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ABSTRACT

A self actuated valve is commonly used in the intake manifold of small hermetically sealed reciprocating compressors. During the gas exchange process the valve losses influence the overall performance due to the throttling process which increases the pressure drop between the intake port and the gas chamber of such a machine. So there is a rise in the gas exchange work respectively the piston work to get the same discharge pressure. These valve losses influence the COP in two ways. First, in the increase of the consumption of electrical power and second, in the decrease of the mass flow rate.

Due to this fact it is important for the designer to understand the dynamic behavior of such a valve.

In this paper, numerical and experimental investigations are done to quantify the influence of main parameters of the suction valve on the overall performance of the compressor. The thermodynamic cycle calculation is performed using the AVL BOOST, a CFD software program. The calculation model covers the whole compressor domain between shell inlet and outlet. The dynamic valve displacement is calculated using an equivalent spring-damper-mass system. The experiments are effected on a calorimetric test bench.

1. INTRODUCTION

In a reciprocating compressor the variation of the gas chamber volume is controlled by a cyclic piston movement. The gas exchange process is typically regulated by self actuating valves (Figure 4) which periodically open and close the intake and outlet ports to and from the gas chamber. The gas enters into the compression chamber through the intake valve. After closing the valve the high pressure cycle is taking place and the gas in the chamber is compressed to a certain discharge pressure. At this time, the outlet valve starts to open and the compressed gas leaves the gas chamber through the outlet port. In the last sequence of the cyclic process the residual gas of the clearance volume expands. This cyclic process (as shown in Figure1, point 1 to 4) determines the indicated power consumption on the piston. In combination with the efficiency of the mechanical and electrical devices the essential input power is determined. The second value for evaluating the thermodynamic compressor performance is the cooling capacity which is indicated by the delivered mass flow over the compressor shell outlet. Following this compressor cycle the valve is a key component of a compressor as it determines both the overall performance and also the reliability of the machine. According to industry studies, valve failures account for more than 40 % of the unscheduled compressor shutdowns. Due to this fact it is important for the valve designer to understand the flow of gas through a valve, the way the valve operates in the compressor and how the basic dimensions influence the valve dynamics in order to improve the efficiency and also the reliability of such a compressor.

Some authors have done experimental and theoretical parametric studies on hermetic reciprocating compressors. Rigola, J., Perez-Segarra, C. D., *et al.* (2005) have done parametric studies focused on geometrical parameters of both, a cylinder group and an intake manifold considering different working conditions. Other papers are dealing with calculations and analyses of the flow conditions through reed-type valves. Pereira *et al.* (2007) made a numerical analysis of the flow through a discharge valve in which the compression process and the gas dynamics in the discharge muffler is included. A parametric analysis for a suction valve was carried out by Prodan *et al*. (2007).

The authors have used a compressor parameter model with control volumes for the compressor shell, the suction line and the discharge pipe. No further information about the valve model was given. Basic investigations of valve vibration behavior in the compressor domain can be seen in the work of Böswirth (1994).

The research object within the framework of this study is a small hermetically sealed reciprocating compressor which is typically used in household refrigeration systems. The main specifications of this compressor are as follows:

- electric power : ~ 58 [W]
- bore : 21.1 [mm]
- stroke : 16.0 [mm]
- compression ratio: 68 (ratio of volume)
- working fluid : R600a (isobutane)

The working conditions for the numerical simulation and the experimental setup are defined by the ASHRAE standards as follows:

- condensation temperature : + 55.0 [°C]
- evaporation temperature : - 23.3 [°C]
- ambient temperature : + 32.0 [°C]

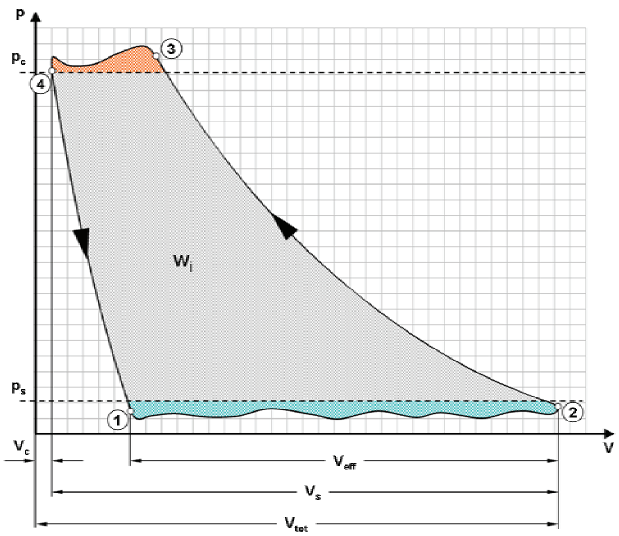

Figure1: Schematic view of the indicator diagram of an reciprocating compressor

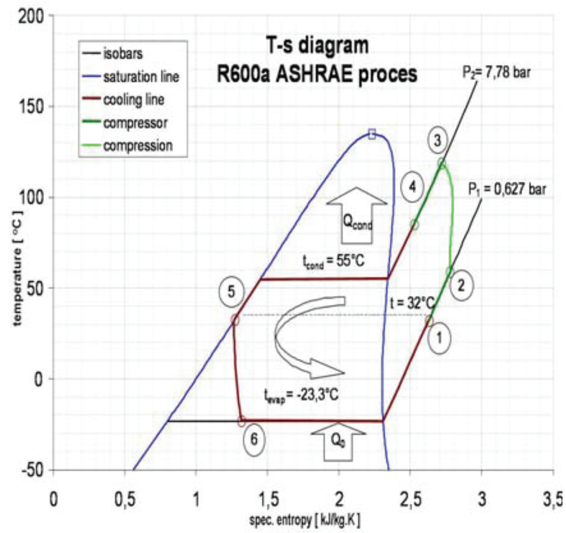

Figure 2: Schematic T,s - diagram for isobutane for the ASHRAE process

In this research study three valve parameters are investigated, i.e. effective moving mass, valve spring rate and valve spring preload. The effective moving mass as well as the valve spring rate directly determines the natural frequency of the valve, which is only varied by changing the valve thickness while the valve geometry is maintained. One additional parameter is the inlet gas bore in the valve plate. As the existing bore diameter represents the possible maximum, it is diminished to quantify its influence. A negative effect is expected.

2. NUMERICAL ASPECTS

The thermodynamic cycle calculations were performed by using the AVL BOOST software. This code can simulate a wide variety of combustion engines and provides optimised simulation algorithms for all available elements.
The investigated parameters at the intake manifold are as described in Table 1:

Table 1: Investigated parameters

<table>
<tr><th>#</th><th>parameter</th><th colspan="6">parameter value</th></tr>
<tr><td>1</td><td>natural frequency</td><td colspan="2">low</td><td colspan="2">standard</td><td colspan="2">high</td></tr>
<tr><td>2</td><td>valve preload</td><td colspan="2">standard</td><td colspan="2">halved</td><td colspan="2">doubled</td></tr>
<tr><td>3</td><td>geometrical flow area</td><td colspan="3">standard</td><td colspan="3">reduced</td></tr>
</table>

2.1 Compressor domain

The compressor calculation model (Figure 3) is designed by a set of elements, like volumes, junctions, restrictions and the cylinder. The specifications for the cylinder cover the basic dimensions of the cylinder and the crank mechanism, like bore, stroke, compression ratio, connecting rod length, and piston pin offset. All these elements are connected by pipe elements. For the pipe elements the flow is treated as one-dimensional. This means that the pressures, temperatures and flow velocities obtained from the solution of the set of full non linear gas dynamic equations, represent mean values over the cross-section of a pipe. For all other elements, the conservation of mass and energy is applied. Flow losses due to three-dimensional effects are considered by appropriate flow coefficients. Some modifications were made to implement the real gas properties of R600a (isobutane) into the program. Therefore two options are available. First, a database with suitable gas properties can be inserted. Second, a general species transport model can be selected where the physical properties of the gas species are calculated using the well known NASA polynomials species properties.

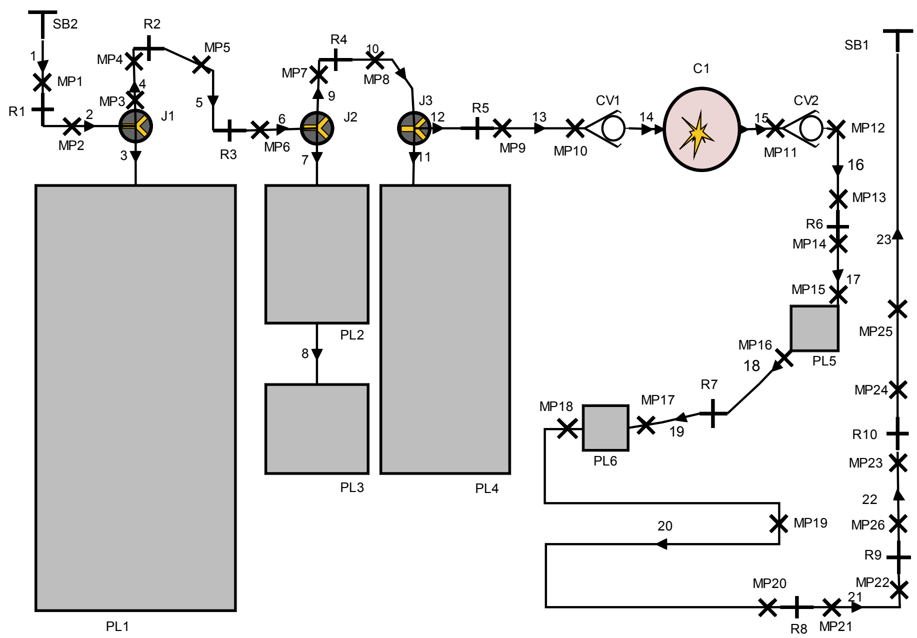

Figure 3: Numerical model of the whole compressor domain

2.2 Valve model

For the compressor reed type valve (Figure 4) an element named "check valve" is used. This element illustrates a spring-damper-mass system with one degree of freedom (Figure 5).

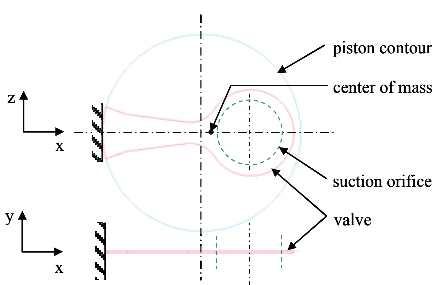

Figure 4: Compressor suction valve

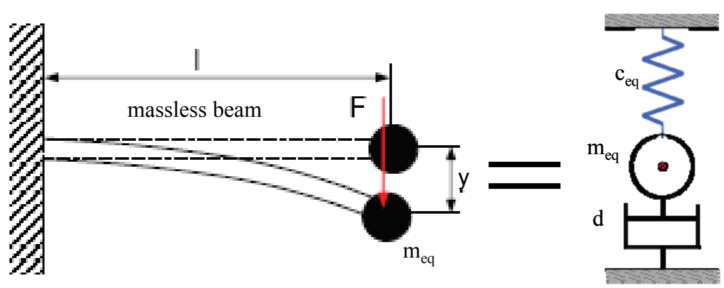

Figure 5: Corresponding check valve model

The motion of the valve is calculated from the following equation:

$$m_{eq} \cdot \ddot{y} + d \cdot \dot{y} + c_{eq} \cdot y + F_0 = \sum F_{pl} \qquad (1)$$

Equation (1) describes the movement of a pressure actuated valve considering its inertia, the spring preload, the spring stiffness and the damping factor. For the simulation of the movement of a check valve element an equivalent valve moving mass is required as well as the valve damping coefficient, the equivalent valve spring rate and the spring preload of the valve. All these parameters should be reduced to the same geometrical location on the valve surface. For the equivalent spring mass, this reduction is effected by a transformation process (Figure 5) considering an energy conservation formulation:

$$m_{eq} = \frac{J_{z,S} + m_V \cdot x_S^2}{l^2} \tag{2}$$

The equivalent spring rate can be calculated according to Equation (3) which is derived from the formulation of the natural angular frequency of a low damped oscillation system:

$$c_{eq} = \omega_0^2 \cdot m_{eq} \tag{3}$$

The simulation of the flow through the valve is based on the energy equation, the continuity equation and the formulae for the isentropic change of state:

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

To consider the particular losses resulting from multi-dimensional flow phenomena and friction losses a specification of flow coefficient is required. The flow coefficient of the orifice is defined as the ratio between the measured mass flow rate and the theoretical isentropic mass flow rate at the same certain pressure drop:

$$\alpha = \frac{\dot{m}_{meas}}{\dot{m}_{th}} = \mu \cdot \sigma; \dot{m}_{th} = A_{geo} \cdot \sqrt{2 \cdot \triangle p \cdot \rho_m} \tag{5}$$

The measured flow coefficients, the piston movement and the valve lift curve of the suction valve are shown in Figure 6.

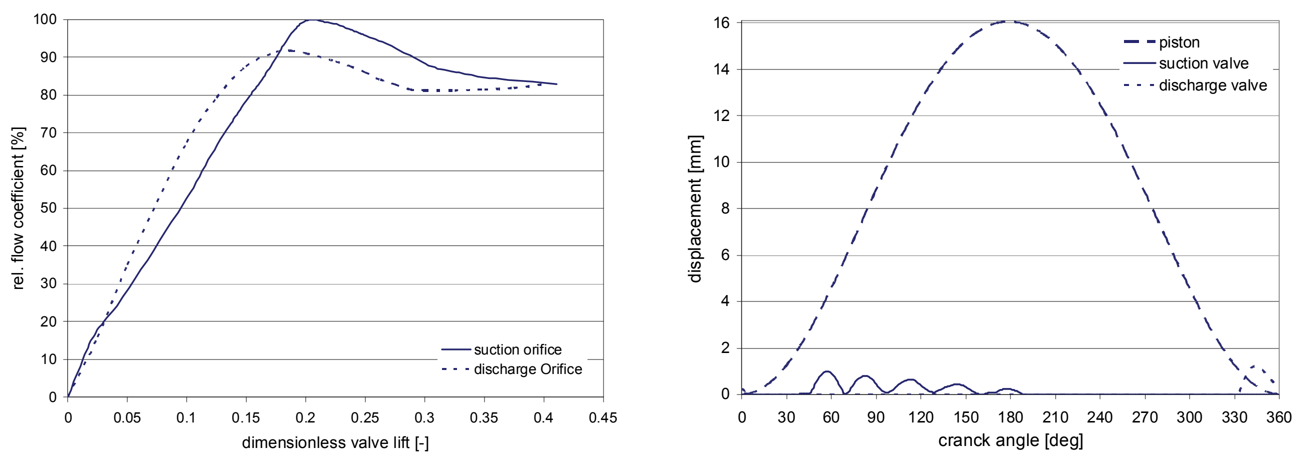

Figure 6: Left: Measured flow coefficient Right: Piston and valve displacement

It is commonly used in the field of combustion engines technologies that this flow coefficient α is divided into to parameters, namely the valve area ratio σ and the flow coefficient μ (Heywood, 1988, p. 220 ff). σ describes the ratio between the port area and the so-called "valve certain area" which varies linearly with the valve lift. μ covers the flow losses due to friction and contraction in the valve port. According to this flow coefficient α, the optimal operating point for a valve can be defined as follows, Equation (6):

$$\mathrm{y}_{\mathrm{opt}} \approx \frac{1}{4} \cdot \sqrt{\frac{4 \cdot \mathrm{A}_{\mathrm{geo}}}{\pi}} \tag{6}$$

3. EXPERIMENTAL SETUP

Experimental analyses were carried out to quantify the influence of the above mentioned parameter changes of the suction valve and to provide input parameters for the simulation model. The influence of valve parameter changes was investigated by measuring the efficiency of the compressor respectively by visualizing the valve lift curves, Nagy *et al.* (2008).

The compressor efficiency measurements were done with a running compressor on the calorimeter test bench. The calorimeter is used to provide constant boundary conditions for the standard ASHRAE test. It measures the coefficient of performance via the cooling capacity (mass flow) and the electric power of the compressor. The valve lift measurements were carried out by using the Laser Doppler Vibrometer (LDV) method.

A LDV is an instrument which can measure the velocity of a moving 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. Nagy *et al.* (2008) shows that the valve movements of a small hermetic piston compressor can be measured using an LDV. Special optical access was developed to avoid any influence which is able to change the operating conditions of the compressor. Another LDV measurement was carried out to determinate natural frequency of valves. A special fixing device was produced for the suction valve on the one hand to clamp it similar to the serial valve plate and on the other hand to enable free vibration. The laser beam of LDV was aligned to the valve surface. The valve was deflected manually and the damped, free oscillated movement was detected with the LDV. The velocity curve of this movement was analyzed by Fast Fourier Transformation (FFT) to obtain the natural frequency of the practical valve.

4. RESULTS

The results are divided into two parts. In the first part, the influence of the investigated parameters is analyzed. The second part shows the analyses of a redesigned low oscillating valve compared to the standard compressor valve.

4.1 Parameter analyses

4.1.1 Influence of natural frequency: Higher natural frequency results in a reduction of the maximum valve lift. The valve vibration and the suction valve opening are increased. Lower natural frequency results in a higher valve lift, decreased opening time and lower vibration, Figure 8

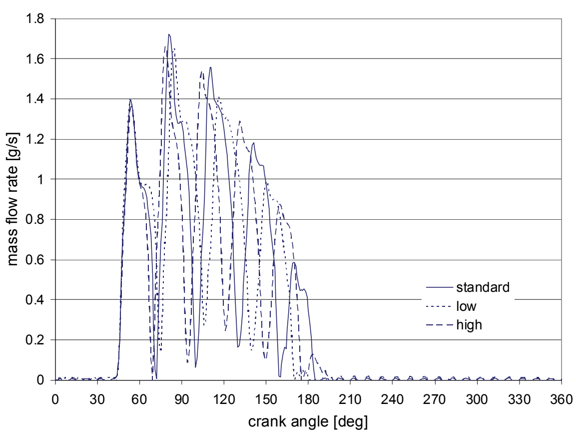

Figure 7: Mass flow rate vs. natural frequency

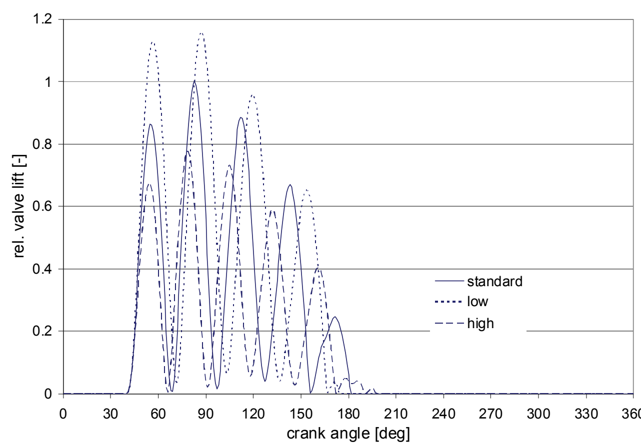

Figure 8: Suction valve lift vs. natural frequency

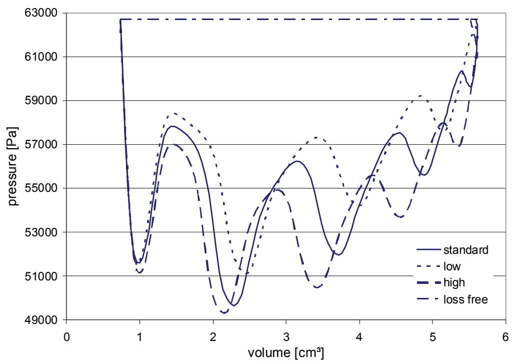

Figure 9: Pressure-volume diagram vs. natural frequency

Table 2: Table of results for natural frequency

natural frequency	simulation				measurement	
	$P_{i,s}$	$\dot{Q}_0$	P	COP	$\dot{Q}_0$	COP
[Hz]	[W]	[W]	[W]	[-]	[W]	[-]
standard	4.45	94.10	58.45	1.6099	94.33	1.6137
low	3.65	93.51	57.90	1.615	-	-
high	4.60	91.74	57.96	1.583	-	-

$P_{i.s}$ …. indicated power consumption during suction phase

The best COP is obtained at the low natural frequency and the best cooling capacity is given with the standard case, Table 2. This is due to the fact that the power consumption for the gas sucking shows a significantly smaller value in case of low natural frequency, Figure 9. The cooling capacity in this case is affected on the one hand by the reduced suction valve opening and one the other hand by an enlarged maximum valve lift, Figure 7. Furthermore, according to Equation(6), the operating point of this valve configuration is near the optimum.

4.1.2 Influence of valve spring preload: Lower spring preload leads to a higher maximum valve lift and to a longer suction valve opening, Figure 11. For the increased valve spring preload less cooling capacity and high suction power consumption were calculated, Table 3. The highest cooling capacity and the best COP were obtained at the lower valve spring preload.

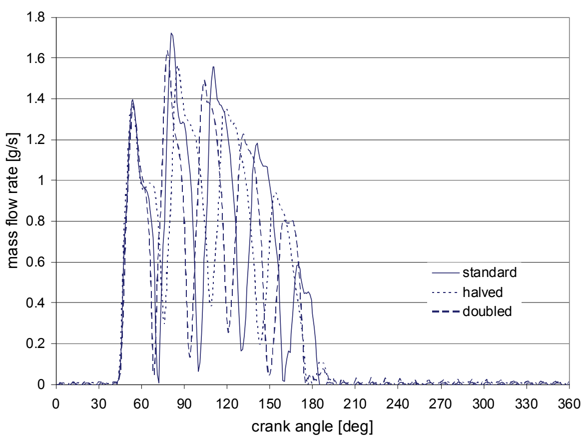

Figure 10: Mass flow rate vs. spring preload

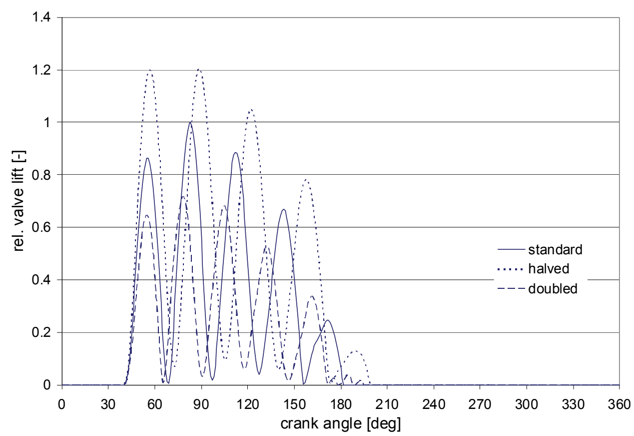

Figure 11: Suction valve lift vs. spring preload

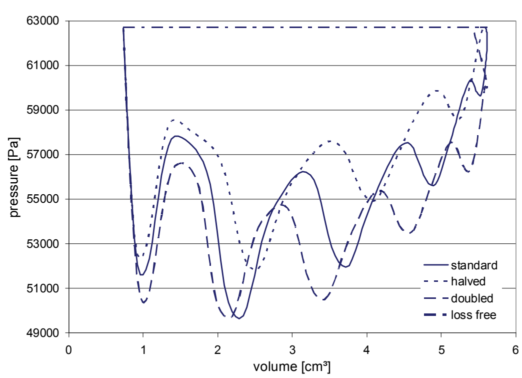

Figure 12: Pressure-volume diagram vs. spring preload

Table 3: Table of results for spring preload

spring preload	simulation				measurement	
	$P_{i,s}$	$\dot{Q}_0$	P	COP	$\dot{Q}_0$	COP
[N]	[W]	[W]	[W]	[-]	[W]	[-]
standard	4.45	94.10	58.45	1.6099	94.33	1.6137
halved	3.23	94.97	58.42	1.6257	-	-
doubled	4.77	89.71	57.07	1.5719	87.23	1.5478

$P_{i.s}$ …. indicated power consumption during suction phase

The measured cooling capacity differs from the calculated in case of doubled valve spring preload by 3%. This can be due to uncertainties in the elaborated valve preload for the test case.

4.1.3 Influence of inlet gas bore area: Table 4 shows calculation and measurement results for different gas flow areas. A reduction of the geometrical flow area of the suction gas bore of about 40% compared to the standard valve plate strongly influences the overall compressor performance. The indicated power consumption during the suction phase is hardly influenced. The value increased by 20% compared to the standard geometrical flow area. The value for the cooling capacity is decreased by 3% and the COP is reduced by 2.5%. The calculated cooling capacity differs from the measured in a range of 1.7 %.

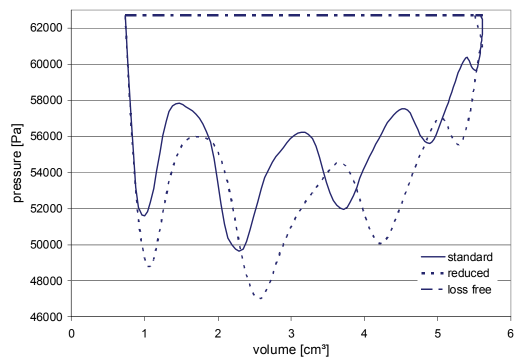

Figure 13: Pressure-volume diagram vs. flow area

Table 4: Table of results for gas bore areas

geometrical flow area	simulation				measurement	
	$P_{i,s}$	$\dot{Q}_0$	P	COP	$\dot{Q}_0$	COP
[mm²]	[W]	[W]	[W]	[-]	[W]	[-]
standard	4.45	94.10	58.45	1.6099	94.33	1.6137
reduced	5.30	91.38	58.18	1.5705	89.85	1.5584

$P_{i.s}$ …. indicated power consumption during suction phase

4.2 Valve improvement

A theoretical improvement of the standard valve design was done by varying the main valve parameters to avoid high valve vibrations. Also a valve retainer was implemented to achieve the same maximum valve lift as the standard valve, Figure 14. For this theoretical valve the valve spring rate must be significantly reduced to approximately 20 % compared to the standard valve. For the equivalent valve mass, this value is reduced to about 25%. Figure 14 shows the valve lift behavior of the improved valve design versus a standard one.

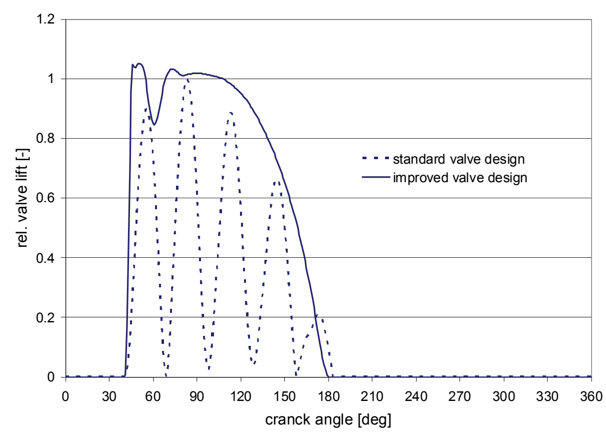

Figure 14: Valve lift diagram; optimized valve design vs. standard valve

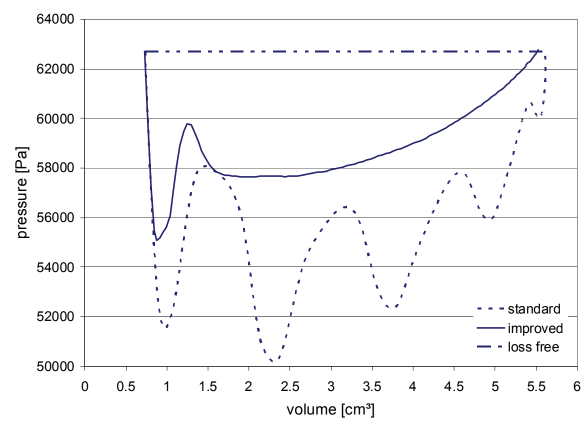

Figure 15: Pressure-volume diagram during suction phase

The indicated power consumption during the suction phase for both valve types is calculated from the pressure-volume diagram in Figure 15. The values are displayed in Table 5. The suction power consumption is decreased by 36%. Finally an increase of 2 % on the overall compressor performance is obtained.

Table 5: Results for influence of valve vibration

valve type	suction power consumption	mass flow rate	power consumption	COP
standard	4.2 [W]	280.4 [mg/s]	58.4 [W]	1.614
improved	2.7 [W]	285.3 [mg/s]	58.2 [W]	1.647

5. CONCLUSION

A numerical simulation model was used to perform the thermodynamic cycle calculation of hermetic reciprocating compressors. For the calculation of the valve lift behavior a valve model was introduced. The purpose of this research study was to show the valve lift behavior of a suction valve considering different valve parameters to understand valve vibration, which is believed to affect the delivery rate of the gas flow and compressor performance. It was shown that the valve lift behavior is strictly coupled to the basic valve parameters and the port geometry. Due to this fact an optimization of the investigated parameters seem to be important to improve the overall compressor performance.

A big influence of the flow coefficient was found during this study. The decline of the flow coefficient leads to a higher valve lift and vice versa. Therefore the port geometry and the main valve parameters should be coordinated. A new value was introduced, Equation(6), which allows the valve designer to check if the valve operates in an optimal way.

Finally, it was shown that a reduction of valve vibration for the studied cases include the potential of a COP improvement of approximately 2 %.

NOMENCLATURE

symbol	comment	unit	symbol	comment	unit
C	cylinder element	[-]	l	effective length	[m]
COP	coefficient of performance	[-]	m_V	mobile valve mass	[kg]
CV	check valve element	[-]	m_{eq}	equivalent valve mass	[kg]
J	junction element	[-]	$\dot{m}$	mass flow rate	[kg/s]
MP	measuring point element	[-]	$\dot{m}_{th}$	isentropic mass flow rate	[kg/s]
PL	plenum element	[-]	p	downstream static pressure	[Pa]
R	restriction element	[-]	p_0	upstream stagnation pressure	[Pa]
SB	system boundary element	[-]	Δp	pressure difference	[Pa]
A_{geo}	geometrical flow area	[m²]	R	gas constant	[kJ/kg,K]
c_{eq}	equivalent valve spring rate	[N/m]	T	temperature	[K]
d	valve damping coefficient	[N·s/m]	y	valve lift	[m]
F	force	[N]	α	flow coefficient	[-]
F_0	valve spring preload	[N]	ρ_m	mean density	[kg/m³]
F_{pl}	impressed plate force	[N]	κ	ratio of specific heat	[kJ/kg,K]
$J_{z,S}$	mass moment of inertia	[kg·m²]	ω_0	eigen circular frequency	[rad/s]

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