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Daniel Nagy
Graz University of Technology

Raimund A. Almbauer
Graz University of Technology

Wolfgang Lang
Graz University of Technology

Adolf Burgstaller
Graz University of Technology

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Daniel NAGY[1]*, Raimund A. ALMBAUER[1], Wolfgang LANG[1], Adolf BURGSTALLER[1]

[1] Graz University of Technology, Institute for Internal Combustion Engines and Thermodynamics,
Christian Doppler Laboratory for Thermodynamics of Reciprocating Engines,
Inffeldgasse 21A, A-8010 Graz, Austria
Phone: +43 (0)316 873-4778, Fax: +43 (0)316 873-8080, E-mail: nagy@vkmc.tugraz.at,
almbauer@vkmb.tugraz.at, langw@vkmb.tugraz.at, burgstaller@vkmb.tugraz.at,
*Indicate Corresponding Author

ABSTRACT

The paper describes the use of a laser measuring system, to detect the movement of self acting valves used in reciprocating compressors. The valve movement is measured on real operation conditions. This is done by the use of an optical access through the shell and the suction muffler for the suction valve respectively through the cylinder head for the discharge valve. This was done to avoid every influence which is able to change the operating conditions of the compressor. The validation of the measuring system is carried out for the piston movement using the identical measuring system. These results are then used for the validation of a 1D compressor simulation model. In the 1 dimensional compressor model a spring-damping-mass model is used to evaluate the movement of the suction and discharge valves.

1. INTRODUCTION

1.1 Simulation and experiment

The market of household appliances in the cooling sector is a quite big one. More than 100 million cooling appliances (Fischer Weltalmanach, 2006) are sold every year. On this account a fierce competition takes place in the field of reciprocating compressors for household appliances. To be competitive and successful in this huge and severe market it is necessary to increase the speed and to decrease the costs of development at the same time. To achieve this ambitious target, the use of numerical tools to predict compressor behavior is indispensable. There are many parts inside, where mathematical modeling is expedient and necessary. Self acting valves controlling the gas exchange, are quite important parts inside the compressor; because they directly influence its efficiency. This paper elaborates how to adjust a mathematical model of valve movement by the use of experimental data.

Arranging a mathematical model, which makes a reliable prediction of the valve movement, is one of the most important indicators of a reasonable compressor simulation model. Many parameters of the compressor influence the valve movement, e.g. pressure difference over the valve, different damping factors caused by gas or oil or the natural frequency of the valve. In the past many researchers (e.g. Costagiola, 1950), have worked to combine these parameters in a mathematical way, this research results in a set of equations which can be summarized as basic valve theory (BVT).

Some parameters of this set of equations cannot be expressed through relations from literature; to evaluate these parameters it is necessary to measure the movement of the valves in a high quality way avoiding any disturbance of the physical phenomena inside the compressor. In the past, several methods for measuring the movement of compressor valves have been presented, but hardly any system was able to measure the valve movement without relevant influence on the working conditions of the compressor.

Small hermetic piston compressors reveal a compact layout, which makes the valve movement measurement difficult:

- Valves are small and light parts, no mechanical contact is allowed.
- Valves are not directly accessible.
- Valves are small in comparison to available transducers. Mounting a transducer or measurement system – without changes of geometry (gas inlet, top dead volume, and compressor shell) – is difficult.
- Compressor shell is hermetic

With this new approach it is possible to measure the movement of the valves in a high quality and a good repeatability without relevant influences on the compressor working condition. Therefore good agreement between simulation and experiments can be achieved.

1.2 State of the art

Because of the facts described in the previous chapter, only contactless measurement methods can fulfill the requirements. The use of a capacitive transducer is one of the simplest ways for contactless measurement. Nevertheless the transducer must be fitted near to the valve, which causes a considerable change in the gas flow. An inductive transducer has a more compact design and can be fitted with only a small influence on the working conditions. Due to its high non-linear characteristic, this type of transducer cannot be used for the total valve lift.
Ludu *et al.* (2000) visualized the suction and discharge valve motion using an endoscope video system. A CCD camera filmed the valve motion with the aid of stroboscopic light. The compressor was installed in a special hermetic shell, which was larger than the original shell. The test conditions were kept constant by the use of a rotameter, thermoelectric resistors and temperature sensors. An optical crank angle marker provided the trigger signal for the video system.
Several authors report the use of a Laser Doppler Vibrometer (LDV) for the valve lift measurement in hermetic piston compressors. An LDV is an instrument which can measure the laser beam direction component of the velocity of a moving reflective solid surface. The laser beam is focused on a surface and the reflected beam is detected. In the case of a moving surface – due to the Doppler Effect – the frequency of the reflected light compared to the source light is shifted. This frequency shift is then evaluated by an interferometer and converted to a voltage signal. The output voltage signal is a function of the velocity of moving surface. The range of velocity and distance in which the signal shows a high quality result depends on optical equipment and the signal processor of the LDV.
Buligan *et al.* (2002) analyzed this measurement method extensively in their work. They describe the effects of optical disturbances (like light scattering, reflection) and of the working environment on the LDV signal. The optical access was made by two quartz windows, which allowed reaching the valve surface with the laser beam through the compressor shell. The problem of light scattering on the oil particles was suppressed by inserting a protection screen. The test design contains no relevant modifications to the real compressor geometry and to its characteristic volumes.

2. TEST BENCH

The LDV method was extended in this work to meet further requirements. In order to keep the conditions as close as possible to real conditions during valve lift measurement, the compressor was run in a calorimeter. The calorimeter is used to provide constant boundary conditions for the standard ASHRAE test. It measures the COP (Coefficient of performance) via the cooling capacity (mass flow) and the electric power of the compressor. The equipment, which is available at the CD-Laboratory, is a "low power" calorimeter, which is especially designed for testing compressors with small cooling capacities. Only one compressor can be measured in the cabinet at the same time in order to achieve high accuracy results. The technical features of the calorimeter are:

- Refrigerant: R600a
- Cooling capacity: < 300 [W]
- Compressor electric (input) power: <200 [W]
- Conditioned room temperature: 25 – 40 [°C] (adjustable) (ASHRAE standard: 32 [°C])
- Suction pressure: 0.3 – 1.5 [bar] (adjustable)
- Discharge pressure: 5 – 10 [bar] (adjustable)

- Compressor voltage: 60 – 270 [V] at 50 – 60 [Hz]

The compressor, which was used for the valve lift measurement, was nearly identical to a HTK-55AA production-compressor. The main specifications of this compressor type are as follow:

- Bore: 21.1 [mm]
- Stroke: 16 [mm]
- Compression ratio: 66 [-]
- Electric power: ~58 [W]

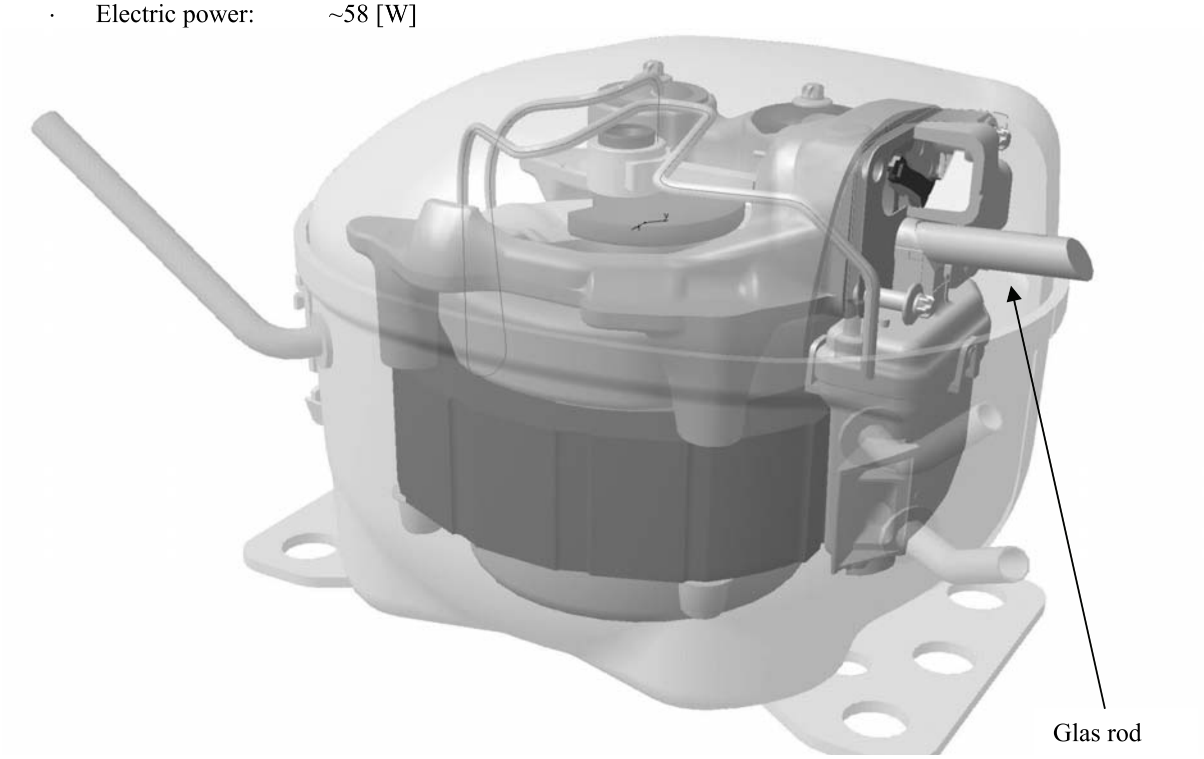
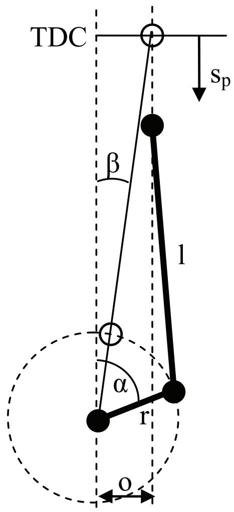


Figure 1: Compressor with optical path

The laser beam is guided through the compressor shell and the suction muffler respectively the cylinder head, depending on the measurement of the suction- or the discharge valve. The optical access is made by two cylindrical acrylic glass elements (with diameters 8 and 10 mm respectively). Disturbances of scattering, reflection and relative movement of two optical windows are avoided. In case of the suction valve measurement the glass rod is fitted rigidly in the neck of the suction muffler (Figure 1). The surface of the optical element provides the same shape as the original gas path in the muffler and does not interfere with the gas movement in the suction line. The discharge valve is accessible through the cylinder head and does not influence the discharge gas flow. The two glass rods generate no relevant modifications – concerning the gas flow and the thermal behaviour – in the shell. Both glass rods are hermetically sealed on their passage through the shell wall. As the glass elements connect the cylinder head with the shell, this design needs a rigid suspension of the inner compressor parts with the shell. This is carried out by replacing the springs of the suspension with rigid elements. Although the compressor is mounted on a rigid mass, some vibrations still exist. These vibrations are corrected by measurement of the compressor shell movement with the same device. This movement has to be subtracted from the measured valve movement to get the correct values. The phase allocation of both signals has been carried out with the signal of an acceleration sensor. The valve lift curve was time-integrated from the valve velocity results.

1.3 Validation of the measurement system

In order to validate the measurement system following experiment was carried out: The compressor was driven without valves, so the laser beam entered in the cylinder touching the piston top surface. The piston top was coated with reflective color as the color of the piston is rather dark. The piston movement was measured with the identical measurement system. The compressor crank drive determines exactly the piston movement (Figure 2). Equations (1) and (2) describe the piston position in dependence of the crank drive parameters. Figure 3 compares experimental and calculated piston velocities. Differences between both curves are small. One part of the difference is explainable by the disagreement of the assumed motor speed for the calculation and the motor speed in the experiment. The compressor electric motor is an asynchronous motor driven with a line frequency of 50 Hz. The motor speed depends on the load and cannot be exactly predicted. After the time fitting of the two curves, the standard deviation of the measured and the calculated velocities are below 0.04 m/s and the maximum deviation is below 0.12 m/s. The third curve shows the residual deviation between both curves. Small aberrations can be seen especially in the dead centers. This is because the piston slapping affects the experiment which is not considered in the calculation.



$$s_p = (r+l)\cos(\beta) - r\cos(\alpha) - l\sqrt{1-\left(\frac{r}{l}\sin(\alpha)-\frac{o}{l}\right)^2} \quad (1)$$

$$\beta = \arcsin\left(\frac{o}{r+l}\right) \quad (2)$$

Figure 2: Crank train geometry and equations

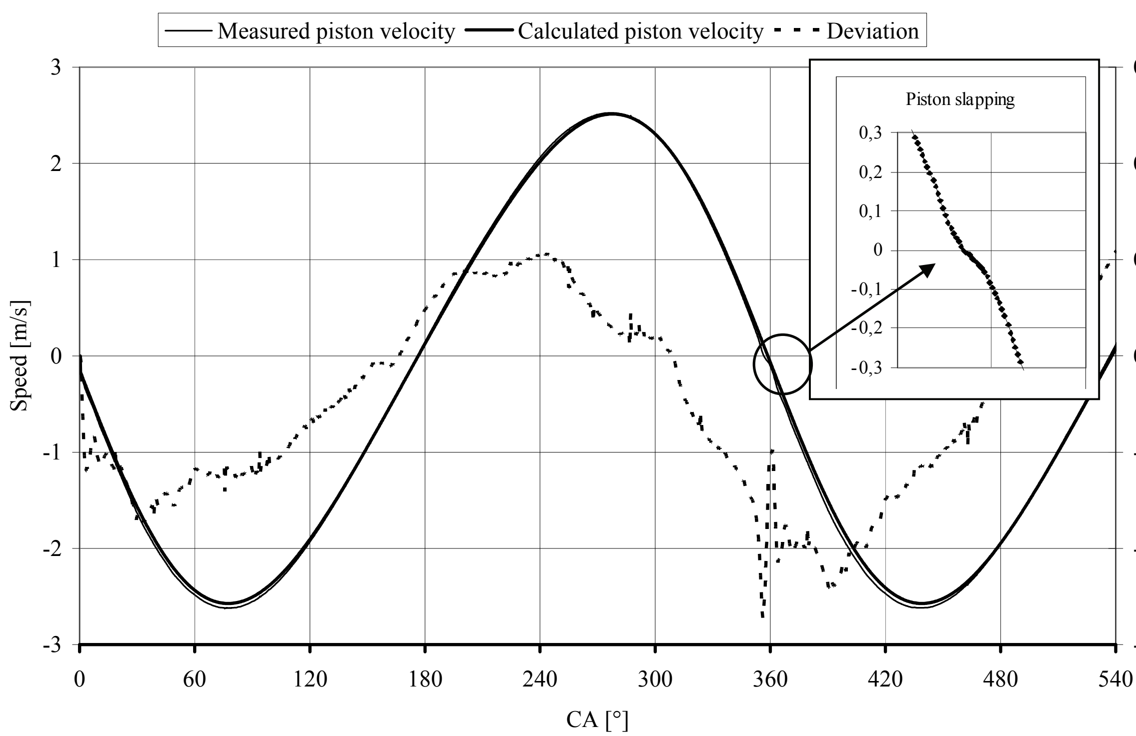


Figure 3: Measured and calculated piston velocity

In Figure 4 the time integral of the deviation velocity of Figure 3 is shown. Obviously there is a trend (see trend line in Figure 4) in the result where the integral tends to become more and more negative. This trend is a measurement mistake which can be corrected by the fact that the valve lift can never be unequal to zero as long as the valve is closed. Therefore this effect has not been further observed. The mistake in the validation measurement is approx. 2% of the piston movement.
The experiment shows that the valve lift can be measured with the presented system within a range of - 2.5 m/s up to 2.5 m/s with accelerations of more than 600 m/s². The displacement which can be covered is far beyond the valve lift values of the investigated compressor. Finally, the changes in working process of the compressor due to the installation of the measurement system are small.

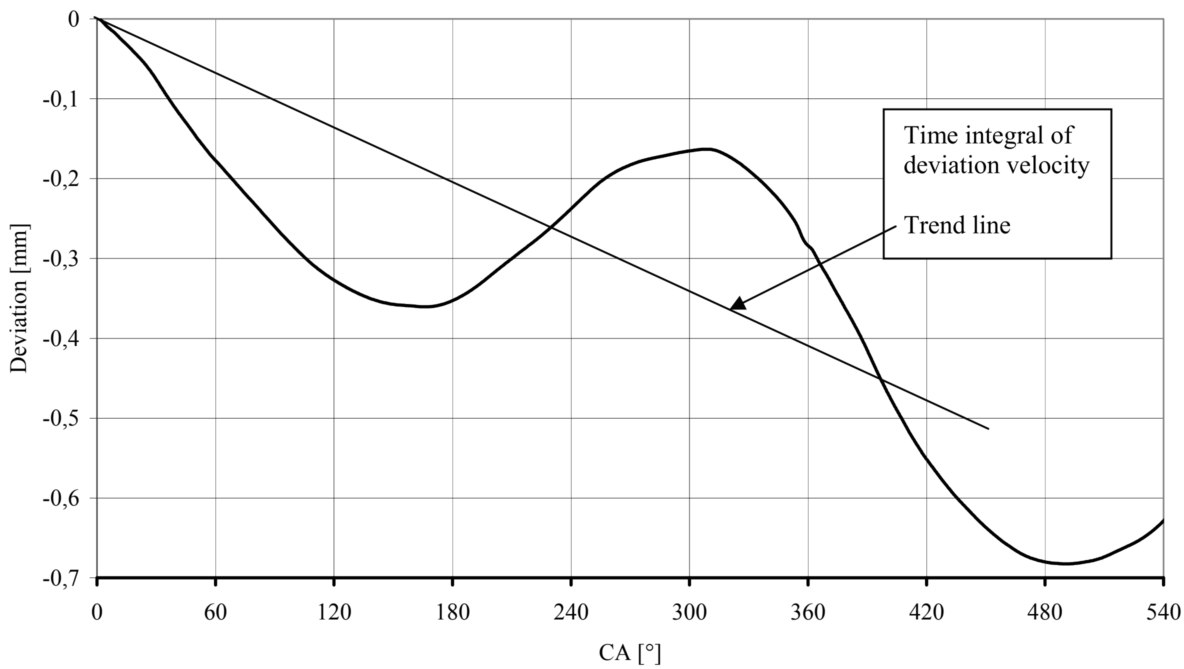


Figure 4: Time integral of deviation of piston velocity

3. VALVE MOVEMENT AND MEASUREMENT RESULTS

The valve movement is the result of all forces which act on it. The forces comprise inertia, damping forces, spring force, gas pressure force, rebound force, and oil sticking. Damping forces include material damping, which is insignificant, and the highly non linear damping effect of gas quenching between valve and valve plate. The gas forces depend on the transient pressure distribution on the valve surface, which show strong gradients on the inflow side. The existence of a rebound force is likely, as the valve interacts with the valve plate with an elastic defection. Thus a part of the kinetic energy of the valve is stored and returned in the opposite direction. This reduces the preload force and helps to lift the valve at a smaller gas pressure force. Oil sticking is an affect which can hardly be measured for such a small compressor. It depends on the oil viscosity, geometry, oil quantity, etc. All effects must be reproduced by any valve model in order to get the correct valve movement. As the gas pressure force is the driving element, the oscillating gas flow in the suction muffler must also be precisely simulated. The valve movement is therefore a very sensitive result for a complete compressor model. A full transient 3D-simulation of the valve movement including gas flow and material behavior together with some special effects like oil sticking and rebound force is demanding. Some parameters in such a simulation still have to be matched with measurement results. The simpler a valve model is the less physical effects can be covered. Nevertheless it is important to include the most important effects and to assess the smaller effects by simplified models with parameter adaptation with measurements.

3.1 Compressor model system

The compressor simulation model consists of the following parts: (a) a 1-dimensional model for the flow in suction and discharge line, (b) a 0-dimensional model of the cylinder with the volume changing according to the piston movement solving the 1st Law of Thermodynamics (Abidin *et al.,* 2006) (Figure 5), (c) a real gas formulation developed by NIST used for the equation of state of isobutene, and (d) a so-called thermal network, assessing the temperatures inside the compressor (Almbauer *et al.,* 2006). This thermal network estimates the conductive heat transfer using a lumped conductance approach. Figure 5 shows the gas path and the cylinder of the compressor model.

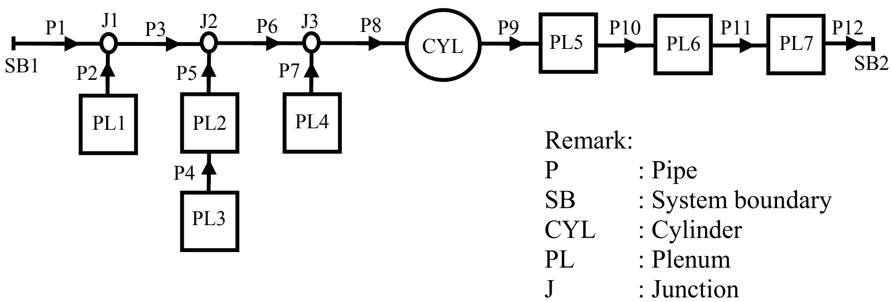


Figure 5: The schematic of the 0- and 1-dimensional models of the compressor

The valve movement is computed according to the already mentioned basic valve theory (BVT). The valve is described as an oscillating one-mass spring system described by Equation (3). It consists of the inertial, the damping, and the spring force on the left side, and on the right side the pressure force and the spring preload are written.

$$m_v \cdot \ddot{x} + d \cdot \dot{x} + c_v \cdot x = A \cdot \Delta p - F_o \tag{3}$$

This differential equation is solved by applying a Runge-Kutta method of 4th order resulting in a valve lift curve over time. The massflow rates at the suction and discharge ports are calculated using the stream flow equation under consideration of the flow efficiencies of the ports, namely in Equations (4) and (5):

$$\dot{m} = A_{eff}\ p_0 \sqrt{\frac{2}{R\,T_0}} \sqrt{\frac{\kappa}{\kappa - 1}\left[\left(\frac{p^*}{p_0}\right)^{\frac{2}{\kappa}} - \left(\frac{p^*}{p_0}\right)^{\frac{\kappa+1}{\kappa}}\right]} \tag{4}$$

$$A_{eff} = \mu\sigma\,\pi \frac{d_v^{\,2}}{2} \tag{5}$$

3.2 Parameters of the valve model

The spring rate was determined by a static measurement of the force. Therewith, the spring preload force was detected. The oscillating mass and the material damping were derived from the measured natural frequency and its damping over time. The LDV was also used for this test. As the pressure distribution on the valve surface depends on the gas flow pattern, it was estimated by the CFD-simulation of a steady state flow through the valve. In addition, the flow coefficient $\mu\sigma$ was determined using the same simulation. Nevertheless it has to be stated that the steady state CFD results do not directly represent the real behavior and oil sticking, gas quenching, and rebound effects are still adjustable parameters. Assuming that the valve movement is correctly measured by the experiment, the parameters can be chosen in the way that the simulated valve movement optimally fits. For the following simulation results the measured values of spring rate, oscillating mass (from natural frequency), and material damping have been taken. The $\mu\sigma$-value was also taken form the static CFD-simulation. In order to get a maximized consistency between measurement and simulation all other parameters have been varied.

3.3 Valve lift results

Figure 6 shows the results of the measurement and the corresponding simulation for the valve lift. Measurement means the time integral of the measured velocity. The simulation runs at exactly the same rotational speed of the measured compressor. The rotational speed of the measurement can be determined by the duration between two distinct amplitudes of the valve lift curve. The time adjustment of measurement and simulation is more sophisticated. As the experimental equipment has no angular position sensor the beginning of the second valve lift of the suction valve has been taken for adjustment of simulation and measurement. In addition, the measurements are corrected for their trend line, so the valve lift is zero when the valve is closed. The measurement of the suction valve shows that it opens five times during the suction period. This characteristic behavior can almost be reproduced by the simulation. The first amplitude can essentially be influenced by the oil sticking force. All other amplitudes depend less on this effect. An important influence on all amplitudes is coming from the area which is used for the calculation of the pressure forces. Nevertheless the consistency is of both curves is not perfect. The discharge valve shows only one amplitude. Here the parameters are varied in a similar way and the agreement between measurement and simulation can be easier obtained.

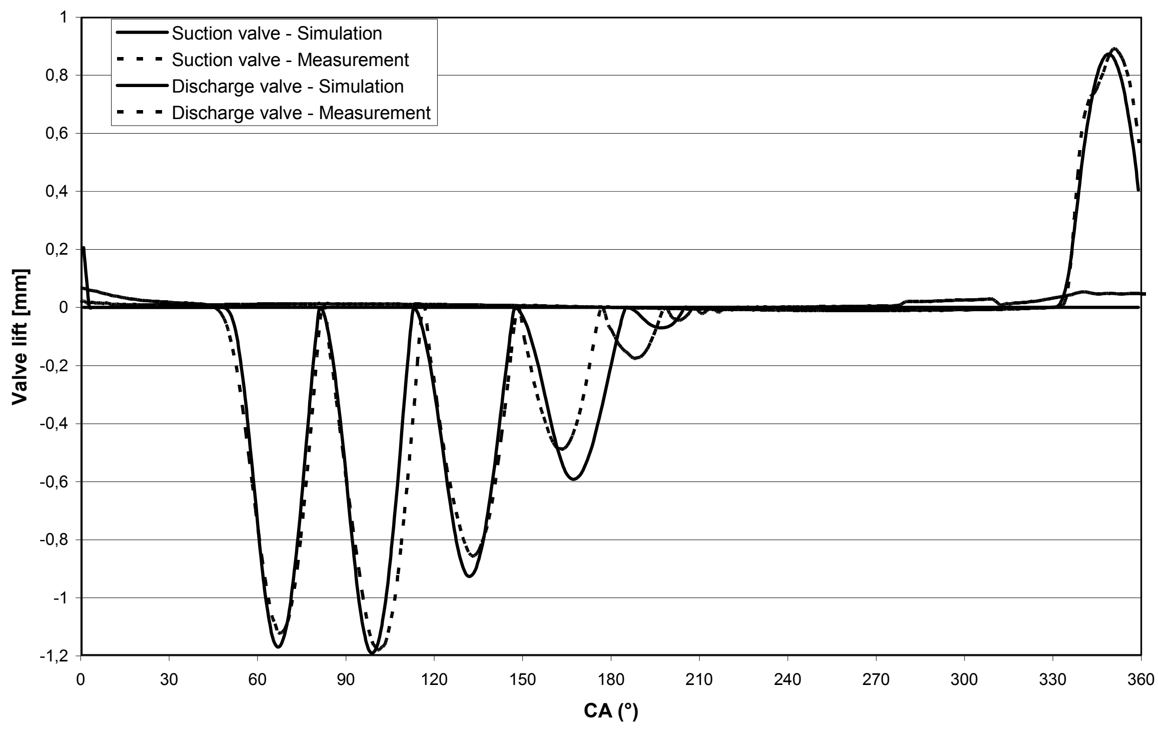


Figure 6: Suction and discharge valve lift curves (Measurement and Simulation)

4. CONCLUSION

The presented work shows that the valve movements of a small hermetic piston compressor can be measured using a LASER Doppler Vibrometer. The application of the LDV needs only an optical access to the valve surfaces, which has been realized with two acrylic glass rods. This results in only a negligible change of the working process. The validation of the measurement equipment shows that the measuring error is in the area of approx. 2% of the valve lift. As the LDV is linear in the entire range of the valve movement, the experiment gives reasonable values for valve speed and valve lift in every position of the valve. This fact has been used to validate a compressor model. The model includes a 1-dimensional simulation of the flows in the suction and the discharge lines. It also includes a 0-dimensional simulation of the cylinder and for each valve a one mass–spring oscillation model. The amount of parameters in the valve models is still too high to be able to adjust the models from scratch. Therefore as many measurements and CFD-simulation results as possible are taken to minimize the uncertainties. Nevertheless there are some parameters which can only be adjusted with the help of exact measurements of the valve movement. The valve movement is interdependent of the gas flow, the gas condition inside the cylinder (according to the piston movement), and its own characteristic. Therefore it is a good indicator of the exactness of the entire compressor model. The adjustment of the valve model has been carried out in a way that the measurements are reproduced as good as possible. The achievable coincidence is reasonable.

NOMENCLATURE

ASHRAE	American Society of Heating, Refrigerating and Air-Conditioning Engineer	
BVT	Basic Valve Theory	
CFD	Computational Fluid Dynamics	
COP	Coefficient of Performance	
LDV	Laser Doppler Vibrometer	
$\mu\sigma$	flow coefficient	(-)
m_v	effective valve mass	(kg)
c_v	spring rate of the valve	(kg/s²)
d	damping constant	(kg/s)
x	valve lift	(m)
$\dot{x}$	valve velocity	(m/s)
$\ddot{x}$	valve acceleration	(m/s²)
A	valve cross sectional area	(m²)
A_{eff}	effective flow area	(m²)
d_v	valve cross sectional diameter	(m²)
Δp	valve flow pressure difference	(Pa)
F_0	preload force	(N)
p_0	upstream stagnation pressure	(Pa)
p^*	downstream static pressure	(Pa)
R	gas constant	(kJ/kgK)
T_0	upstream stagnation temperature	(K)
κ	specific heat ratio	(-)

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