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Experimental testing of pneumatically forced actuated compressor-expander valves

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Abstract. The possibility of forced actuation of valves enables new operation modes for existing reciprocating compressors, e.g. capacity control or reverse operation as expander. A pneumatically forced actuated valve was tested within a reciprocating compressor and a quite complex behaviour was found, as shown in previous publications. A test rig outside of the compressor was set up for a better understanding of the system. Two measurement techniques were used to observe the sealing element motion over the time at varying operating conditions, like applied pressure levels and actuation time. The dependency of the valve motion to those operation conditions is evaluated and discussed. Improvements to the test bench and measurement techniques are also proposed to achieve more accurate results.

1. Introduction

Challenging the reasons for the global warming is one of the main driving factors of research and development in industrial technologies. Beside the development of completely new processes the optimization of existing technologies and machinery is a suggestive option. During the "International Conference on Compressors and their Systems" in 2017 a new valve design for reciprocating compressors was introduced, which allows to retrofit an existing piston machine to operate in compression as well as in expansion mode [1]. This development was driven by the energetic analysis of underground gas storages (UGS) for natural gas, which are used for the compensation between constant supply from the gas fields and the seasonal fluctuating demand by the consumers. A machine working in compression mode during the gas storage process and being able to be driven in expansion mode to unload the UGS is an economically and ecologically reasonable solution. Hence the efficiency of the storage process itself would be enhanced.

A numerical study with simplified models and assumptions was performed to investigate the most important design criteria for the valves, like tolerances and the dead volumes for the pneumatic actuator. Based on the theoretical results prototypes for the high pressure valves were manufactured and tested on a reciprocating compressor [2,3]. The functionality was proven and a compressor capacity control near to zero flow was achieved. The experimental results were compared to the simulation data presented in [1] and showed good agreement. However, the valve timing within the simulation needed to be adapted. It was assumed that this mismatch within the timing was caused by two main influencing factors. On the one hand the cylinder of the compressor is lubed with oil which is transferred by the pressurized gas to the valves and seems to lead to a significant sticking effect, which was not considered within the valve dynamic model. On the other hand the certainty of the

signal chain of the experimental test bench between the generation of the crank angle signal and provision of the control signal for the valve was inaccurate.

For the evaluation of the simulation a less complex experimental setup should be used, to avoid the previously mentioned additional intricate phenomena. A valve test outside the compressor would enable a more stable operation, since boundary conditions like pressure levels and temperatures could be kept constant. Also the mentioned valve sticking caused by the lube oil could be prevented, since the valve itself does not need any lubrication and clean air can be used for testing. An extension of the investigation to the additional effects can take place in a later step.

2. Valve design and working principle

During the following discussion only the compressor discharge valve which was investigated within the present study is discussed. Since the flow through an expansion machine is exactly the opposite of that through a compressor, the valve is further denoted as high-pressure (HP) valve in reference to the pressure level.

The proposed valve design is a poppet valve. Multiple mushroom-shaped sealing elements (poppets) are assembled in the valve body. Two different pressure levels can be applied to the cavity behind the poppets (control chamber) to enable the possibility of forced actuation. When operating non-forced actuated the control chamber is connected to the cavity which is surrounding the valve, in case of the discharge valve in a compressor this is the discharge chamber 'dc'. For the forced actuation a lower pressure than the surrounding pressure is needed. By switching the pressure level to low pressure a pressure difference is acting on the poppet which is lifting up the sealing element. In a compressor this lower pressure level can be provided by the suction chamber. The switching of the pressure levels takes place via an electromagnetically actuated slider valve.

Figure 1 and Figure 2 are illustrating the working principle of the HP valve within a reciprocating compressor. In the initial position (step1) the piston is in the bottom dead center and in the cylinder is at suction pressure. The slider (green) connects the control chamber behind the poppet ('p') with the discharge chamber ('dc'). While the piston (dark blue) moves towards the top dead center (step 2) the pressure inside the cylinder increases and at a specific point the poppet lifts up caused by the pressure difference between the cylinder and the discharge chamber.
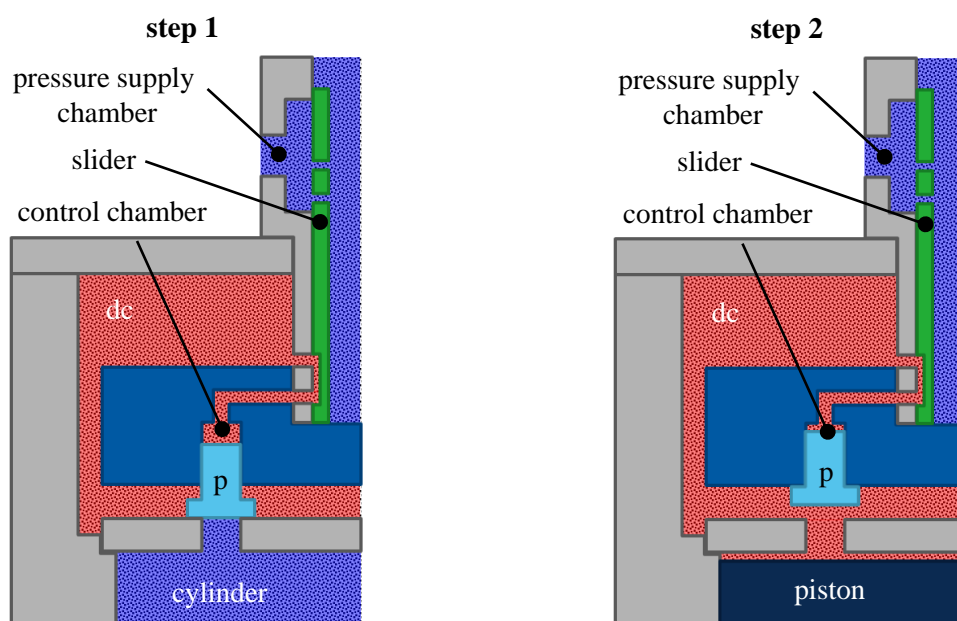

Figure 1. High pressure valve operating self-actuated.

The case of a forced-actuated operation is shown in Figure 2. The slider is being moved in its upper position (step 3). That leads to the connection of the pressure supply chamber, being at suction

pressure, with the control chamber. Now the pressure difference acting on both poppet surfaces leads to the forced actuation. The piston moves back to the bottom dead center (step 4) and the gas flow can pass the valve from the discharge chamber back to the cylinder.
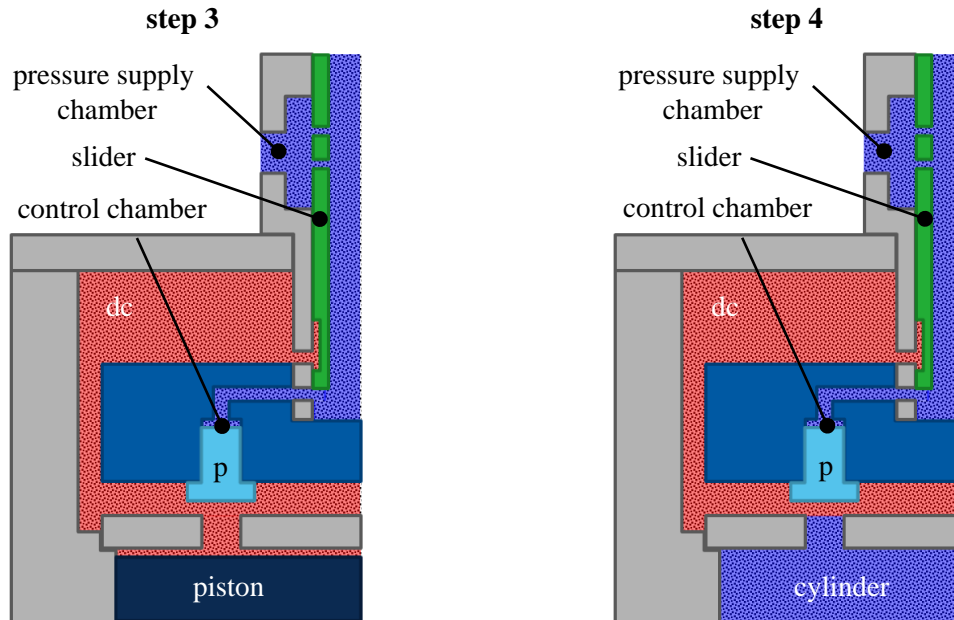

Figure 2. High pressure valve operating forced-actuated.

With the described methodology the high pressure as well as the low pressure valve are able to operate self-actuated for a regular compression mode, forced actuated for a capacity controlled compression mode and forced actuated for a reasonable expansion mode within the same machine.

The dynamic valve behavior can be described by

$$m_v \ddot{x}_v = A_v \Psi_{eff} \left(p_{con} - p_{cyl} \right) - k_f \left(x_{v,0} - x_v \right) \tag{1}$$

m_v is the mass of the sealing element and $\ddot{x}_v$ the poppet acceleration. The first term on the right hand side of the equation describes the force acting on the valves caused by the pressures applied on the surfaces of the poppet, with A_v the valve seat area and Ψ_{eff} the force coefficient. Ψ_{eff} varies slightly with the valve lift but is nearly equal to 1. Furthermore the cylinder pressure p_{cyl} and the pressure inside the control chamber p_{con} have to be applied. The second term takes account of the force generated by the valve spring. k_f is the spring rate, $x_{v,0}$ the spring pretension and x_v the valve lift. The damping factor and effects caused by valve sticking and friction are neglected.

As already described in [4] the varying pressure inside the control chamber p_{con} is an additional unsteady variable in the formula describing the dynamic valve behavior. Besides the slider position, which defines the pressure level at the inlet of the control chamber, the pressure is also influenced by the valve movement itself: The poppet is acting like a single piston within a cylinder and is increasing respectively decreasing the control chamber volume. Also the leakage over the poppet shaft needs to be considered.

Equation system 2 is describing the change of states. If not subscripted otherwise the values refer to the poppet control chamber. dt describes the time derivative, V the volume of the control chamber, which is coupled by the temporal derivative of the valve lift $\dot{x}_v$ to the formula for the dynamic valve behavior (equation 1). T identifies the temperature levels and m is the mass of gas inside the control chamber. It is only changing due to the mass flow from or respectively to the pressure supply $\dot{m}_{supply}$

and due to leakage $\dot{m}_{\text{leak}}$. For the first step a one-dimensional, adiabatic nozzle behaviour was assumed. With a predefined cross-section and flow coefficient the mentioned mass flows can be calculated according to [4,5]. The heat transfer between the valve and its surrounding is neglected and the heat Q is hence also only influenced by the enthalpy flow due to the leakage and the pressure supply. κ is the isentropic coefficient and c_p the isobaric specific heat capacity.

$$\frac{\mathrm{d}p}{\mathrm{d}t}=p\left(\frac{1}{T}\frac{\mathrm{d}T}{\mathrm{d}t}-\frac{1}{V}\frac{\mathrm{d}V}{\mathrm{d}t}+\frac{1}{m}\frac{\mathrm{d}m}{\mathrm{d}t}\right)$$

$$\frac{\mathrm{d}T}{\mathrm{d}t}=T\left(\frac{\kappa-1}{\kappa}\frac{1}{p}\frac{\mathrm{d}p}{\mathrm{d}t}-\frac{1}{m}\frac{\mathrm{d}m}{\mathrm{d}t}\right)+\frac{1}{m\cdot c_p}\frac{\mathrm{d}Q}{\mathrm{d}t}$$

$$\frac{\mathrm{d}V}{\mathrm{d}t}=A_v\cdot\dot{x}_v \qquad\qquad (2)$$

$$\frac{\mathrm{d}m}{\mathrm{d}t}=\dot{m}_{\text{supply}}+\dot{m}_{\text{leak}}$$

$$\frac{\mathrm{d}Q}{\mathrm{d}t}=\dot{m}_{\text{supply}}\cdot c_p\cdot T_{\text{supply}}+\dot{m}_{\text{leak}}\cdot c_p\cdot T_{\text{leak}}$$

3. Experimental setup

3.1. General setup
Purpose of the valve test rig is the investigation of the dynamic valve behaviour. By testing the valve outside a reciprocating compressor the valid equations can be simplified, since the unsteady boundary conditions like cylinder, suction and discharge pressure, as well as temperatures can be set constant. Also the switching frequency as well as the duty time of the electromagnetically controlled slider valve can be varied independently from the test compressor speed allowing the observation of a wider operating range. The test rig shall have the following specifications:

- adjustment of pressure levels for the cylinder p_{cyl}, the valve pocket p_{poc} and the pressure supply p_{sup} for the control chamber p_{con}
- control unit for setting of changing frequency and duty time of electromagnetic slider
- temporally high-resolution measurement of the valve lift and the pressures within the test rig

In the first step, the valve is tested under ambient conditions. Thus, the cylinder and chamber pressure correspond to the atmospheric pressure. A vacuum pump is used to provide the supply vacuum pressure in order to generate the required pressure difference on the poppet for the forced actuation. By adding a T-piece with an additional ball valve into the pressure supply line an adjustment of the vacuum to different levels is possible.

3.2. Pressure measurement
For high speed measurements piezo-resistive pressure sensors are used. The input pressure range is from 0 up to 7 bar absolute and the received signal is 100 mV at full scale output. The signal error is quoted with +/-0.5 % of the full scale output for the combination of non-linearity, hysteresis and repeatability [6]. In the experiments presented in this paper, one pressure sensor was used to monitor the pressure inside the poppet control chamber.

3.3. Valve lift measurement with eddy current sensor
For the valve lift measurement of one single sealing element an eddy current distance sensor is installed. Due to limitations caused by the valve working principle, it was not possible to mount the sensor in a way that the valve lift can be measured directly along the directional movement but orthogonal to it. Figure 3 shows the set up schematically.
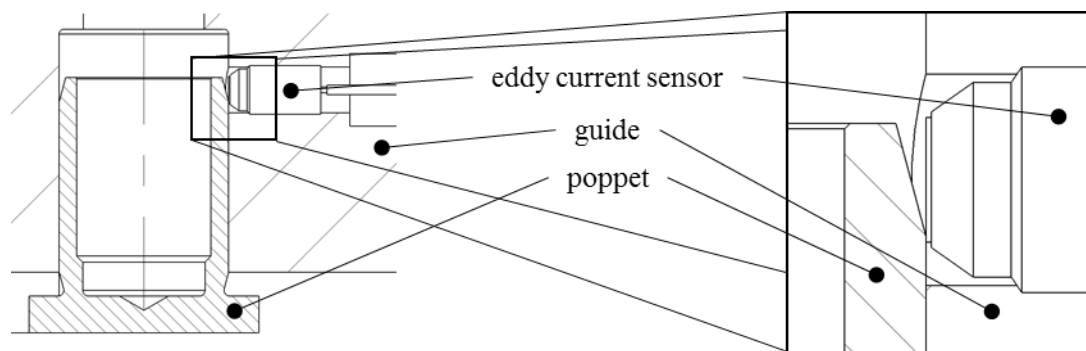
Figure 3. Experimental setup for the valve lift measurement with an eddy current sensor.

A high-frequency alternating current feeds a solenoid coil in the sensor. This induces an eddy current in the poppet. The distance between sensor and poppet influences the impedance of the coil. The impedance is measured and converted into the valve lift. Since a continuous valve lift signal is required a chamfer is applied to the poppet and the distance between sensor and chamfer is detected.

For calibration of the probe a feeler gauge is used to set specified valve lifts. The output signal is acquired at a sample rate of 10 kHz over 10 s for each data point and the mean values and standard deviations are calculated. The obtained calibration curve can be found in Figure 4; since the standard deviation is below 10 µA the error bar is not shown in the graph.
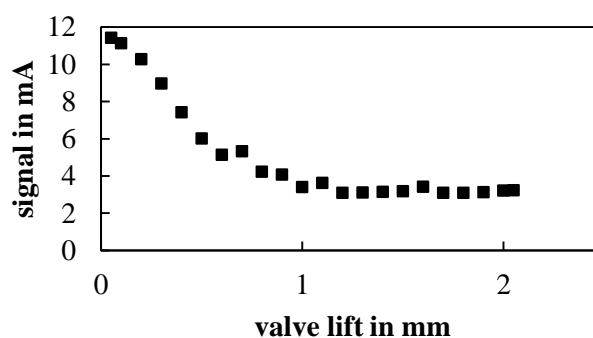

Figure 4. Calibration signal of the eddy current distance sensor for valve lift measurement of one single sealing element in mA.

Two characteristics can be seen in the curve. On the one hand the signal is very sensitive in a range between 0 and 1 mm valve lift with a slope of about 8 mA/mm. On the other hand beginning from 1 mm lift there is nearly no change in the signal value, hence an evaluation in that area is not possible.

For future measurement campaigns it is recommended to manufacture the poppet shaft a bit longer and also to increase the length of the chamfer to fit the measurement area. This might significantly increase the sensor sensitivity in the range from 1 mm valve lift to full stroke.

3.4. Valve lift measurement with laser distance scanner
For the determination of the valve lift a second measuring technique is used. A laser line scanner is installed in a way that the movement of the sealing element already equipped with the eddy current sensor can be observed simultaneously with that technique. The acquisition hardware allows a maximum sample rate of 4 kHz and a maximum resolution of 12 µm, the linearity of the signal is about 0.37 mm [7]. The measurement distance is between 190 mm and 290 mm; the measurement center is positioned on the contact face between poppet and valve seat plate. The test set up is shown in Figure 5.
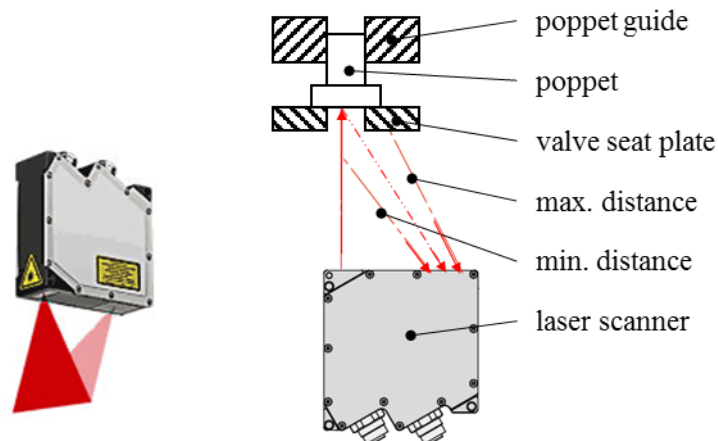
Figure 5. Valve lift measurement using a laser distance scanner. Maximum measurable distance is blocked by the valve seat plate.

3.5. Tested valve

The HP valve investigated in this paper was designed for the 1st stage of a 1955's Atlas Copco AR1KT compressor. 24 sealing elements, each with a seat diameter of 14 mm, are arranged on two concentric circles on the valve seat plate; 16 poppets on the outer circle, 8 poppets on the inner circle. A picture can be found in Figure 6.
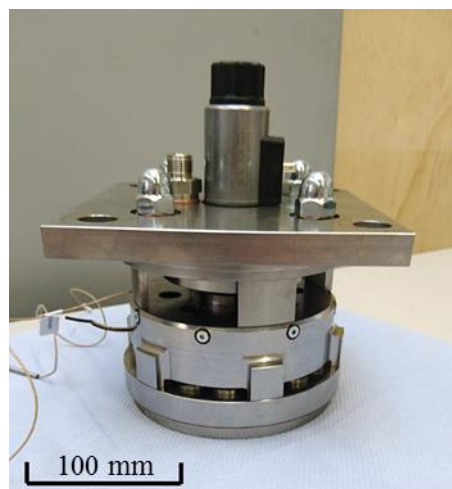

Figure 6. Tested discharge valve. On top: solenoid for actuation of the slider valve.

4. Experimental results and discussion

4.1. Experimental approach

Different pressure levels were applied to the poppet control chamber. The slider valve switching frequency was set to 10 Hz and different actuation intervals were applied. The experimental matrix is shown in Table 1. It is also shown in which operating point the actuation was inadequate, by means that not all sealing elements moved or the full valve lift was not achieved, or even the actuation failed.

Table 1. Experimental matrix for the tested valve. + indicates usable results, (+) tests where not the whole valve lift was achieved and - tests where the actuation failed.

	10 Hz switching frequency		
pressure supply	**actuation duty**		
	25%	50%	75%
400 mbar(a)	+	+	-
600 mbar(a)	(+)	+	+
700 mbar(a)	-	+	+
800 mbar(a)	-	(+)	+

After setting the switching frequency and the actuation duty the pressure level of the vacuum supply was adjusted to the defined value. The signals of the eddy current sensors, the laser scanner and the pressure probe were acquired with a sample rate of 4 kHz within a time of 20 seconds. Thus at 10 Hz switching frequency about 200 valve lift characteristics could evaluated. The mean valve stroke and pressure curves were calculated. An example for the achieved characteristic at 10 Hz switching frequency, 50% duty time and 400 mbar(a) applied pressure can be seen in Figure 7. The actuation duty time is defined as the percentage of time of the magnet slider being actuated during one cycle.
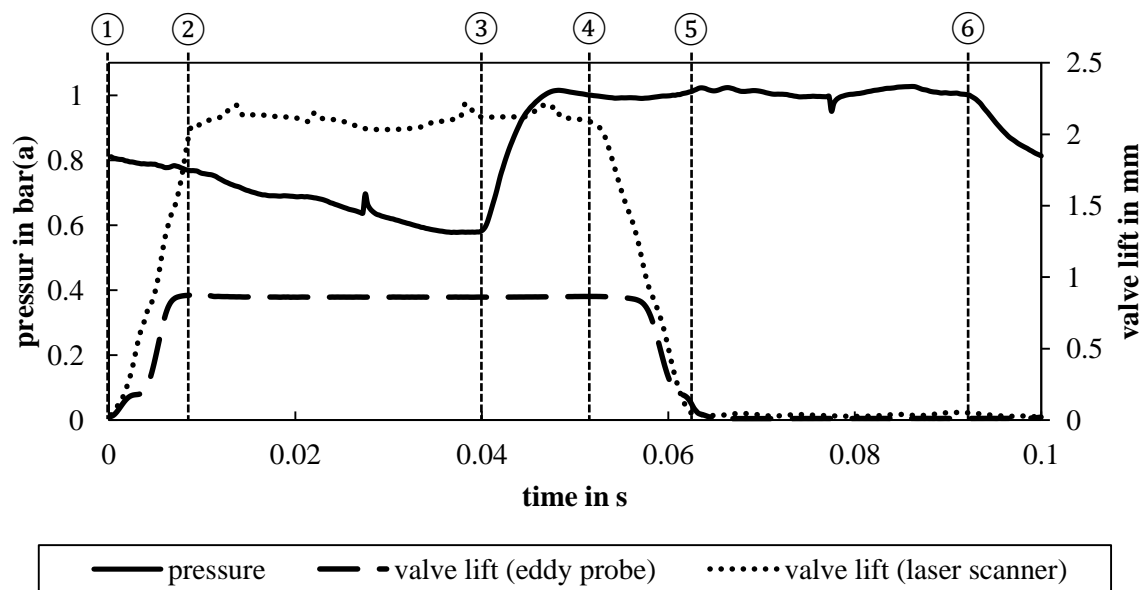

Figure 7. Valve lift and pressure characteristic for HP valve at 10 Hz switching frequency, 50% duty time and 400 mbar(a) pressure.

Six characteristic points can be derived from Figure 7:
① the sealing element starts to lift
② the sealing element reaches the maximum stroke
③ the pressure in the control chamber starts to increase
④ the sealing element begins to lower from maximum stroke
⑤ the sealing element reaches its initial position
⑥ the pressure in the control chamber starts to decrease

The time at which the poppet starts to lift off and at which it is completely closed again is the same for both position sensors. However, there are small deviations in time up to 1 mm valve lift and from

1 mm the eddy current sensor can no longer be evaluated satisfactorily. However, the laser scanner is generating reasonable results.

　　Four different time intervals characterising the valve movement are derived from the timeslots corresponding to the six characteristic points (Figure 7). These intervals define the reaction time of the sealing elements to the applied pressure change (subscript "reaction") and the time difference for the movement between the two end positions (subscripts "lift,increase" for valve opening and "lift,decrease" for valve closure).

$$\Delta t_{reaction,lift,increase} = t_① - t_⑥ + 0.1 \text{ s}$$

$$\Delta t_{lift,increase} = t_② - t_①$$

$$\Delta t_{reaction,lift,decrease} = t_④ - t_③ \tag{3}$$

$$\Delta t_{lift,decrease} = t_⑤ - t_④$$

t denotes the time at the characteristic points ① to ⑥.

4.2. Variation of the control pressure level

Figure 8 shows the valve timing at 50% duty time of the actuation at different supply pressure levels. The test was performed at ambient pressure, hence a low pressure level in the control chamber means a high pressure difference acting on the sealing elements.

　　In the range between 400 mbar and 700 mbar the sealing element's reaction on the opening pulse from the pressure change is only slightly increasing; at 800 mbar a significant increase to nearly the double of the initial value is noticeable. The time for the valve opening movement seems to follow a quadratic trend. The reaction of the sealing element to the closing pulse is decreasing linearly with increasing supply pressure and the time for valve closure is nearly constant.
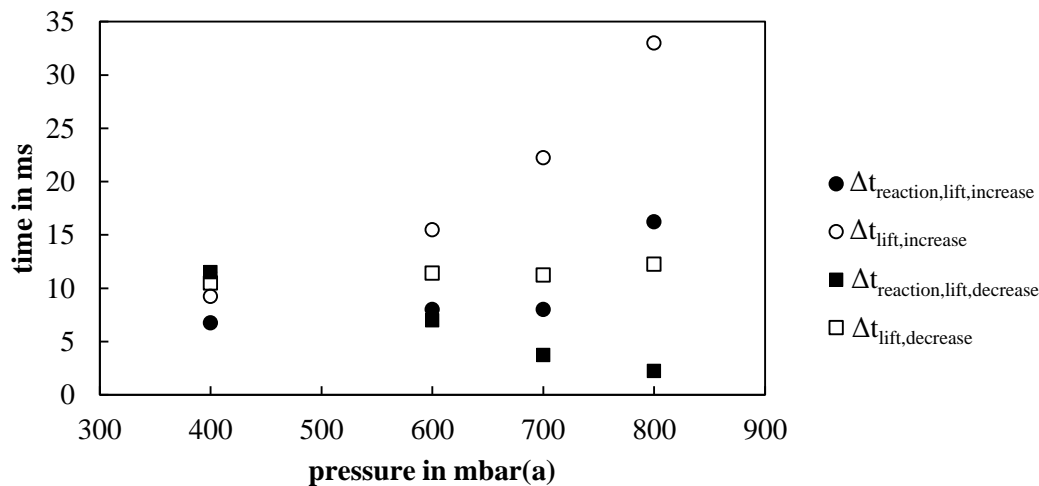

Figure 8. Sealing element reaction time and time for valve movement at 10 Hz switching frequency, 50% duty time for different supplied pressures.

4.3. Variation of the actuation duty time

Figure 9 and Figure 10 show the valve timings for varying actuation duty times at 400 mbar(a) and 800 mbar(a). The reaction of the sealing element to the pressure change in the control chamber seems to be independent from the actuation duty. This is different to valve closure, where the reaction time at 400 mbar is increasing dramatically. At 400 mbar(a) the times for valve lift and closing movement is

nearly constant over the varying duty time. At 800 mbar(a) the time for lifting the sealing element is increasing, the time for motion to the initial position is nearly constant.
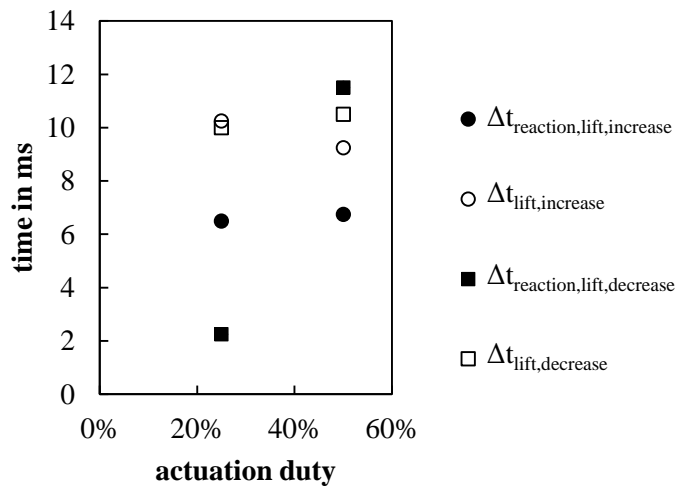

Figure 9. Sealing element reaction time and time for valve movement at 10 Hz switching frequency, 400 mbar(a) supply pressure for different actuation duty times.

Figure 10. Sealing element reaction time and time for valve movement at 10 Hz switching frequency, 800 mbar(a) supply pressure for different actuation duty times.
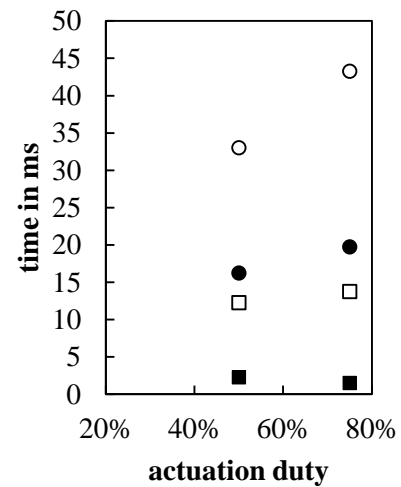
4.4. Discussion

The calculated valve timings for different operation conditions are illustrating the complexity of the proposed actuation system, although the test conditions are already simplified. The response time of the valve's sealing elements to the applied pressure changes for opening are nearly independent from the actuation duty of the slider valve, but is increasing with increasing control pressure, which means a decreasing pressure difference acting on the poppet. As soon as the pressure difference that is defined by the preload of the valve spring is exceeded, the poppet opens. The time for that change in the pressure difference is mainly influenced by the pressure level surrounding the valve and the vacuum supply. Another important factor is the pressure loss between vacuum provision and poppet control chamber, caused by the suction flow $\dot{m}_{supply}$. The higher the pressure loss, the smaller the suction flow and the longer the pressure decrease in the control chamber.

The time for lifting the poppet to the upper position is increasing with increasing control pressure. This can also be explained by the applied pressure difference between poppet control chamber and cylinder (refer to equation 1). This pressure difference is necessary to overcome the valves inertia and to compensate the valve spring tension. Lowering the supply vacuum leads to faster decrease of the pressure level inside the control chamber and also to lower pressure levels. Thus the poppet can move up faster at lower supply pressures. The significant increase of the lifting time at high actuation duties cannot be explained simply by the equations given in section 2. Further investigation of the causality by calculating the valve behavior by the simulation presented in [1] is necessary.

The reaction to an increase of the pressure inside the control chamber for closing the valve is decreasing with an increase of the applied pressure. Similar to the valve opening process the pressure difference over the poppets needs to fall below a certain value so that the valve spring force can overcome the force caused by the pressure difference over the poppet. At higher pressure levels in the control chamber and thus lower pressure differences acting on the poppet the limit for starting the valve closure can be reached faster.

Once the sealing element has started to close the interval for the complete closing is almost constant for different pressure levels and duty times. During that motion the main driving forces are the spring load and the valve inertia.

The faster reaction for closing and also the faster closing itself in comparison to the valve opening might also be an indicator for the pressure losses inside the slider valve for the change of the control pressure. A higher pressure difference between the supply pressure inlet and the control chamber inlet than between the control chamber and the pressure outlet over the slider valve would also lead to a faster pressure change during closure motion. The vacuum for valve lifting is generated by a vacuum pump which is connected by a hose to the slider valve. The depressurized volume inside that hose is small and hence during lift the supply pressure level might increase significant. An additional pressure reservoir directly in front of the slider valve and a fast reacting pressure probe monitoring the supply pressure level might lead to a better understanding of the valve motion and thus to an improved controllability of the valve.

5. Summary and outlook

In the present paper the function principle of a pneumatically forced actuated compressor discharge valve is described. A test rig for the investigation of the dynamic valve behaviour without the effects of the periodically changing influencing variables of the reciprocating compressor is proposed. The functionality of the measurement techniques used for valve lift observation, namely an eddy current distance sensor and a laser scanner, in combination with the newly developed valve needed to be proven. For this purpose a simplified test rig was built. Different pressure levels were applied for forced actuation and also the duty time of valve actuation was varied. A strong dependency of the sealing elements opening time and also the reaction on an applied pressure change for valve closure from the supply pressure level was found. The valve duty time only seems to have a lower influence.

The preliminary test rig revealed potential for improvement of the experimental set-up. The reaction of the eddy current sensor for distance measurement is limited to only half of the valve lift. A change in the sealing elements geometry within the probes measurement area might enhance the generated information. Furthermore the pressure supply for the actuation needs to be optimized. Attaching a pressure supply reservoir in front of the slider might compensate fluctuations in the pressure level caused by the actuation of the poppet. In addition, another high speed pressure sensor should monitor the pressure supply in order to enhance the evaluation of the valve movement.

The investigation on the valve behaviour with a first simplified approach showed good results. For a detailed analysis the complex boundary conditions have to be taken into account, e.g. the change of the pressure inside the working chamber as well as in the valve pocket. The achieved data can be used to optimize the simulation for the valve within a compressor model and to predict the behaviour inside the compressor more precisely.

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