Force assisted discharge valve for piston compressors

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Abstract. Due to the enormous number of domestic refrigeration appliances worldwide and the associated high overall energy demand of the cooling sector, manufacturers are required to develop more efficient cooling devices. For this reason, variable speed compressors are increasingly being used in order to better match refrigeration demand and supplied cooling capacity. However, variable speed operation is associated with challenges concerning the valve dynamics. Valve oscillation as well as opening and closing delays reduce the energy efficiency (COP) of the compressor.

The underlying study is about a simulation based potential analysis of a novel force assisted discharge valve to improve the valve dynamics of a small hermetic reciprocating compressor for domestic refrigeration. Discharge valve and valve support mechanism are modelled by a 1d-spring-damper-mass multi-body system and implemented in an existing in-house compressor simulation model.

Systematic methods of statistical design of experiments (DOE) are applied not only to improve the COP but also to generate a robust valve behaviour which is largely independent of the operating conditions of the compressor. The findings of the force assisted discharge valve are basically transferable to any other piston compressor with flutter valves.

1. Introduction

In reciprocating compressors for domestic refrigeration appliances reed valves are widely used to control the in- and outflow of the cylinder. The advantages of such valves are their simple and inexpensive design as well as their capability to automatically adapt to different operating conditions as the actuation is caused by the pressure difference between cylinder and suction or discharge line. However, reed valves are crucial components in terms of efficiency, cooling capacity, acoustics and reliability of the compressor.

The oscillation frequency of the valve must be matched to the compressor speed to ensure correct timing of the opening and closing process. This is well manageable for fixed-speed compressors since the oscillation frequency of conventional reed valves mainly depends on the effective moving mass, the stiffness and the preload and is therefore a system constant. In variable-speed compressors, however, correct valve timing is far more challenging as the duration of the suction and discharge phase varies greatly. Variable-speed compressors are increasingly used in domestic refrigeration as they allow for a better match of refrigeration demand and supplied cooling capacity of the refrigeration unit [7]. A further improvement of the valve dynamics of variable-speed compressors requires the consideration of new valve concepts.

The connection between valve closing delays and gas back-flow is investigated in [9], where the authors measured the valve motion as well as the pV-diagram of a hermetic reciprocating compressor. [8] measured the suction valve motion of a reciprocating compressor for varying compressor speeds. It
was found that the volumetric efficiency varies in relation to the compressor speed. Reed valves open and close several times during the suction phase and the discharge phase, which is commonly referred to as valve flutter. Since this flutter frequency is a system constant, different compressor speeds lead to valve closing delays that reduce the volumetric efficiency. The importance of reducing the valve flutter to increase the efficiency is also mentioned in [2]. The authors highlighted, that the COP improvement potential of a theoretical suction valve without valve flutter is 2%.

Recently, a few studies [3–5] were published which focus on practical approaches to reduce valve flutter and increase the valve dynamics of suction reed valves in reciprocating compressors for domestic refrigeration. The authors investigated electromagnetic and mechanical actuations to influence the valve dynamics of conventional suction reed valves.

The focus of this study is on the valve dynamics of the discharge valve in a hermetic reciprocating compressor for domestic refrigeration. A novel concept of a force assisted discharge valve based on a valve support mechanism is investigated. For this purpose, an in-house 0d/1d compressor simulation model is adopted to allow a simulation based potential analysis of the force assisted discharge valve.

Methods of statistical design of experiments (DOE) are applied to find optimal parameters for the discharge valve support mechanism.

2. Methodology
The first part of this chapter gives a short overview of the discharge valve support mechanism and the overall simulation strategy that was chosen for the simulation based potential analysis. Subsequently, the selected compressor simulation model and the modelling approach of the force assisted discharge valve are introduced. The last part of this chapter outlines the applied design of experiments approach of the simulation based potential analysis.

2.1. Discharge valve support mechanism
The principle of the discharge valve support mechanism can be seen in Figure 1 a.) and b.). The mechanism basically consists of a spring element and a rotating lever which transfers the supporting force to the discharge valve. This additional supporting force should lead to faster valve opening, reduced valve flutter as well as timely valve closing. Shortly before the piston reaches top dead center, the lever is moved back to allow the valve to close properly in time.

![Figure 1. Force assisted discharge valve](image)

2.2. Overall simulation strategy
The underlying study is about a first potential analysis of a novel discharge valve system. Due to the novelty of the system and the associated lack of system knowledge, a large number of potentially relevant system factors were identified. Many simulations are therefore required to determine the improvement potential of this new system, which is why a computationally intensive 3d simulation approach is not considered appropriate. Instead, an already existing in-house 0d/1d compressor simulation model based on [6] is adapted to comply with the compressor under consideration. It resolves
the most important physical effects with sufficient accuracy and is therefore particularly suitable for a potential analysis.

The simulation based potential analysis focuses on two different operating conditions of the ASHRAE process [1], which only differ with regard to the condensing temperature level. In the following, these two operating conditions are labeled as -23.3°C/55°C and -23.3°C/45°C, referring to the evaporating and condensing temperature level. Since the compressor is operated by the refrigerant R600a, the corresponding pressure levels are 0.64 bar/7.73 bar (-23.3°C/55°C) and 0.64 bar/6.04 bar (-23.3°C/45°C).

2.3. The 0d/1d compressor simulation model

The schematic gasline structure of the compressor simulation model is illustrated in Figure 2. It mainly consists of volumes connected by pipes. The cylinder is modelled based on the first law of thermodynamics according to equation (1).

\[-p \frac{dV}{dt} + \delta Q_a + \sum_i \left( \frac{dm_i}{dt} \cdot h_i \right) = \frac{d(m \cdot u)}{dt} \]  

(1)

The volumes of the compressor simulation model are calculated similarly to the cylinder. However, compared to equation (1), the first term corresponding to the work associated with a change in volume is not present. In contrast to the cylinder and the volumes, which are modelled as 0d components, the pipes are modelled as 1d components. The flow calculation inside the pipes is carried out numerically based on a 1d-CFD solver which is implemented in the compressor simulation model. The mass flows through the valves are calculated through equation (2), where \( \mu \) is a flow coefficient capturing all flow losses and \( \dot{m}_{th} \) is the theoretical mass flow of an isentropic flow assuming perfect gas behaviour. The flow coefficient \( \mu \) is a function of the valve lift and has to be calculated through 3d-CFD simulations or, alternatively, can be determined by measurements. The valve movement of the compressor simulation model is modelled using a 1d spring-damper-mass system. A more detailed description concerning the modelling of the force assisted discharge valve is given in the next section. The heat transfer is represented by a thermal network which is included in the compressor simulation model. For more information regarding the modelling approach of the compressor simulation model, see [6].

\[ \dot{m} = \mu \cdot \dot{m}_{th} \]  

(2)

Figure 2. Schematic gasline structure of the in-house 0d/1d compressor simulation model [6]
2.4. Modelling of the force assisted discharge valve

Figure 3 shows the standard discharge valve system of the considered compressor, a 9.5cm³ hermetic reciprocating compressor for domestic refrigeration appliances. The valve has a three-part structure consisting of the valve lamella, a damper and the clamp that fixes damper and lamella to the valve plate. The damper tensioned between the discharge valve and the clamp increases the stiffness and limits the lift of the lamella.

Due to the symmetrical design of the discharge valve, the system can be reduced from 3d to 2d. Since only the valve lift influences the mass flow rate, a description of the valve system in 1d is sufficient. Therefore, simple spring-damper-mass systems for moving parts like the valve lamella and the damper were introduced.

\[ \omega_{01} = \sqrt{\frac{c_1}{m_1}} \]  

(3)

In addition to the parts of the standard discharge valve system, the valve support mechanism is included in the force assisted discharge valve model, forming a 1d multi-body-system as illustrated in Figure 4.

The point mass \( m_1 \) which corresponds to the valve lamella is modelled according equation (3) where \( c_1 \) is the stiffness of the lamella in valve opening direction and \( \omega_{01} \) is the first undamped natural angular frequency of the lamella. Both system constants, \( \omega_{01} \) and \( c_1 \), were obtained from a 3d-FEM calculation. The second point mass \( m_2 \) which corresponds to the damper element was modelled according to the same principle as the lamella, with \( m_2 \) located at the contact point between the damper and the valve lamella. The discharge valve support mechanism is modelled as a rigid rotary oscillator. The piston, clamp and valve plate are also modelled as rigid bodies, whereby in contrast to the piston, the clamp and valve plate are fixed in position.

All contacts are modelled based on 1d force elements, except for the contact between valve lamella and valve plate, where a rigid body contact model is used. Force elements basically consist of a spring and a damper element. A trigger function, which is influenced by displacement and velocity differences of the contact points, ensures that only compressive contact forces are transmitted. The rigid body contact between valve lamella and valve plate is described as an impact without temporal expansion. A so-called restitution factor \( \varepsilon \) is used to calculate the velocity of the valve lamella after the impact on the valve plate.

The valve motion is influenced by the conventional valve force driven by the pressure difference between upstream and downstream side of the discharge valve, as well as the additional supporting force.
generated by the valve support mechanism. This supporting force is influenced by the piston position, the torsional spring stiffness, the preload angle and the moment of inertia of the lever.

The multi-body-system of the force assisted discharge valve is mathematically described by an inhomogeneous, non-linear differential equation system of first order which is solved through a fourth-order Runge-Kutta approach.

2.5. Application of the design of experiments approach

The potential analysis of the force assisted discharge valve is carried out using a design of experiments (DOE) approach. The DOE approach requires a clear definition of the system and consideration with all its potential influences. For this purpose, a system boundary must first be defined. The parameter diagram shown in Figure 5 systematically summarises all system variables related to the force assisted discharge valve system.

The input parameters are divided into design and signal parameters. The design parameter pool includes the geometrical shape of the support mechanism as well as the spring stiffness, the preload angle and the density of the material, while the compressor speed and the condensing pressure level are defined as signal parameters. Factors represent the subset of the parameters whose values (levels) will be varied within the DOE approach to determine the effect on the responses. These responses represent measurable output variables which provide a meaningful information about the system under study. In this case, the discharge mass and the discharge work are chosen as responses, as both are directly influenced by the discharge valve dynamics. It should be noted that depending on the final design of the discharge valve support mechanism, the cylinder clearance volume might be influenced. However, this influence is not yet considered in this potential analysis.

Once the factors and the responses are defined, an adequate experimental design must be selected. Effect diagrams, i.e. main effect and interaction effect plots which involve simple linear models, are chosen as primary tool to improve the design parameters of the force assisted discharge valve. Thus, mainly two-level fractional factorial experimental designs are selected as experimental designs for the simulation based DOE.
Several design parameter adjustment loops resulted in a set of design parameters (see Table 1) that lead to a better overall performance of the compressor and its responses.

| nr.  | parameter                  | unit    | value   |
|------|---------------------------|---------|---------|
| 1    | lever length left         | [mm]    | 2.0     |
| 2    | lever length right        | [mm]    | 13.5    |
| 3    | lever height down         | [mm]    | 0.5     |
| 4    | lever height up           | [mm]    | 1.8     |
| 5    | lever thickness           | [mm]    | 0.9     |
| 6    | lever width               | [mm]    | 6.0     |
| 7    | preload angle             | [°]     | -4.5    |
| 8    | torsional spring stiffness| [N m/°] | 0.135   |
| 9    | lever density             | [kg/m³] | 7900    |

### 3. Results

In this chapter, the final results of the potential analysis are presented. Figure 6 a.) and b.) show the discharge valve motion over the crank angle and the corresponding pV-diagram at -23.3°C/45°C and a compressor speed of 3000 rpm. The black solid line in Figure 6 a.) refers to the displacement of the discharge valve lamella at point 1, while the black dashed line shows the displacement of the valve support mechanism at point 5 (see Figure 4). The grey solid line represents the standard discharge valve lamella motion, having no valve support mechanism. It can be seen that the discharge valve support mechanism eliminates the intermediate oscillation (valve flutter) of the valve lamella, resulting in a smoother valve motion and a reduced valve closing delay. As a consequence of the smoother valve motion, the pressure fluctuations during the discharge phase are reduced, see Figure 6 b.).
Figure 7 a.) and b.) show the responses discharge mass and discharge work over a wide compressor speed range. The force assisted discharge valve leads to a smoother relationship between discharge mass and compressor speed. Thus, between 3000 rpm and 4500 rpm a significant improvement of the discharge mass can be observed. Furthermore, the difference between the two operating conditions -23.3°C/45°C and -23.3°C/55°C is smaller, indicating that the delivered discharge mass is less dependent on the operating conditions if a force assisted discharge valve is used.

Figure 6. Comparison of standard valve and force assisted discharge valve

Figure 7 b.) shows the corresponding curves of the discharge work over the compressor speed. Although the qualitative behaviour of the force assisted discharge valve is similar to that of the standard valve, some improvements of the discharge work are obtained especially at lower compressor speeds. At low compressor speeds, valve flutter is more intensive than at high compressor speeds, leading to significant cylinder pressure oscillations during the discharge phase which increase the discharge work. The force assisted discharge valve reduces these pressure oscillations and thus reduces the required discharge work.
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Table 2. Results of the force assisted discharge valve and comparison with the standard valve

| $\Delta_T/\Delta_c$ [°C] | speed [rpm] | discharge mass [mg] | $\Delta$ [%] | discharge work [mJ] | $\Delta$ [%] | $\Delta$COP [%] |
|------------------------|-------------|---------------------|--------------|---------------------|--------------|---------------|
| -23.3/45              | 1500        | 11.480              | -1.9         | 31.966              | -19.8       | 0.4           |
| -23.3/55              | 1500        | 10.926              | -3.4         | 25.501              | -21.5       | 0.2           |
| -23.3/45              | 2000        | 11.933              | 4.9          | 48.550              | -15.8       | 0.5           |
| -23.3/55              | 2000        | 11.441              | -1.4         | 38.357              | -18.0       | 0.4           |
| -23.3/45              | 2500        | 11.649              | -3.0         | 65.137              | -15.4       | 0.6           |
| -23.3/55              | 2500        | 11.529              | 0.1          | 51.982              | -15.8       | 0.6           |
| -23.3/45              | 3000        | 11.497              | 3.9          | 83.277              | -10.7       | 0.9           |
| -23.3/55              | 3000        | 11.199              | 9.8          | 64.880              | -10.2       | 1.0           |
| -23.3/45              | 3500        | 11.064              | 12.2         | 102.826             | -5.0        | 1.2           |
| -23.3/55              | 3500        | 10.768              | 22.1         | 78.524              | -4.0        | 1.8           |
| -23.3/45              | 4000        | 10.518              | 16.8         | 122.703             | 2.0         | 1.7           |
| -23.3/55              | 4000        | 10.226              | 5.5          | 90.707              | -1.6        | 0.4           |
| -23.3/45              | 4500        | 9.688               | 0.0          | 135.217             | -2.3        | -0.2          |
| -23.3/55              | 4500        | 9.411               | 0.0          | 97.767              | -2.6        | 0.1           |
| -23.3/45              | 5000        | 9.034               | 0.1          | 146.680             | -1.9        | 0.0           |
| -23.3/55              | 5000        | 8.661               | 0.0          | 104.188             | -2.7        | 0.2           |
| mean                  | 4.1         | -9.1                | 0.6           |                      |              |               |

The quantitative values of the responses as well as their relative change compared to the standard valve are shown in Table 2. On average, the force assisted discharge valve increases the discharge mass by 4.1% and reduces the discharge work by 9.1%, considering two different operating conditions and a wide compressor speed range. The last column in Table 2 represents the theoretical COP improvement neglecting changes in the mechanical and the electrical efficiency of the compressor. Although significant improvements of discharge mass and discharge work are achieved, the theoretical COP improvement is only moderate.

4. Conclusion

The underlying study includes a potential analysis of a novel force assisted discharge valve system. The results of the potential analysis show that the valve support mechanism significantly improves the valve dynamics of the discharge valve. In general, the valve dynamics is more robust in relation to different operating conditions and compressor speeds. Especially in variable speed operation, reduced valve oscillation and reduced valve closing delays lead to improvements of the discharge mass and the discharge work over a wide compressor speed range. Local break-ins in the discharge mass as a result of an unfavorable valve timing can be avoided. Depending on the operating condition and the compressor speed, the discharge mass increases up to 22.1% and discharge work reductions up to 21.5% were obtained. Furthermore, the analysis indicated a theoretical COP improvement potential up to 1.8%. However, such high improvements can only be achieved locally rather than over the entire compressor speed range. The average COP improvement potential of the force assisted discharge valve over two different operating conditions and a wide compressor speed range was determined to be 0.6%.

The potential analysis of the force assisted discharge valve shows promising results, however, the practical implementation of the valve support mechanism still faces some challenges regarding the influence on reliability and cylinder clearance volume.

Finally, it should be mentioned that the investigated concept of a force assisted discharge valve is not limited to hermetic reciprocating compressors for domestic refrigeration. It can be applied to any piston compressor equipped with reed valves.
Nomenclature

\[ c_1 \] spring stiffness valve lamella
\[ h \] enthalpy
\[ J_6 \] moment of inertia support mechanism
\[ m \] mass
\[ m_1 \] effective mass valve lamella
\[ m_2 \] effective mass valve damper
\[ \dot{m} \] mass flow
\[ \dot{m}_{th} \] theoretical mass flow (isentropic flow)
\[ n_c \] compressor speed
\[ p \] pressure
\[ p_c \] condenser pressure
\[ \dot{Q}_a \] external heatflux
\[ u \] internal energy
\[ V \] volume
\[ x_1 \] displacement valve lamella – centre
\[ x_5 \] displacement support mechanism – contact point to valve lamella
\[ \varepsilon \] restitution factor
\[ \mu \] flow coefficient
\[ \omega_{0_1} \] first undamped angular frequency valve lamella

Abbreviations

0d zero-dimensional
1d one-dimensional
2d two-dimensional
3d three-dimensional
ASHRAE American Society of Heating, Refrigerating and Air-conditioning Engineers
CFD Computational Fluid Dynamics
COP Coefficient Of Performance
DOE Design Of Experiments
FEM Finite Element Method
rpm revolutions per minute
TDC Top Dead Centre
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