Experimental Investigation of a Mechanically Assisted Suction Reed Valve in a Small Hermetic Reciprocating Compressor

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Abstract. Reed valves are widely used in hermetic reciprocating compressors and are responsible for a major share of thermodynamic losses. In particular, the suction valve movement of a hermetic reciprocating compressor contains a potential for further improvement. The self-actuated valve is opened only by the pressure difference created by the moving piston, which leads to a characteristic flutter. Furthermore, the suction valve is usually preloaded in order to close properly before the compression phase, causing additional flow resistance.

In this work a mechanically assisted suction valve is experimentally investigated. A mechanically actuated spring element generates an additional force acting on the conventional suction reed valve. This force leads to a completely different valve motion. While a conventional reed valve motion is characterized by several opening- and closing periods during one suction phase, the mechanically assisted valve remains open during the whole suction phase. Furthermore, the maximum impact velocity of the valve can be reduced significantly. Thus, not only the suction losses but also the valve stress can be reduced.

To investigate the mechanically assisted suction valve, a conventional small capacity hermetic reciprocating compressor is fitted with measurement equipment and additional components for the mechanical valve actuation. Preliminary investigations are made without the hermetic shell to find an appropriate design. Finally, measurement results of the compressor running at steady-state conditions on a calorimeter test bench (R600a) show significant improvements of the coefficient of performance (COP) at different rotational speeds and operating conditions.

1. Introduction

Reed valves are used in almost all reciprocating compressors. They are characterized by a simple and inexpensive design and by their automatic adaption to different operating conditions. The actuation is caused only by the pressure difference between cylinder and suction or discharge line. Reed valves are usually preloaded to ensure timely closing at the end of the suction or discharge phase. In reciprocating compressors, reed valves are crucial elements in terms of efficiency and reliability. For this reason, great efforts have been made in the past to analyse the relationships between valve dynamics, compressor efficiency and reliability.

In the work of [1] for example, the pV-Diagram as well as the motion of the suction- and discharge valves of a hermetic reciprocating compressor were determined experimentally. The authors used this information to observe, whether or not gas back-flow due to valve closing delays occurred. Also [2] measured the suction valve motion of a reciprocating compressor for a domestic refrigerator, but with a
different measurement approach. They investigated the relation between the suction valve motion and the volumetric efficiency by varying the rotational speed of the compressor. The authors found that the sticking force of the lubricant is more dominant at lower compressor speeds. Furthermore, they observed that the volumetric efficiency varies in relation to the compressor speed, with the oscillation amplitude being higher at lower compressor speeds. This performance variation is due to different closing delays of the suction valve, depending on the compressor speed. Suction reed valves open and close several times during one suction phase. The frequency of this valve flutter is influenced by the effective moving mass, the valve spring rate and the preload and is thus a system constant. However, different compressor speeds lead to different time periods of the suction phase which may cause valve closing delays. The authors of [3] conducted a sensitivity analysis of different suction valve parameters using a 1D compressor model. They emphasized the importance of reduced valve flutter for the reduction of valve losses. Similar findings were obtained by [4]. They investigated the potential of a theoretical valve without the flutter. The suction power consumption decreased by 36 % leading to a COP increase of 2 %. [5] studied the reduction of the suction losses using a suction valve with a negative preload force in combination with an electromagnetic coil. Calorimeter measurements showed a COP improvement potential up to 1.7 %. However, the COP improvement was strongly influenced by the operating conditions and the energy consumption of the electromagnetic coil was not considered in the calculation. Nevertheless, it is another example that illustrates the importance of the suction valve behaviour with respect to the efficiency of the reciprocating compressor.

The aim of this study is to investigate whether a suction valve support mechanism is capable of improving the efficiency of a reciprocating compressor. The principle of the mechanically assisted (m.a.) suction reed valve is based on an additional supporting force which acts on the conventional suction valve, while maintaining the automatic adaption of the valve to different operating conditions. The magnitude of the force must vary so that the valve opens as quickly as possible, remains open during the suction phase and finally closes with a low impact velocity. For this purpose, a mechanism was developed and assembled to a small capacity hermetic reciprocating compressor for cooling appliances (R600a). Preliminary investigations of the suction valve dynamics show a favourable behaviour of the m.a. valve. The mechanism is capable of keeping the valve open during the entire suction process. Furthermore, the maximal impact velocity of the m.a. valve is reduced significantly compared to the standard valve. COP measurements at steady-state conditions according to ASHRAE show a consistently improved performance if a m.a. valve is used.

2. Experimental Approach
The experimental investigation can be divided into two main sections. In the first section a preliminary study was carried out in order to analyse the dynamic behaviour of the suction reed valve under specific

Figure 1. Reciprocating hermetic compressor with suction reed valve support mechanism.
conditions while in the second section, calorimeter measurements at steady-state conditions according to ASHRAE were performed.

The valve support mechanism was assembled on a small hermetic reciprocating compressor. As shown in figure 1, the mechanism basically consists of a grooved disc fixed to the crank shaft, a lever and a spring element attached to the lever. The lever is required to transform the rotational motion of the grooved disc into a translational motion of the spring element. Therefore, a bearing pin at one end of the lever slides in an eccentric circular groove. The eccentric circular groove and the lever were designed to achieve a sinusoidal motion of the spring element with an amplitude of 1.5 mm and a phase shift of 90 °CA ahead of the crank drive. In addition, an offset of the spring element in the direction of the suction valve can be adjusted. Hence, the baseline of the sinusoidal motion can be transferred in the valve lift direction.

2.1. Suction Reed Valve Dynamics

The purpose of this preliminary study is to increase the system knowledge as well as to find an appropriate design configuration for further performance measurements. Therefore, the response of the suction valve to the variation of four different system parameters (spring element stiffness, spring element offset, discharge pressure and compressor speed) is investigated. The tests were carried out without the upper hermetic compressor shell, using air as flow medium. A throttle valve was installed in the discharge line outside the shell to adjust different discharge pressure levels. The suction pressure level was restricted to ambient pressure. All relevant system parameters of the four different variation experiments are listed in table 1 to table 4.

A scheme of the measurement setup is shown in figure 2. Laser Doppler Velocimetry (LDV) was used for valve lift measurements as it is a robust, cost effective and simple method. Furthermore, it enables contactless measurements so that the object is not affected by the measurement device. The laser beam emitted from the LDV sensor head pointed to the surface of the suction valve and detected the valve velocity in the direction of the laser beam. The valve lift was determined by further processing of
the measured velocity according to (1). At this point it must be mentioned, that in the event of unfavourable conditions “drop-outs” may occur during the LDV measurement. This term describes the failure of the demodulation circuit to derive an analogue velocity signal from a Doppler signal of insufficient amplitude, see [6]. A “drop-out” can be observed by a sudden step in the velocity signal. In this measurement application occasional “drop-outs” can hardly be avoided. The measured velocity signal was averaged over five cycles to counteract this effect.

\[
s_{i+1} - s_i = \int_{t_i}^{t_{i+1}} v(t) \, dt \approx \frac{(v_{i+1} - v_i)}{2} \cdot t_s
\]  

(1)

A Kulite F78-17 was installed in the cylinder head to measure the discharge pressure. To monitor the contact behaviour between the spring element and the suction valve, a contact measurement was included into the measurement setup. It is based on a simple electrical circuit which consists of a voltage source and an ohmic resistor. If contact between spring element and suction valve occurs, the circuit is closed and the voltage drop across the resistor can thus be measured. A hall sensor (SS496A1) was used to relate the measurement signals to the crank angle position. It was fixed to the crank case while a neodymium magnet was attached on the outer surface of the grooved disc. Each time the magnet passes the hall sensor, a peak signal can be detected. All measurements were conducted with a sampling rate of 65.5 kHz.

| Suction valve | Spring element stiffness (N/mm) | Spring element offset (mm) | Compressor speed (rpm) | Discharge pressure (MPa) |
|---------------|---------------------------------|---------------------------|------------------------|------------------------|
| standard valve| -                               | -                         | 3000                   | 0.6                    |
| m.a. valve    | 0.5                             | 0.4                       | 3000                   | 0.6                    |
| m.a. valve    | 1.0                             | 0.4                       | 3000                   | 0.6                    |
| m.a. valve    | 1.5                             | 0.4                       | 3000                   | 0.6                    |
| m.a. valve    | 2.9                             | 0.4                       | 3000                   | 0.6                    |

| Suction valve | Spring element stiffness (N/mm) | Spring element offset (mm) | Compressor speed (rpm) | Discharge pressure (MPa) |
|---------------|---------------------------------|---------------------------|------------------------|------------------------|
| standard valve| -                               | -                         | 3000                   | 0.6                    |
| m.a. valve    | 1.0                             | 0.4                       | 3000                   | 0.6                    |
| m.a. valve    | 1.0                             | 0.8                       | 3000                   | 0.6                    |
| m.a. valve    | 1.0                             | 1.2                       | 3000                   | 0.6                    |
Table 3. Test conditions of the discharge pressure variation experiment.

| Suction valve | Spring element stiffness (N/mm) | Spring element offset (mm) | Compressor speed (rpm) | Discharge pressure (MPa) |
|---------------|---------------------------------|----------------------------|------------------------|-------------------------|
| standard valve| -                               | -                          | 3000                   | 0.3                     |
| standard valve| -                               | -                          | 3000                   | 0.5                     |
| standard valve| -                               | -                          | 3000                   | 0.7                     |
| m.a. valve    | 1.0                             | 0.4                        | 3000                   | 0.3                     |
| m.a. valve    | 1.0                             | 0.4                        | 3000                   | 0.5                     |
| m.a. valve    | 1.0                             | 0.4                        | 3000                   | 0.7                     |

Table 4. Test conditions of the compressor speed variation experiment.

| Suction valve | Spring element stiffness (N/mm) | Spring element offset (mm) | Compressor speed (rpm) | Discharge pressure (MPa) |
|---------------|---------------------------------|----------------------------|------------------------|-------------------------|
| standard valve| -                               | -                          | 1000                   | 0.6                     |
| standard valve| -                               | -                          | 3000                   | 0.6                     |
| standard valve| -                               | -                          | 4000                   | 0.6                     |
| m.a. valve    | 1.0                             | 0.4                        | 1000                   | 0.6                     |
| m.a. valve    | 1.0                             | 0.4                        | 3000                   | 0.6                     |
| m.a. valve    | 1.0                             | 0.4                        | 4000                   | 0.6                     |

2.2. Calorimeter measurements

Calorimeter measurements at steady-state conditions according to ASHRAE were performed in order to determine the performance of the valve support mechanism. A full list of the test conditions is given in table 5. A spring stiffness of 1.0 N/mm and a spring offset of 0.4 mm were selected for these performance measurements based on the results of the parameter variation experiment. Two different combinations between the evaporation and condensation temperature levels were considered. The ambient temperature was set to 32 °C in all measurements. The compressor speed was varied between 1500 rpm and 4000 rpm. Each measurement at a given operating condition was performed with the m.a. valve and with the standard valve.

Table 5. Test conditions of performance measurements.

| Compressor speed (rpm) | Evaporation temperature (°C) | Condensation temperature (°C) |
|------------------------|------------------------------|-------------------------------|
| 1500                   | -23.3                        | 45                            |
| 2000                   | -23.3                        | 45                            |
| 3000                   | -23.3                        | 45                            |
| 4000                   | -23.3                        | 45                            |
| 1500                   | -10                          | 45                            |
| 2000                   | -10                          | 45                            |
| 3000                   | -10                          | 45                            |
| 4000                   | -10                          | 45                            |
3. Results
The first part of this chapter outlines the results of the preliminary investigation of the suction valve dynamics while the second part gives the results of the calorimeter measurements.

3.1. Variation of the spring stiffness
A total of four different spring elements with a stiffness of 0.5 N/mm, 1.0 N/mm, 1.5 N/mm and 2.9 N/mm were tested. The spring offset, the compressor speed and the discharge pressure remained constant. Figure 3 shows the motion of the standard and the m.a. suction valve at different spring element stiffnesses. Regardless of the spring element used, the valve motion of the m.a. suction valve is fundamentally different in comparison to the standard valve. In general, the m.a. valve opens a few degrees earlier than the standard valve, remains open and finally closes gently after the bottom dead centre (bdc). The higher the stiffness of the spring element, the steeper the valve opening curve. The maximal valve lift of the m.a. valve is significantly lower compared to the standard valve. A spring stiffness of 2.9 N/mm is probably too high and may cause the suction valve to open before pressure equalisation between cylinder and suction line has occurred. However, this cannot be verified with the underlying measurement setup.

The small offsets of the valve lift curves at the end of the suction phase are due to occasional “drop-outs” of the LDV measurement as mentioned in chapter 2. These sudden “drop-outs” of the velocity measurement typically occurred within a time range of a few measurement samples (sampling rate: 65.5 kHz). Since the valve lift curve is obtained from (1), a small offset is added over the suction phase. Nevertheless, the valve closing point is determined by the position where the valve lift continues with a horizontal line.

![Figure 3. Motion of standard and m.a. valve with spring element stiffness variation. Δx_s = 0.4 mm, n_comp = 3000 rpm and p_dis = 0.6 MPa. Preliminary investigation without hermetic compressor shell and with air as flow medium.](image)

3.2. Variation of the discharge pressure
The spring element with a stiffness of 1.0 N/mm was selected for further investigation. Figure 4. shows the valve motion and the valve velocity of the standard and the m.a. valve at different discharge pressure levels. In general, rising discharge pressure levels lead to later valve openings due to an enlarged re-expansion. In addition, larger valve lifts occur, as the piston speed and thus the time gradient of the cylinder pressure increases up to a crank angle of approx. 78 °CA. The functionality of the m.a. valve is maintained over the different discharge pressure levels. Comparing c) and d) one major advantage of the m.a. valve is clearly visible. By using a valve support mechanism, the maximal valve impact velocity can be considerably reduced.
Figure 4. Motion of m.a. valve a) and standard valve b) at different discharge pressure levels. The corresponding velocities of the m.a. valve c) and the standard valve d). $c_s = 1.0 \text{ N/mm}$, $\Delta x_s = 0.4 \text{ mm}$ and $n_{\text{comp}} = 3000 \text{ rpm}$. Preliminary investigation without hermetic compressor shell and with air as flow medium.

3.3. Variation of the spring element offset

Figure 5 shows valve motion and contact state caused by different spring element offsets. The displacement of the spring element is linked to a shift of the sinusoidal excitation, which results in an earlier contact point, extended contact periods and higher supporting forces. The higher the offset, the earlier and faster the valve opens. An offset of 1.2 mm even leads to a collision between the suction valve and the piston.

Figure 5. Motion and contact state of standard and m.a. valve with spring element offset variations. $c_s = 1.0 \text{ N/mm}$, $n_{\text{comp}} = 3000$ and $p_{\text{dis}} = 0.6 \text{ MPa}$. Preliminary investigation without hermetic compressor shell and with air as flow medium.
3.4. Variation of compressor speed
A comparison of the suction valve motion and velocity at different compressor speeds is given in figure 6. While the valve motion of the standard valve differs strongly at different compressor speeds, the m.a. valve shows a rather uniform behaviour. In general, a higher compressor speed leads to larger valve lifts and higher impact velocities. At 4000 rpm the maximal valve impact velocity of the standard valve is about 5 m/s. However, in case of the m.a. valve it drops to about 1.5 m/s.

![Figure 6](image)

Figure 6. Motion of m.a. valve a) and the standard valve b) at different compressor speeds. The corresponding velocities of the m.a. valve c) and the standard valve d). c_s = 1.0 N/mm, Δx_s = 0.4 mm and p_{dis} = 0.6 MPa. Preliminary investigation without hermetic compressor shell and with air as flow medium.

3.5. Calorimeter measurements
Figure 7. shows the measured COP related to the compressor speed at two different combinations between the evaporation and condensation temperature. In both cases, the highest COP was obtained at 3000 rpm. The m.a. valve permanently leads to a higher COP than the standard valve. Table 6 gives the relative improvement of the COP in % according to (2). On average, the COP improvement of the m.a. valve over all measurements is approx. 1.9 %.

\[
\Delta \text{COP}_{\text{rel}} = 100 \times \frac{\text{COP}_{\text{m.a. valve}} - \text{COP}_{\text{standard valve}}}{\text{COP}_{\text{standard valve}}} 
\]  

(2)
Figure 7. COP at different compressor speeds of standard and m.a. suction valve with spring element stiffness of 1.0 N/mm and spring element offset of 0.4 mm. Test conditions according to ASHRAE, -23.3 °C evaporation temperature and 45 °C condensation temperature in a) and -10 °C evaporation temperature and 45 °C condensation temperature in b), 32 °C ambient temperature respectively, R600a.

Table 6. Relative COP improvement related to the compressor speed at two different temperature conditions

| Compressor speed (rpm) | ΔCOP_{rel} at -23.3 °C / 45 °C (%) | ΔCOP_{rel} at -10 °C / 45 °C (%) |
|------------------------|-----------------------------------|----------------------------------|
| 4000                   | 1.93                              | 2.35                             |
| 3000                   | 1.76                              | 1.05                             |
| 2000                   | 2.10                              | 2.03                             |
| 1500                   | 1.98                              | 1.91                             |

4. Conclusion
A suction reed valve support mechanism to improve the valve dynamics and finally the compressor efficiency was experimentally investigated in this study. The concept is based on a variable supporting force, generated by a simple moving spring element, which acts on the surface of a conventional suction valve. The magnitude of such a variable force must be within a range that maintains the automatic adaption to different operating conditions of the valve. The motion of the spring element is produced by a lever and a grooved disc which is fixed to the crank shaft. Preliminary examinations of the mechanism at different system conditions exposed a favourable behaviour of the assisted valve. Compared to the standard valve, the m.a. valve opens faster, remains open during the suction phase and finally closes gently. Although the preliminary test was conducted with ambient air as flow medium, calorimeter measurements with R600a under standardized test conditions confirmed the positive influence of the valve support mechanism. Compared to the use of a standard valve, an average COP improvement of approx. 1.9% was achieved among a variety of different operating conditions.

The current trend in refrigeration technology toward variable speed compressors is particularly challenging for the valve dynamics. The design possibilities of today's reed valves are strongly limited by the valve flutter and the associated reliability as well as the opening and closing delays. The use of a valve support mechanism offers totally new possibilities in valve design. Current work is focused on design simplification of the mechanism which is more suitable for serial application. The final design is obtained by applying a design of experiment approach.
Nomenclature

\( c_s \)  
spring element stiffness

\( n_{comp} \)  
compressor speed

\( p_{dis} \)  
discharge pressure

\( R \)  
electric resistance

\( s_i \)  
suction valve lift at sampling time \( i \)

\( t_s \)  
sampling time

\( v_i \)  
suction valve velocity at sampling time \( i \)

\( \Delta x_s \)  
spring element offset

Abbreviations

\( \text{bdc} \)  
bottom dead centre

\( \text{CA} \)  
crank angle

\( \text{COP} \)  
coefficient of performance

\( \Delta \text{COP}_{rel} \)  
relative improvement of coefficient of performance

\( \text{DC} \)  
direct current

\( \text{LDV} \)  
Laser Doppler vibrometer

\( \text{m.a.} \)  
mechanically assisted

\( \text{rpm} \)  
revolutions per minute

\( \text{tdc} \)  
top dead centre

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