately 0,75 dB.

Summarizing one can say, that without a dominant noise source (like the opened inlet valve in case a), which causes very high airborne noise levels) the described method delivers relatively few information. Both valves emit noise levels in the same order of magnitude and this leads to very small level differences L_D.

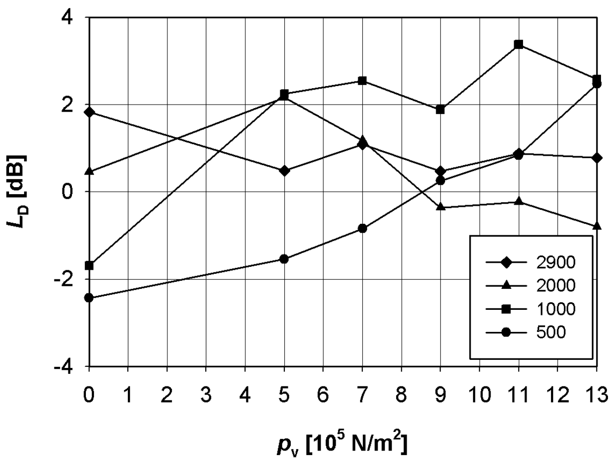

Fig. 9. *Compressor with suction tube:* Calculated level differences L_D, for different rotation speed (500 min^{-1} to 2900 min^{-1}) and pressures in the vessel (0 to 13 bar)

4. SUMMARY

This paper describes a measurement method suitable for the determination of noise sources, showing time-shifted noise signals. Sound emissions of inlet and outlet valve of a reciprocating compressors are possible fields of application for this method.

The described method uses time signals measurements of the sound pressure. Its behaviour considering other measurements (such as the elevations of the valve lamellas) is discussed in detail and sound level calculations are done using the time signals. Applied signal analysis makes it possible to investigate the influence of many additional parameters. For future research such factors can be the quantity of air flow, several parameters of the valves, (for example size of the valve port and design of the valve lamella) as well as experiments with other materials of cylindrical head of the compressor, valve plate and valve lamellas.

Finally this paper deals with computations of the time signals for different periods to try to separate the noise of inlet and outlet valve. These results are presented and could be used as an approach to a better understanding of the reasons causing the noise emissions of machines [11, 12], and especially of reciprocating compressors.

5 FUTURE STEPS

As mentioned in section 4, further measurements of sound pressure time signals and examinations of these signals using signal analysis will be done to evaluate the influences of the valve parameters. These are the cross-section of the inlet port (and thus the quantity of air flow), thickness and elevation of the lamellas and the design of the limitation bracket/s. Further results will be presented at one of the forthcoming conferences on acoustics.

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ACKNOWLEDGEMENTS: This research work was kindly supported by **Hoerbiger Ventilwerke,** Schongau.