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Investigations on pneumatically forced-actuated compressor valves

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Abstract. In the present paper the performance of a novel designed valve for reciprocating piston machines is investigated, which makes existing compressors utilizable for operating as expander. Three design parameters were identified as critical for the valves performance particularly in forced actuated mode. Within a numerical simulation a study on the crucial geometrical parameters, the influence could be observed. Afterwards the experimental setup for the integral test of the valve design is presented and also additional tests for single valve components.

1. Introduction

Intermediate storage of natural gas is necessary in regional and national provision, because of the distinction between an almost constant supply from the gas fields and the seasonal fluctuation of the gas demand for both industrial and domestic heat as well as electricity supply. The pressure levels in the most common used underground storage sites (UGS) are varying from less than 100 bar up to more than 300 bar [1]. A significant amount of energy has to be spent to compress the natural gas from the pipeline pressure to the storage’s pressure level. Contrary the nowadays usual way of unloading the gas storage is the expansion by the use of a simple throttling device, which means a waste of exergy. Previous investigations [1] have shown a potential power output of about 1 MW up to 4 MW when using an expansion machine instead of the simple throttling device for the storage unloading process. An even more complex plant is described in [2]: the necessary energy for preheating the expansion gas is provided by a cogeneration plant (CHP). The fuel for the CHP is the natural gas itself; only 0.5 % of the expansion gas is needed. For that configuration the produced electrical energy of CHP and expansion machine is 6.9 MW. However, the extension of an existing UGS to a power generating plant at the unloading process requires an additional gas expander. Alternatively in [3] a novel valve system was presented, which could retrofit an existing reciprocating piston compressor to a reversible usable expansion machine. Since the UGS anyhow needs the compression stages for loading the storage, the use of a retrofittable valve seems promising with respect to investment costs and space consumption.

To understand the working principle of the pneumatic actuated system a numerical simulation is a reasonable method. However, for the evaluation of the simulation a small scale compressor test with low pressure using air is appropriate.
2. Pneumatically actuated poppet valve

2.1. Valve design
In [3] the most promising design concept for a combined valve for both compression and expansion was found: the poppet type. The shape of the sealing element allows the control of the valve lift by the use of the pneumatic pressure difference provided by the process itself. That control pressure is applied to the backside of the sealing element in the poppet control chamber. By changing the position of the additionally installed electromagnetically actuated slider valve, the pressure level in the control chamber could be altered between suction and discharge pressure. In case of the valve on the high pressure side of the reciprocating machine (HP valve) following both modes can be performed: when the poppet control chamber is connected to the discharge chamber the valve works self-actuated. Connecting the pressure control chamber to the suction side a lifting force is acting on the sealing element caused by the pressure difference. The proposed design for the discharge valve of the reciprocating machine is shown in Figure 1.

![Figure 1: CAD-model of the pneumatically actuated discharge valve (valve springs hidden) [3].](image)

2.2. Important design aspects
Using a valve system like described above leads to additional aspects which have to be considered during the design process. The following control signal chain is looked at:
- Crank angle position delivered by sensor
- Signal processing by programmable control unit
- Activation of the electromagnetically actuated slider
- Build-up of the magnetic field until positional change of the slider valve
- Change of the pressure level inside the poppet control chamber
- Poppet lift response caused by the pressure difference
The time delay of the electronical signal between the measurement of the crank angle position until the application of the voltage to the coil of the magnet anchor is insignificantly small in this case. The first relevant lag is generated by the interaction between the magnet field and the movement of the magnet anchor. This switching time depends on the electrical current of the magnet coil. Furthermore the inertia of the magnetic anchor and the spring rate of the return spring have an influence on the slider movement. However, multiple valve system manufacturers have already introduced electromagnetic actuator systems which can be operated within the required circuit times and also with the switching frequencies.

The most influencing time delay is expected by the pneumatic control itself. The poppet control chamber is a finite volume filled with the compressible process gas. Changing the pressure level on the chamber’s inlet leads to a dynamic response, depending on the size of the volume and also on the pressure loss between the inlet and the control chamber. This behavior is superimposed by the change of the size of the control chamber by the poppet movement itself. Another important detail for that consideration is the leakage over the sealing elements between poppet control chamber and the suction respectively the discharge chamber.

Another notable point to be observed is the pressure loss over the valves. When retrofitting an existing machine with new valves, the overall efficiency should not decrease. Also the backward directed flow in the expansion mode has to be considered, since the efficiency during the expansion process has a significant influence on the amount of energy which could be recovered and is associated to the economical profitability.

Summarizing the above mentioned aspects it can be concluded that for the proper functionality of the pneumatically controlled valve, two important points have to be investigated: The electrodynamic behavior of the magnet which operates the slider valve and the extended valve dynamics caused by the poppets control chamber. For those observations numerical simulation and experimental testing are required.

3. Numerical observation

3.1. Numerical model

For preliminary observations of the system’s functionality, the valve dynamics and the overall reciprocating machine’s operational behavior a numerical model was set up. A simple thermodynamic multi chamber model was defined, as already described in [3]. The geometrical data for the numerical model as well as the thermodynamic boundary conditions were taken from a real reciprocating compressor, which will be equipped for the validation of the model. This compressor is described in chapter 4.1. The specific valve parameters like geometry, valve spring rate and mass of the sealing elements are known from the initial valve design fitted to the existing compressor.

The pressure loss caused by the flow through the valve has a strong influence on the dynamic valve behavior. That pressure loss is defined by the flow speed inside the valve and the pressure loss coefficient $\zeta$. A larger flow cross section leads to lower velocities; the $\zeta$-value depends on the valve lift and the overall valve geometry. Pressure loss coefficients for poppet valves used in internal combustion engines are well described in the literature. Nutting and Lewis [4] for example observed already in 1918 the pressure loss for different geometries for stationary flow experimentally. Waldron [5] found that the values derived from stationary testing are also applicable to intermittent flow. However, the flow pattern over the valve into the working chamber in engines is quite different to the one in compressors. Idelchik [6] gives resistance coefficients for various types of valves like single plates with and without guide or conically shaped sealing elements, which are geometrically very alike poppets. Similar observations were done by Vaughan [7] by the use of CFD-simulation; he also found specific points in the valve lift, where flow separation occurred and thus the function of the valve loss coefficient became discontinuous. Important geometrical parameters are the valve lift, the sealing element diameter, but also the shape of the valve seat and the sealing element. From Pohlenz [8] we can derive coefficients particularly for compressor ring and plate valves. In more recent publications, e.g. [9], [10] or [11], $\zeta$-values for
Compressor poppet valves are given. However, the coefficients of those publications are mostly afflicted by one unknown geometrical parameter, hence the conversion to our specific problem is not possible. An overview of pressure loss coefficients depending on the valve opening ratio is given in Figure 2, were x denotes the valve lift and d the poppet diameter.

![Figure 2: Pressure loss coefficient ζ of poppet and similar valves over the dimensionless valve lift x/d.](image)

It can be seen, that apart from the data of Idelchik [6] the ζ-values follow similar trends and are actually almost equal for small valve lifts. However, apart from the data of Nutting the values are only valid for single poppets. The ζ-value for the flow through multiple poppets might be larger since the mutual influence of the flow through each single poppet. Also Schulz [12] showed that each single sealing element will open differently from the others, caused by the valve pockets shape and internal pressure waves inside the cylinder. For the initial calculations an approximation of the curve presented by Pohlenz [8] is used:

$$\zeta = 0.5 + 4.5 \cdot \frac{x}{d} \quad (1)$$

A resolution of the crank angle revolution of 0.1° was defined, which is a temporal resolution of about 28.5 µs at the chosen drive speed of 585 rpm. The calculation was conducted with 10 iterations and the mass flow balance for in and outlet and also the converging of the mass flow was checked, to ensure a stationary point of operation. Using the preliminary valve design the geometrical parameters of the valve assembly were varied to observe the influence on the sealing elements motion.

3.2. Results from simulation
For the first study the fit between the sealing element and its guide was changed. A clearance fit is necessary to facilitate the poppet’s movement along its axis. However, a leakage path between poppet control chamber and suction respectively discharge chamber occurs caused by this clearance. The simulation was performed with different common tolerance fields, which are in accordance with DIN ISO 286. The gap was assumed evenly distributed over the circumference and the maximum width resulting from the clearance fit was set.

The relative leakage is defined as the mean value of the absolute leakage of the poppet normalized to the mass flow of the machine. Figure 3 shows this value during expansion mode. On the secondary y-axis the area of the gap of a single sealing element is presented. The leakage increases during the forced actuated mode as expected with a looser fit. The increase follows a similar trend like the growth of the gap area. Since the actuation time for the low pressure side is longer, the relative leakage is higher than for the high pressure valve.
The influence of the leakage on the poppet movement of the high pressure valve is shown in Figure 4. The increasing leakage at larger clearances leads to a slower valve response when forcing the poppet to move. The forcing starts at a crank angle of 325° and is completed about 17.5° later for the smallest calculated tolerance of “H7/g7”. For the largest observed fit the calculation leads to an additional delay of 5° of the crank angle. Contrary when stopping the forcing the sealing element with the looser fit moves faster to the closed position. This takes about 12.5° for the “H8/d9” and about 15° for the “H7/g7” respectively. Simulations with even larger clearances were performed, but the resulting calculated pressure levels in the control chamber did not converge. The poppet sealing element’s movement of the low pressure valve follows a similar trend and is hence not presented within that paper. Since the influence of the gap is higher during forced operation mode it can be concluded, that the loose fit should be chosen as small as possible.

In a second observation the size of the poppet control chamber was varied. Based on the volume of the preliminary design the controlled volume was simply multiplied by a factor. Two limits for that factor were found for the domain of convergence of the calculations. The lower limit of the mentioned factor of 0.35 is defined by the clearance for the poppet movement itself. Smaller volumes lead to a collision between the sealing element and its guide, hence the calculation aborted. Above the upper limit of 3 the gas inside the control volume reacts too slowly, so the pottets do not react on the applied pressure change.

The results for the forced actuated high pressure valve for factors between the above discussed limits are shown in Figure 5. For the initial design (factor = 1) a delay of the valve response between the signal

![Figure 3](image-url)

**Figure 3:** Mean value of the relative leakage (flow through the gap between poppet and guide normalized to the compressor mass flow) and gap area for a single poppet.

![Figure 4](image-url)

**Figure 4:** Influence of the clearance fit between poppet and its guide on the high pressure valve movement during expansion mode.

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The results for the forced actuated high pressure valve for factors between the above discussed limits are shown in Figure 5. For the initial design (factor = 1) a delay of the valve response between the signal
and the movement to the end position of about 15° and 20° can be seen. Minimizing the control volume to factor 0.35, leads to a decrease of the valve delay of about 7°. Increasing the volume leads to a significant increase of the delay. Also the starting point of the valve movement relocates to higher crank angles substantially. The calculated time delays have to be considered during the forced actuation to achieve the desired charge. From these observations it can be concluded that the volume of the control chamber has to be chosen as small as possible. This can be achieved by positioning the surface of the control volume, were the pressure change is applied, as close as possible to the sealing elements. Hence the ideal design would mean a pressure switch above each sealing element. However, in respect to the economic effort the usage of a single slider valve which is connected to a manifold, as already presented in Figure 1, is more reasonable.

![Figure 5: Influence of the volume of the poppet control chamber on the high pressure valve movement during expansion mode.](image)

### 4. Experimental setup

#### 4.1. Test compressor

For the functionality proof of the proposed design and validation of the data derived from the numerical model a test compressor is needed. A two-stage L-shaped Atlas Copco AR1 compressor (year of manufacture about 1955) is used. Each cylinder is double acting with each two suction and discharge valves per working chamber. The machine data is given in Table 1.

|                              | 1st stage | 2nd stage | unit   |
|------------------------------|-----------|-----------|--------|
| compressor speed             | 585       | min⁻¹     |
| free air delivery in normal operation | 8.6       | m³/min    |
| max. shaft input             | 46        | kW        |
| cylinder bore                | 285       | 170       | mm     |
| stroke                       | 150       | 150       | mm     |
| normal working pressure      | 1.77      | 6.87      | bar    |
| piston rod length            | 295       | 295       | mm     |
4.2. Instrumentation

To investigate the valve behaviour and the overall machine performance, an extended pressure indicator setup was intended. Piezo-resistive pressure transducers are installed in the working, suction and discharge chamber and additionally in the poppet control chambers. In Figure 6 the positioning of the sensors for the high pressure valve can be seen. The crank angle position will be determined by an inductive dead point sensor. The test equipment for the high dynamic measurements will be completed by a positional sensor for one single sealing element using a method based on eddy current. Thus it is possible to evaluate the $\zeta$-value of the valve with the help of pressure drop and the valve lift.

![Figure 6: Positions of temperature and pressure probes for dynamic valve performance testing.](image)

The volume flow will be measured by means of a calorimetric principle. Additionally installed pressure and temperature sensors are used for the acquisition of the different operation points. Overall the instrumentation allows the measurement of the compressor performance in a stationary operating point, but also the evaluation of the highly dynamical, transient processes.

4.3. Testing procedure

The first testing series will be performed using the modified high pressure valves for the first compression stage only. The reciprocating machine will be operated as a compressor, but forced opening of the discharge valve will be applied to check the valve performance. Simultaneously the possibility of a compressor capacity control by the rather uncommon forced opening of the discharge valve is checked. With the achieved data the numerical model will be adjusted and optimizations can be adapted to the low pressure valve of the first stage and also for the valves of the second stage.

Afterwards the compressor will be fully equipped with the controllable valves and the first stage will be operated as a compressor and the second stage as an expansion machine. Using the pressure indicator diagrams, the efficiency of both compression and expansion stage can be illustrated. Also the readout of the electric power of the frequency converter controlling the machine’s motor will be used for the assessment of the compressor and expander stage. By changing the machines drive speed and varying the valve timing different operation points can be observed and the limits of the valve control can be determined.
4.4. Complementary testing
Like already mentioned in section 2.1., the time delay caused by the build-up of the electromagnetic field in the slider is very important for the functionality of the valves. Thus a simple test setup is planned to determine the time between the two end positions of the slider. The electromagnet is controlled by a pulse generator. Simultaneously the position of the slider is measured by usage of an optical distance sensor. By synchronizing the pulse generator and distance sensor the slider valve delay can be determined. The pulse generator allows different switching frequencies, so a change of the reciprocating machines drive speed can be simulated. Another focus for this testing lies on the applied electric current on the magnetic coil. By increasing the current a faster slider movement should be possible.

The knowledge of the pressure loss coefficient of the valve is another important parameter to know. For this special case the pressure loss for both flow directions is of interest. In cooperation with Mario Cozzani Srl stationary flow tests will be performed to evaluate the necessary flow coefficients.

5. Conclusions and Outlook
In the present paper a critical view on design aspects for a force-actuated compressor valve was formulated. The proposed design allows the reciprocating machine valves to operate in the typical self-actuated compressor mode but also in a forced actuated mode, to operate the compressor reversible as expansion machine. By the use of a simulation model it could be shown, that the influence of the fit between the sealing element and its guide is small, as long as common tolerances for the clearance fit are applied. As expected the leakage of the pneumatic control chamber increases with larger tolerances and leads to a slower valve response. The size of the dead volume for the control chamber, which is unavoidable to ensure the flow paths for the control, has a much higher influence on the valve performance. An upper limit of the volume’s size where valve control is no longer possible was shown, since the gas volume becomes too inert.

The next step is the completion of the experimental test setups, which have been also presented in this paper. The expected results will give important data to improve the numerical model. Additionally the influence of the flow path to the poppets control chamber has to be observed more in detail. Especially strong pressure drops caused by flow deflection and sudden changes in the cross section might lead to pressure pulsations inside the control chamber and worse valve performance.

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